Coupling a Network HVAC Model to a Computational Fluid Dynamics Model Using Large Eddy Simulation

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Why model HVAC systems?

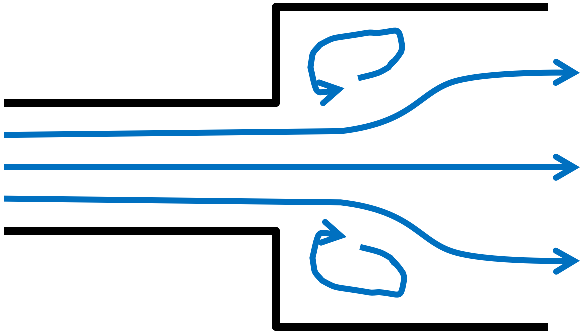
- Model smoke movement in systems with recirculation
- Exhaust and supply behavior changes due to pressurization from a fire
- Smoke movement through ducts

FDS v5.5 HVAC capabilities

- Define an inlet or outlet mass (or volume) flow with a predefined flow rate, temperature, and species.
- Simple quadratic fan model to adjust flow rate based on the local pressure.
- Cannot couple an inlet to an outlet
- Cannot couple a single fan to multiple inlets or outlets.

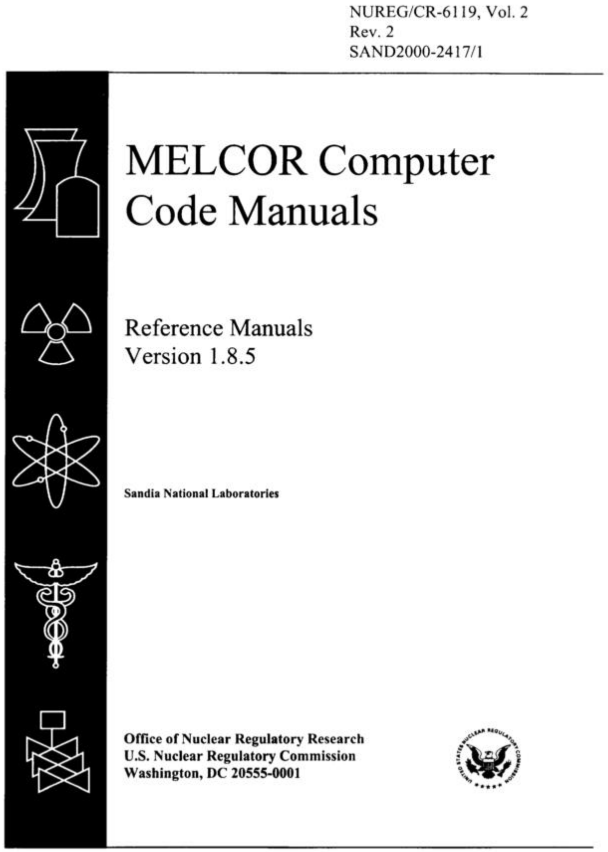
Why not mesh ducts?

- Expense - Determining form losses requires fine resolution of duct fittings
- Validity - User would need to validate that accurate losses were determined for all HVAC components



Solution approach

- Network HVAC solver based on MELCOR algorithm (US NRC containment safety code)
- Indirect coupling to FDS



NUREG/CR-6119, Vol. 2
Rev. 2
SAND2000-2417/1

MELCOR Computer
Code Manuals

Reference Manuals
Version 1.8.5

Sandia National Laboratories

Office of Nuclear Regulatory Research
U.S. Nuclear Regulatory Commission
Washington, DC 20555-0001

MELCOR conservation equations

- Conservation of Mass

$$\sum_{j\text{ connected to }i} \rho_j u_j A_j = 0$$

- Conservation of Energy

$$\sum_{j\text{ connected to }i} \rho_j h_j u_j A_j = 0$$

- Conservation of momentum

$$\rho_j L_j \frac{du_j}{dt} = \left(P_i - P_k\right) + \left(\rho g \Delta z\right)_j + \Delta P_j - \frac{1}{2} K_j \left|u_j\right| u_j$$

A	flow area
h	enthalpy
g	gravity
i, k	node
j	duct
K	loss coefficient
L	duct length
P	pressure
u	velocity
t	time
ρ	Density
ΔP	fan pressure
Δz	elevation change

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MELCOR momentum equation

$$u_j^n = u_j^{n-1} + \frac{\Delta t^n}{\rho_j L_j}\left(\tilde{P}_i^n - \tilde{P}_k^n + \Delta P_j + (\rho g \Delta z)_j^{n-1}\right) - \frac{K_j}{2L_j}\left(\left|u_j^{n-} - u_j^{n+}\right| u_j^n - \left|u_j^{n+}\right| u_j^{n-}\right)$$

~ indicates extrapolated of end of time step pressure

n is the time step

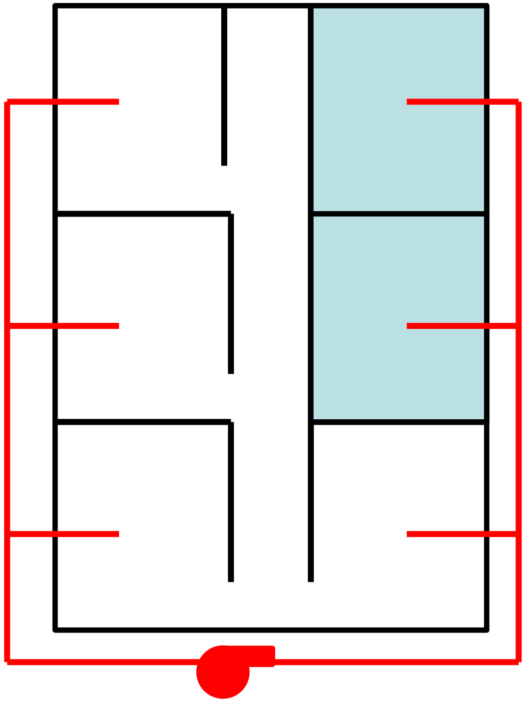
$n-$ is the previous iteration value

$n+$ is the previous iteration if flow direction the same or 0 if flow direction changes

Since K is a function of flow direction, the linearization aids in stability when pressure forces are low

Extrapolated pressure (1/2)

- For a duct connected to a room, the end of time step pressure is a function of all the flows in and out of that room and any other room which is connected to it.
- To account for this, a prediction of the end of time step pressure is made using the velocities of any duct that is connected to the room directly or indirectly



Duct flow into any un-shaded compartment will impact the pressure in every other un-shaded compartment

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Extrapolated pressure (2/2)

In the divergence routine FDS computes:

$$\frac{dP_m^n}{dt} = \left(\oint_{\Omega_m} \mathrm{D}\, dV - \oint_{d\Omega_m} u \cdot dS \right) \Big/ \oint_{\Omega_m} P\, dV$$

where *m* is a pressure zone

$$\tilde{P}_m^n = P_m^{n-1} + \frac{dP_m^n}{dt} \Delta t^n$$

$$\frac{dP_m^n}{dt} = \frac{dP_m^n}{dt}_{\,divg.f90} + \left(\sum_{j \text{ in } m} u_j^{n-1} A_j - \sum_{j \text{ in } m} u_j^n A_j \right) \Big/ \oint_{\Omega_m} P\, dV$$

$$\frac{dP_m^n}{dt} = \frac{dP_m^n}{dt}_{\,hvac.f90} - \sum_{j \text{ in } m} u_j^n A_j \Big/ \oint_{\Omega_m} P\, dV$$

Fully discretized momentum

$$u_j^n\left(1+\frac{K_j}{2L_j}\left|u_j^{n-}-u_j^{n+}\right|\right)-$$

$$\frac{\Delta t^{n^2}}{\rho_j L_j}\left(\sum_{j\text{ connectedto i}} u_j^n A_j \Big/ \int_{\Omega_i} P\,dV-\sum_{j\text{ connectedto k}} u_j^n A_j \Big/ \int_{\Omega_k} P\,dV\right)=$$

$$\frac{\Delta t^n}{\rho_j L_j}\left(\tilde{P}_{i\ \ hvac.f90}^{n}-\tilde{P}_{k\ \ hvac.f90}^{n}+\Delta P_j+\left(\rho g\Delta z\right)_j^{n-1}\right)+\frac{K_j}{2L_j}\left|u_j^{n+}\right|u_j^{n-}$$

- If *i* or *k* is an internal node, no pressure extrapolation is done and the pressure is solved for directly
- Densities are upstream

Wall BC

$$u_{wall} = \frac{\dot{m}_{duct}}{\rho_{wall} A_{wall}}$$

$$\rho_{wall} = \frac{\overline{W}_{wall} P}{R T_{wall}}$$

$$T_{wall} = T_{duct}$$

$$\overline{W}_{wall} = f\left(Y_{wall}\right)$$

$$Y_{wall,i} = \frac{\dot{m}_i'' + \frac{(\rho D)_{wall}}{\Delta x} Y_{gas,i}}{\frac{(\rho D)_{wall}}{\Delta x_{wall}} + u_{wall}\rho_{wall}}$$

- ρ, u, and Y are coupled
- Iterate solution
- In a typical calculation, values rarely change quickly and little iteration is required

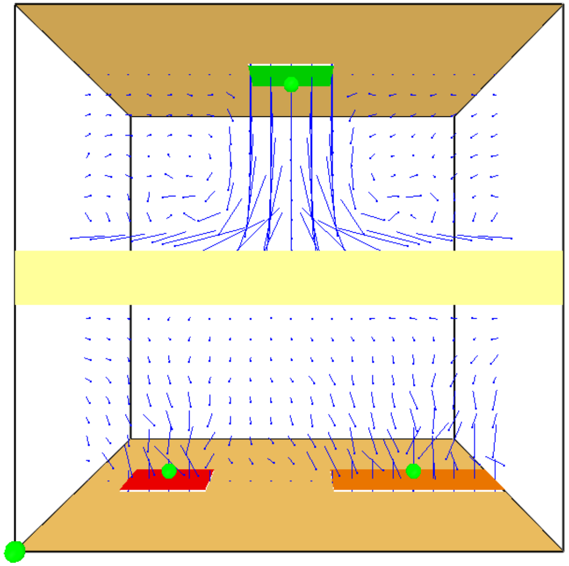
Solution method

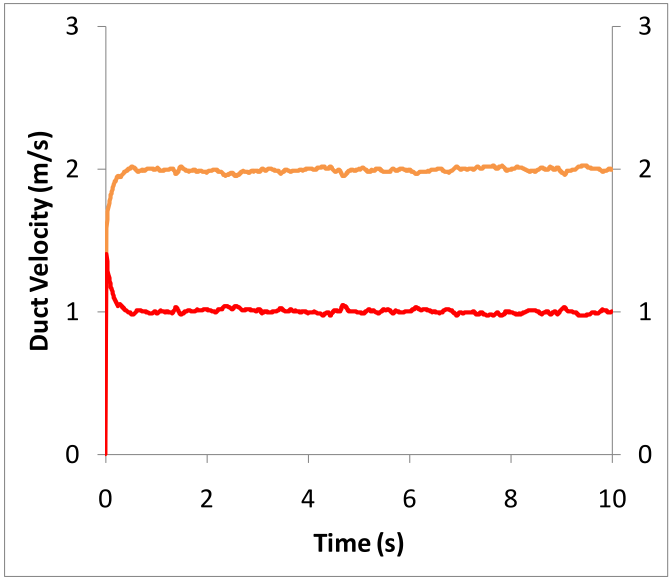
1. Determine ρ, T, Y, and P at external duct nodes (average over VENT)
2. Solve for u
3. Update ρ, T, and Y at internal nodes
4. Check for convergence of u and that net mass flow is 0 for internal nodes
5. Return to step 2 if un-converged

Coupling to FDS pressure solution

- Pressure solution for HVAC is not coupled to pressure solution for FDS domain
- Typical FDS time step is << 1 s
- Momentum length of ducts limits rate of change of duct solution
- Volume flow at duct connections to domain change “slowly” and error from not coupling will be small

Verification Case 1





- Green : 0.3 m^3/s exhaust
- Red duct: Loss of 16
- Orange duct: Loss of 4
- Ducts 0.1 m^2

$$K_{red}u_{red}^2 = K_{orange}u_{orange}^2$$

$$16u_{red}^2 = 4u_{orange}^2$$

$$\frac{u_{orange}}{u_{red}} = 2$$

- FDS = 2

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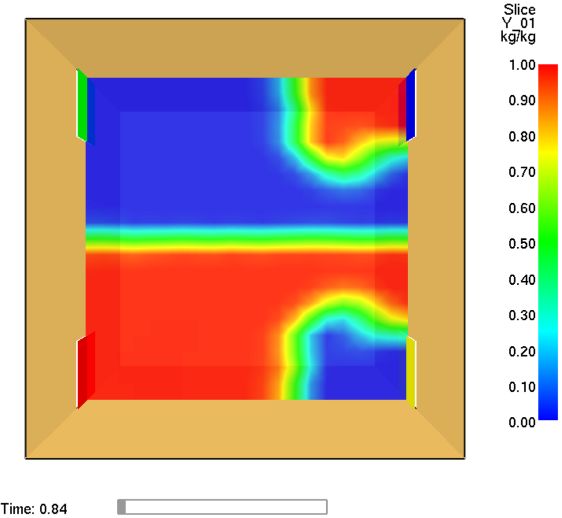
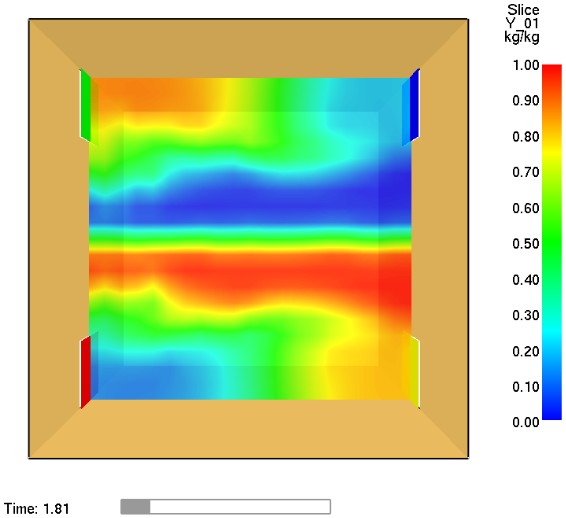
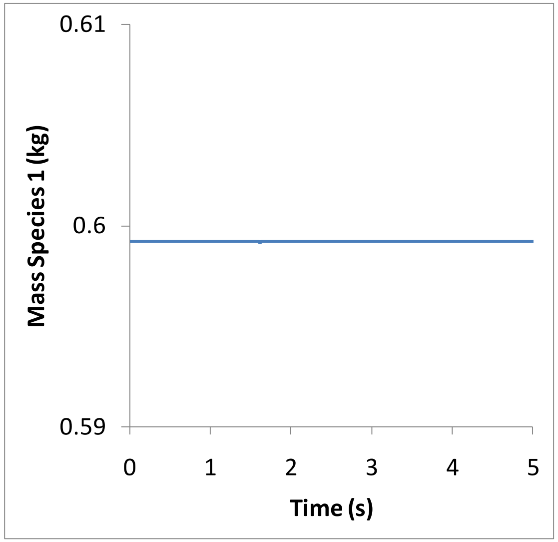
Verification Case 2

- Bottom half of compartment Species 1
- Red / Blue + Green / Yellow are Suction / Discharge







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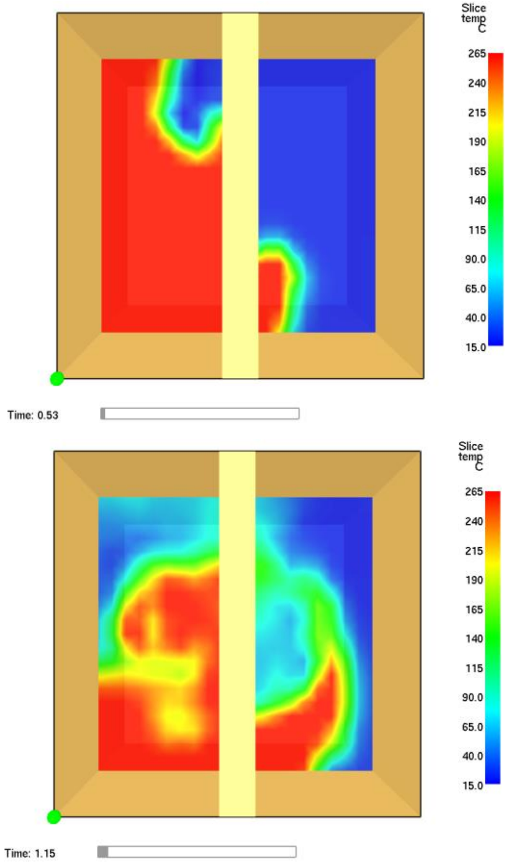
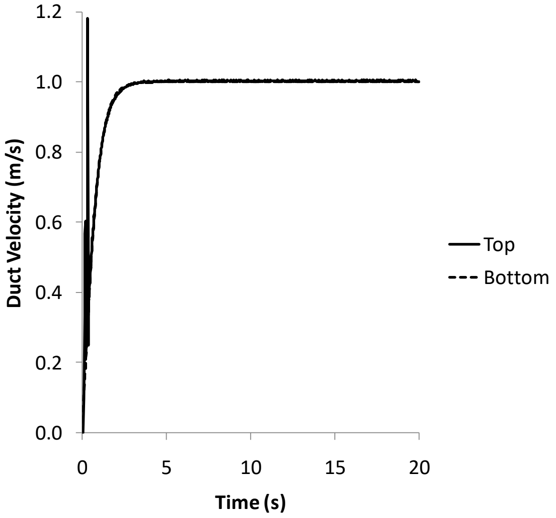
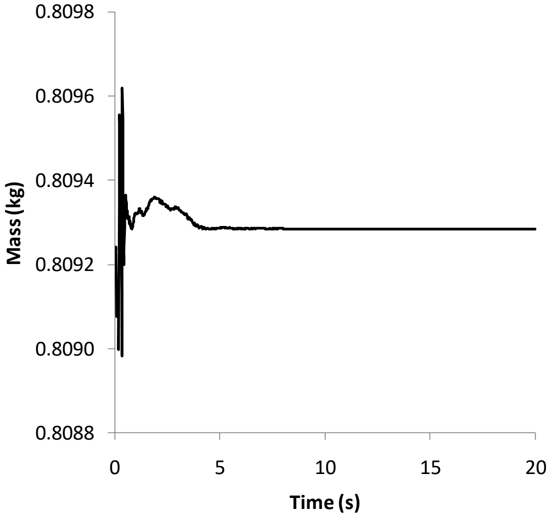
Verification Case 3

- Left half at 313.15 °C, adiabatic walls
- Top duct – 0.1 m^2, right to left, flow unspecified
- Bottom duct – 0.1 m^2, left to right, 0.1 m^3/s







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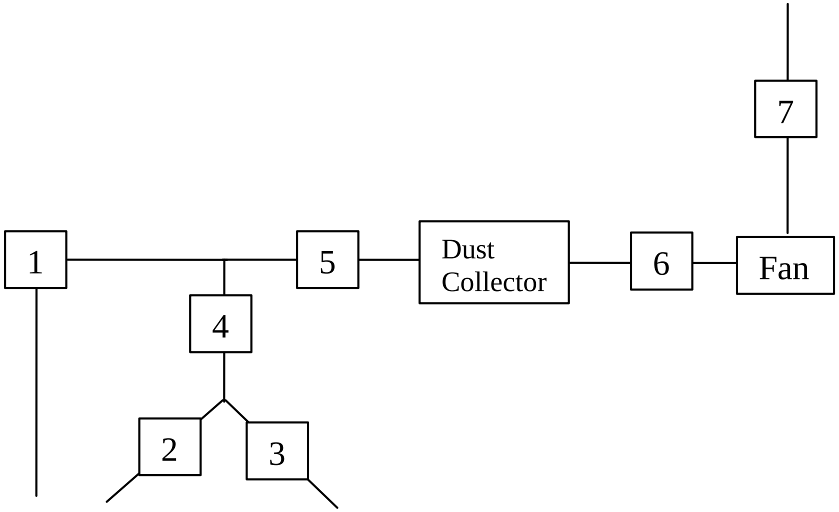
ASHRAE Fundamentals #7

Duct	ASHRAE ΔP	FDS ΔP	Error %
1	739	731	-1.1
2	458	449	-1.9
3	281	282	0.3
4	124	124	-0.2
5	746	744	-0.4
6	32	33	3.3
7	318	321	-0.5

Note: ASHRAE has fixed density in ducts, FDS density varies slightly due to pressure drops.



Metal working exhaust system:
3 pieces of equipment with a dust collector

Quadratic fan curve plus fitting and duct losses

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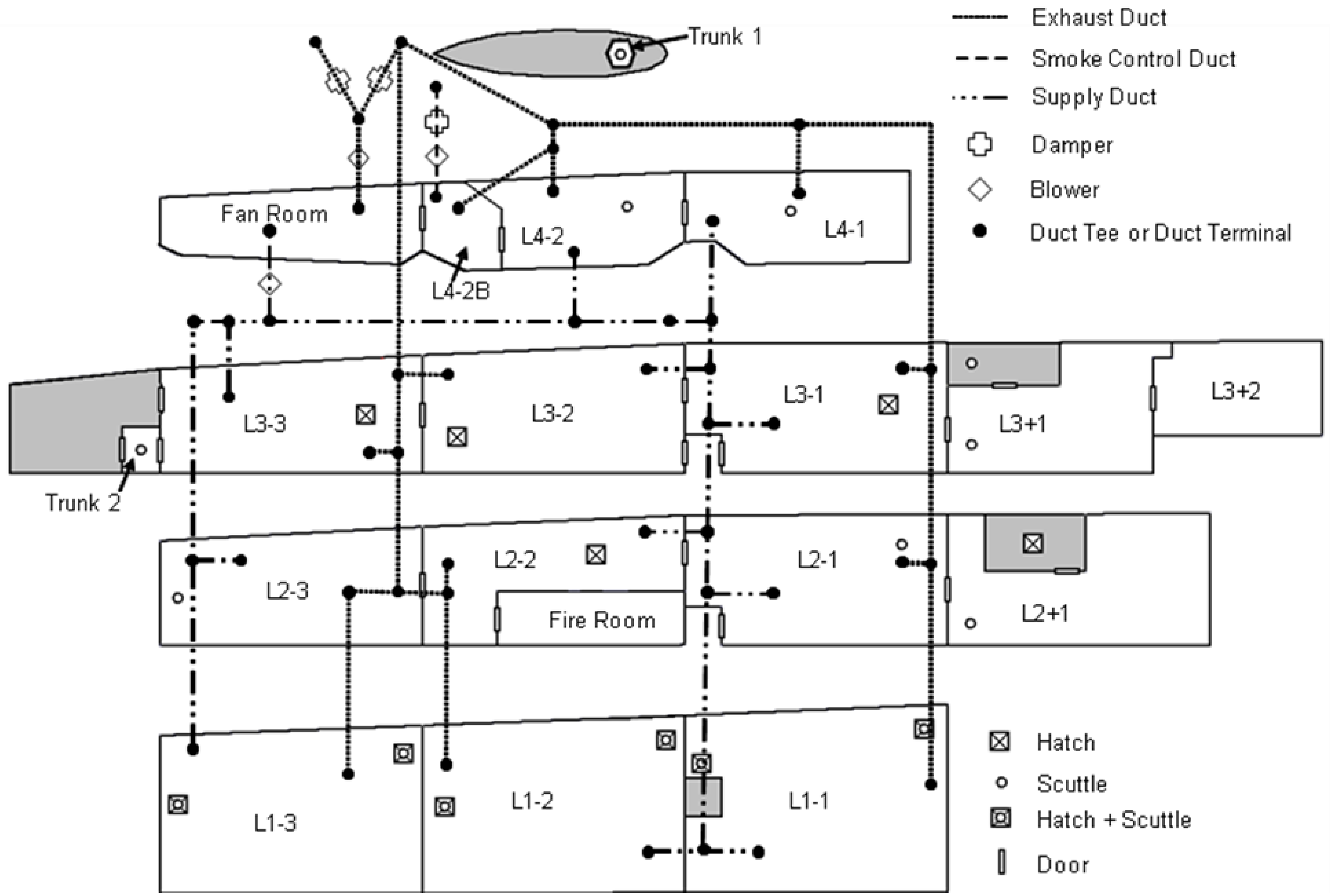
Confined Space Facility (1/3)

- 23 compartments
- 4 levels
- 20 wall / ceiling openings
- 129 HVAC components
  - Supply system takes suction from fan room and discharges to all compartments
  - Exhaust
    - Takes suction from all compartments and discharges to fan room
    - With damper re-alignment allows fresh air to be drawn into fan room
  - Smoke control takes suction from nav room and discharges outside

Confined Space Facility (2/3)



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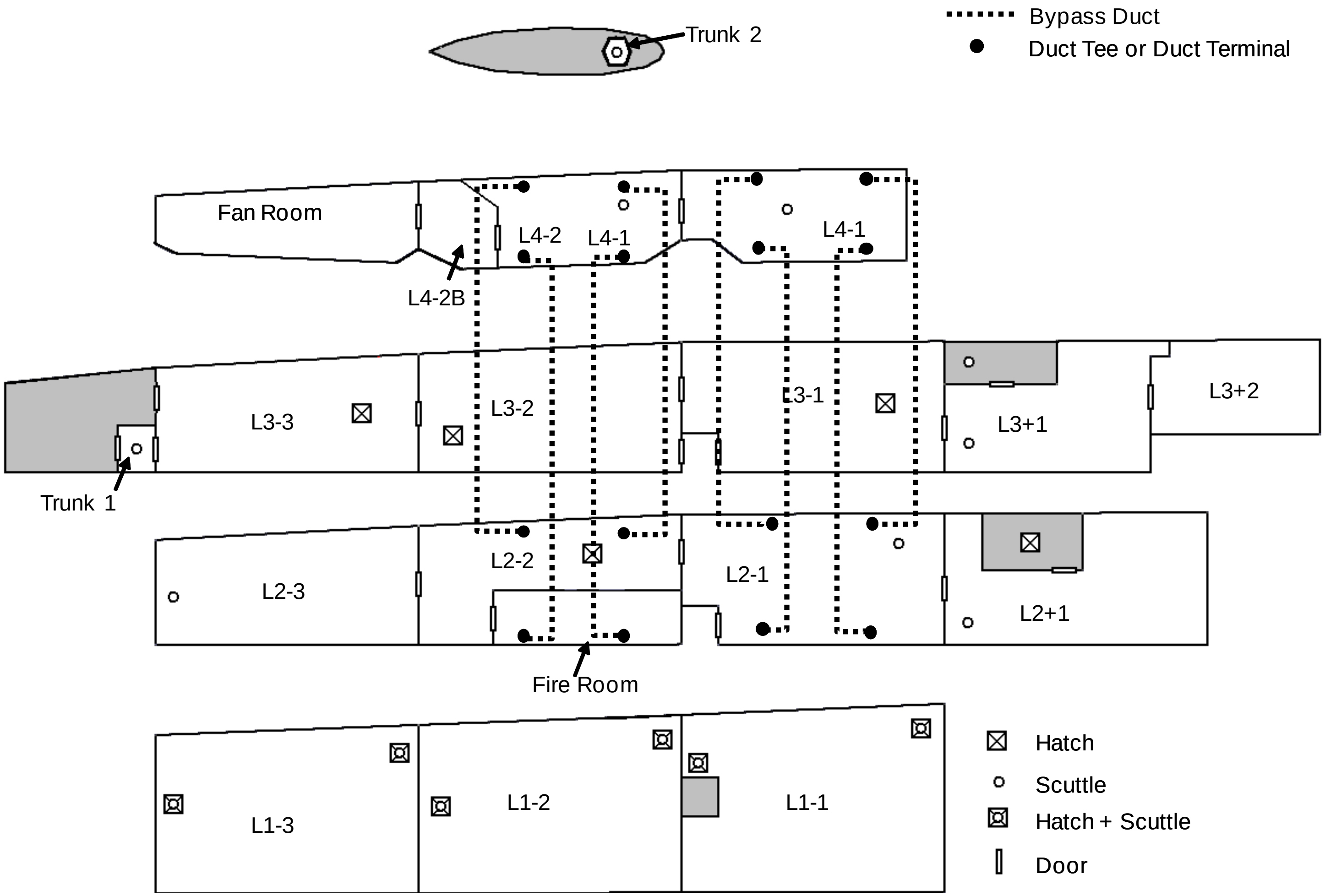
Confined Space Facility (3/3)



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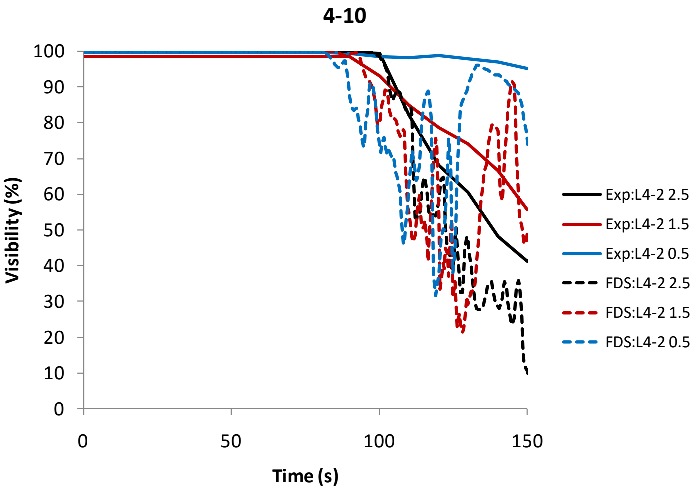
Test Descriptions

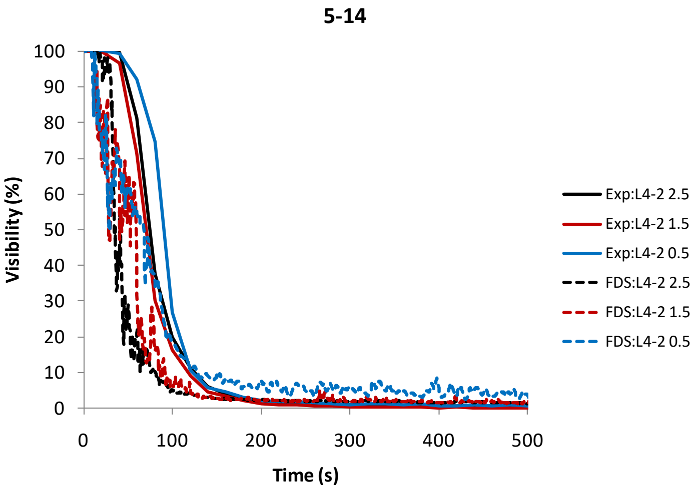
- 4-10: 1.05 m diameter diesel fire in fire room
  - No HVAC
  - All internal closures opened, no external closures opened
- 5-14: 0.68 m diameter diesel fire in fire room
  - Supply and exhaust fans on then off at 1 minute
  - Frame bay ducts installed
  - One external closure opened
  - Most internal doors closed (many with ventilation grills)
  - 1 minute realign exhaust and turn on smoke control

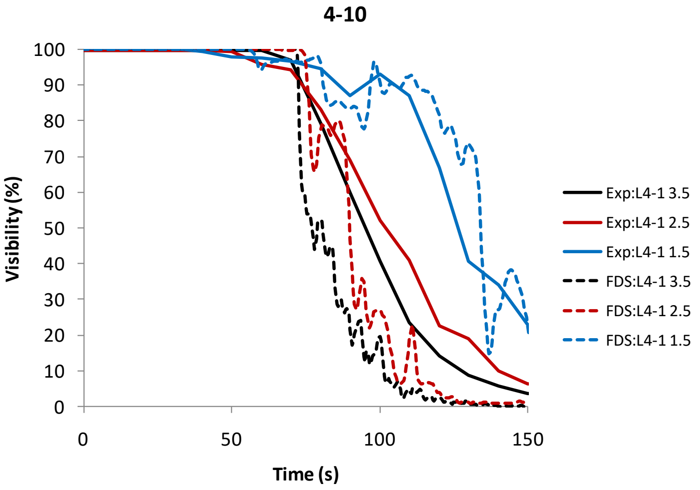
Model Inputs

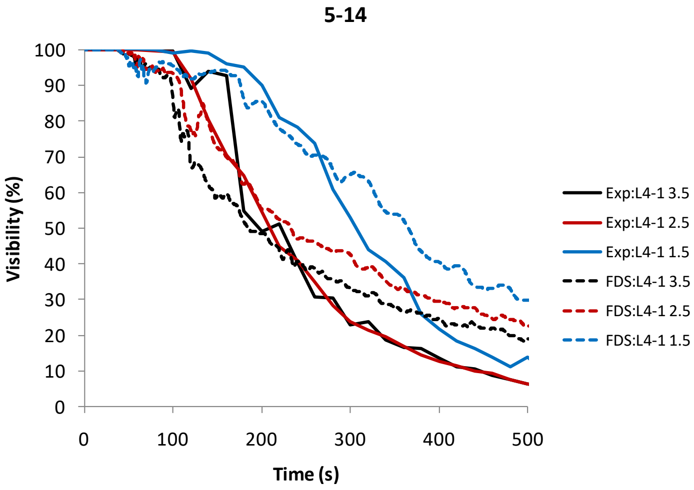
- HVAC losses taken from ASHRAE tables based on as-built drawings of ductwork
- Fire size based on load cell under fuel pan (measurement very noisy)
- Fan curves from manufacturer’s data adjusted for fan frequency

Upper Level Visibility

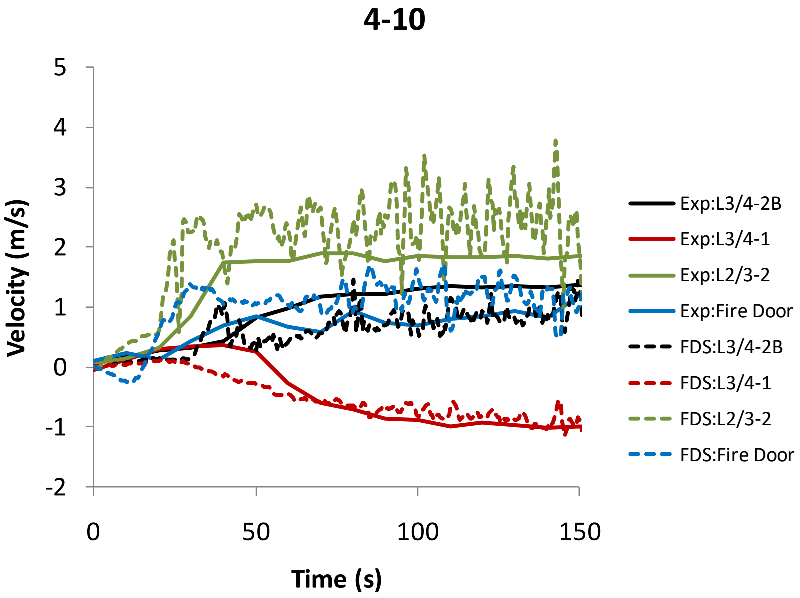

4-10

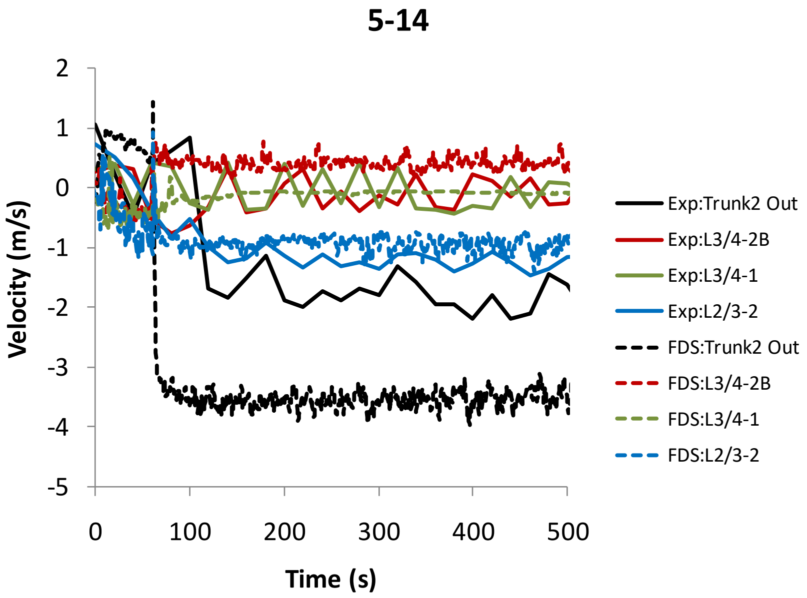

5-14


4-10


5-14

Door / Hatch Velocities





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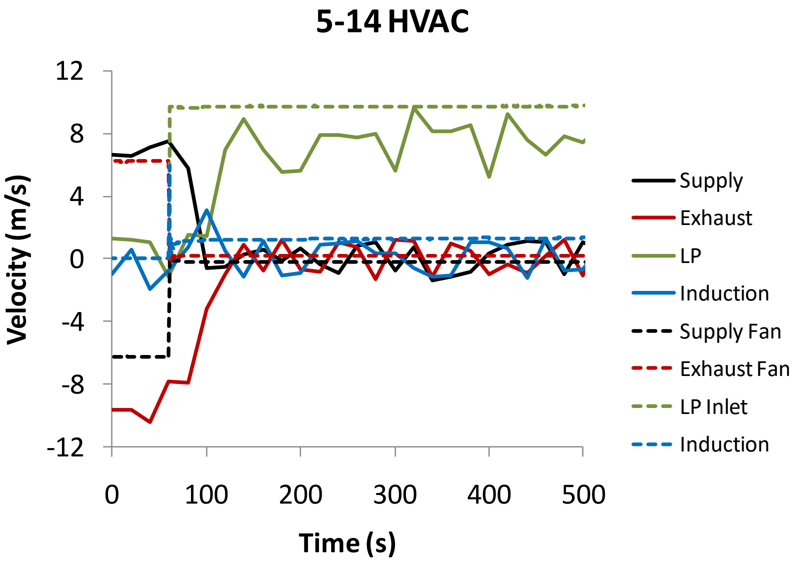
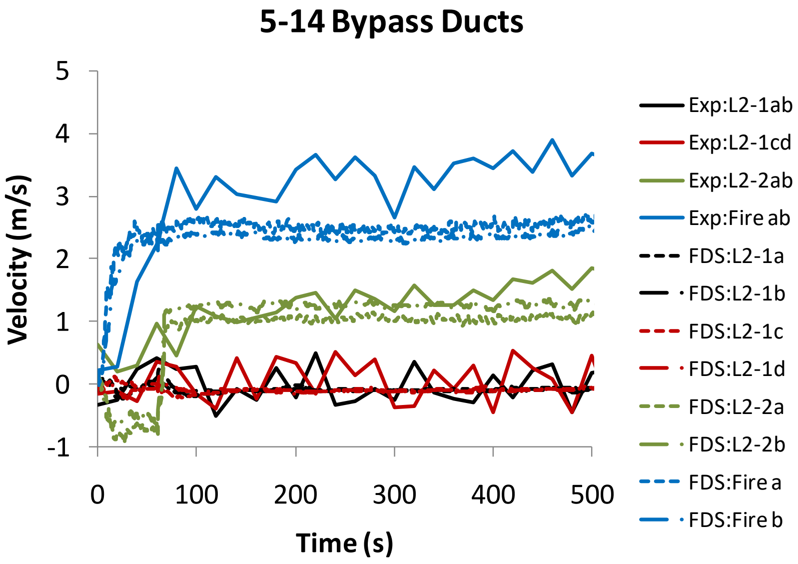
Duct Velocities





Filtration

- Filters, especially HEPA, prone to clogging from soot
- Flow loss can be expressed as a clean loss (no loading loss) plus a loss due to loading ($K_{loading}$)

$$K_{loading} = \begin{array}{l} \text{linear function: } c\sum_n \chi_n L_n \\ \text{RAMP function: } f\left(\sum_n \chi_n L_n\right) \end{array}$$

- Where L_n is the species loading and χ_n is a multiplier

Filtration

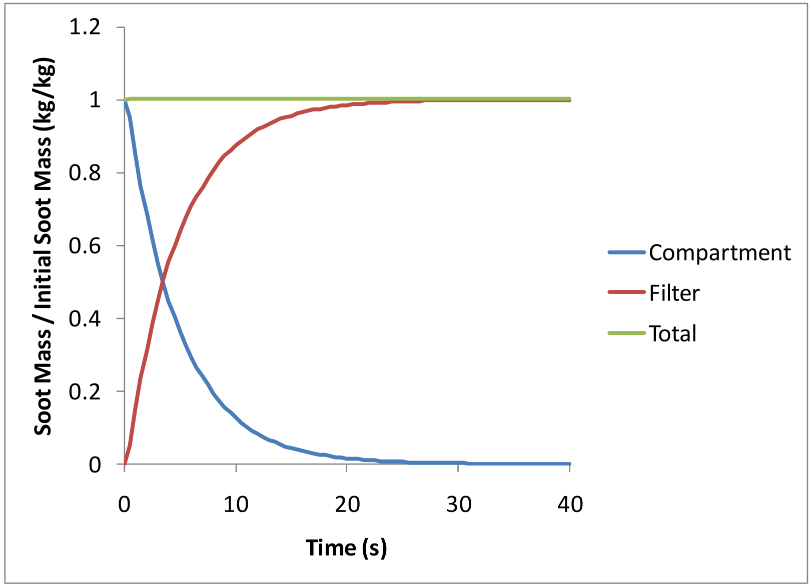
- A filter implemented as a special class of a duct node
- Filter removal rate computed as

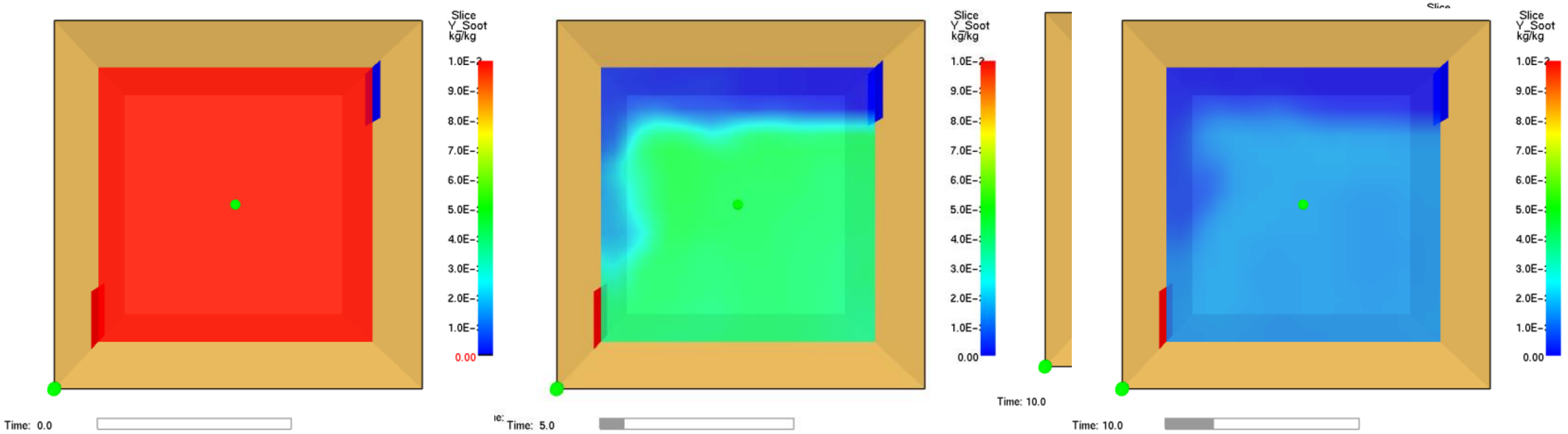
$$\dot{L}_n = \rho_d u_d A_d Z_{d,n} \varepsilon_n$$

- Where ε_n is a species removal efficiency
- Removal rate is added as a loss term to the duct node mass conservation equation

Filtration Example

- 1 m^3 compartment with 1 % soot mass fraction.
- HVAC system with a 100 % efficient filter flowing 0.2 m^3/s





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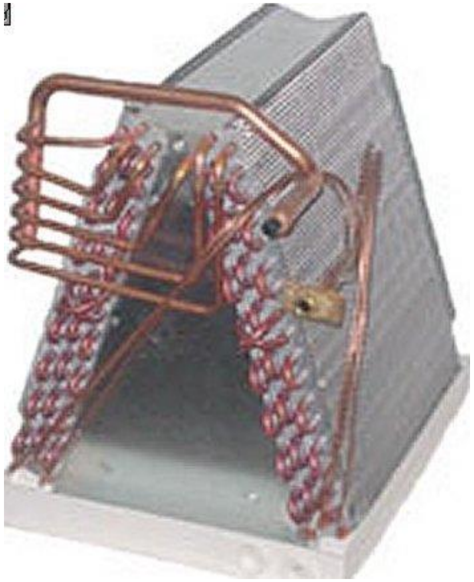
Aircoil

- Heating / Cooling within a duct
- Theoretical maximum heat exchange when exiting air temperature = exiting fluid temperature



$$T_{out} = \frac{c_{p,gas}\dot{m}_{gas}T_{in,gas} + c_{p,coolant}\dot{m}_{coolant}T_{in,coolant}}{c_{p,gas}\dot{m}_{gas} + c_{p,coolant}\dot{m}_{coolant}}$$

$$\dot{q}_{\max} = c_{p,coolant}\dot{m}_{coolant}\left(T_{out} - T_{in,coolant}\right)$$

- Actual heat exchange given by an efficiency, η.

$$\dot{q}_{actual} = \eta\dot{q}_{\max}$$

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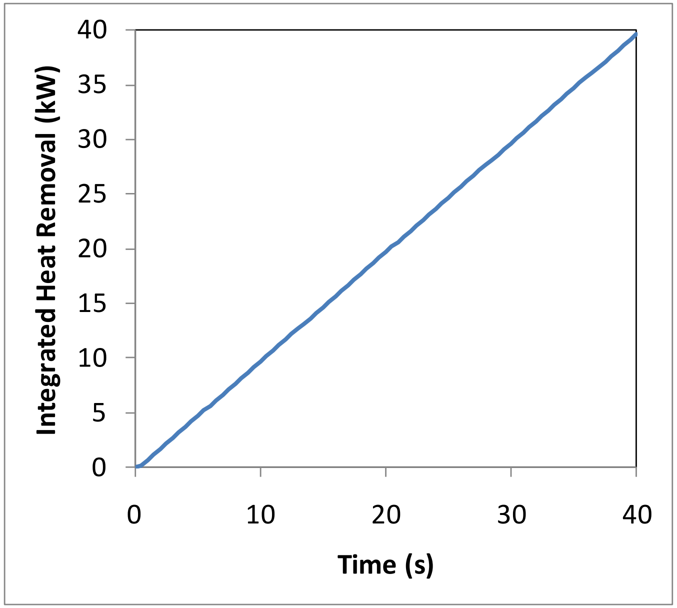
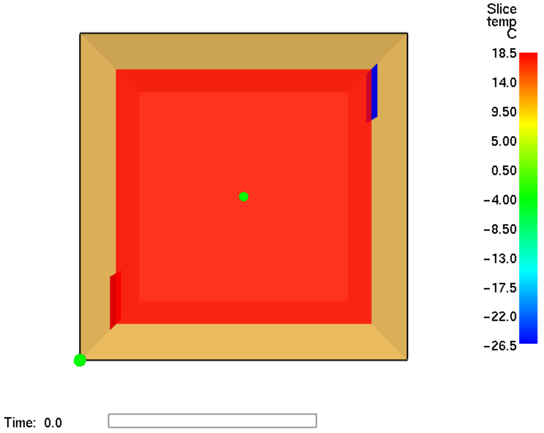
Aircoil

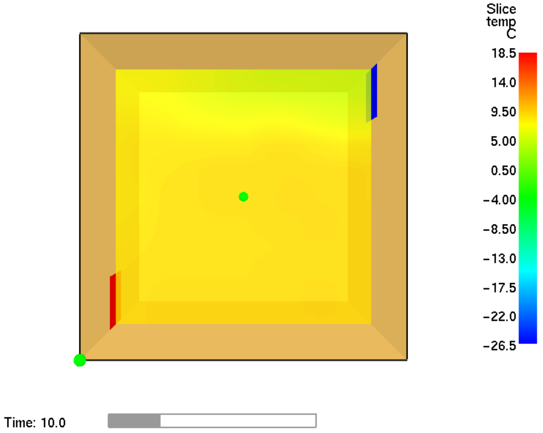
- An aircoil is implemented as a component of a duct
- The downstream node energy balance (used to compute node temperature and density) is updated to reflect heat removal / addition of the aircoil

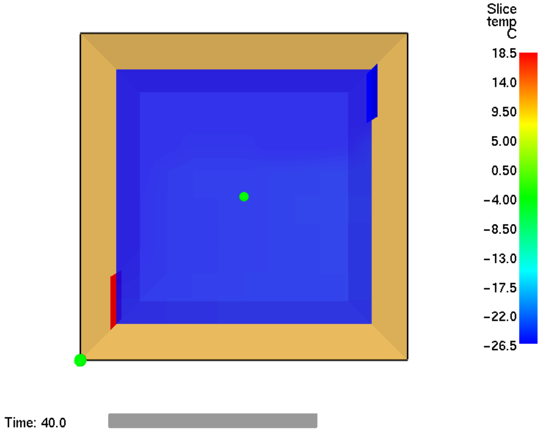
Aircoil Example

- 1 m^3 compartment
- HVAC system with a 1 kW coiling coil, flowing 0.2 m^3/s









Potential additional capabilities

- Transient operation of dampers with position dependent losses
- Condensation / evaporation on filters
- Transient operation of fans
  - Spin up / spin down
  - Variable motor speed
- Duct wall heat transfer