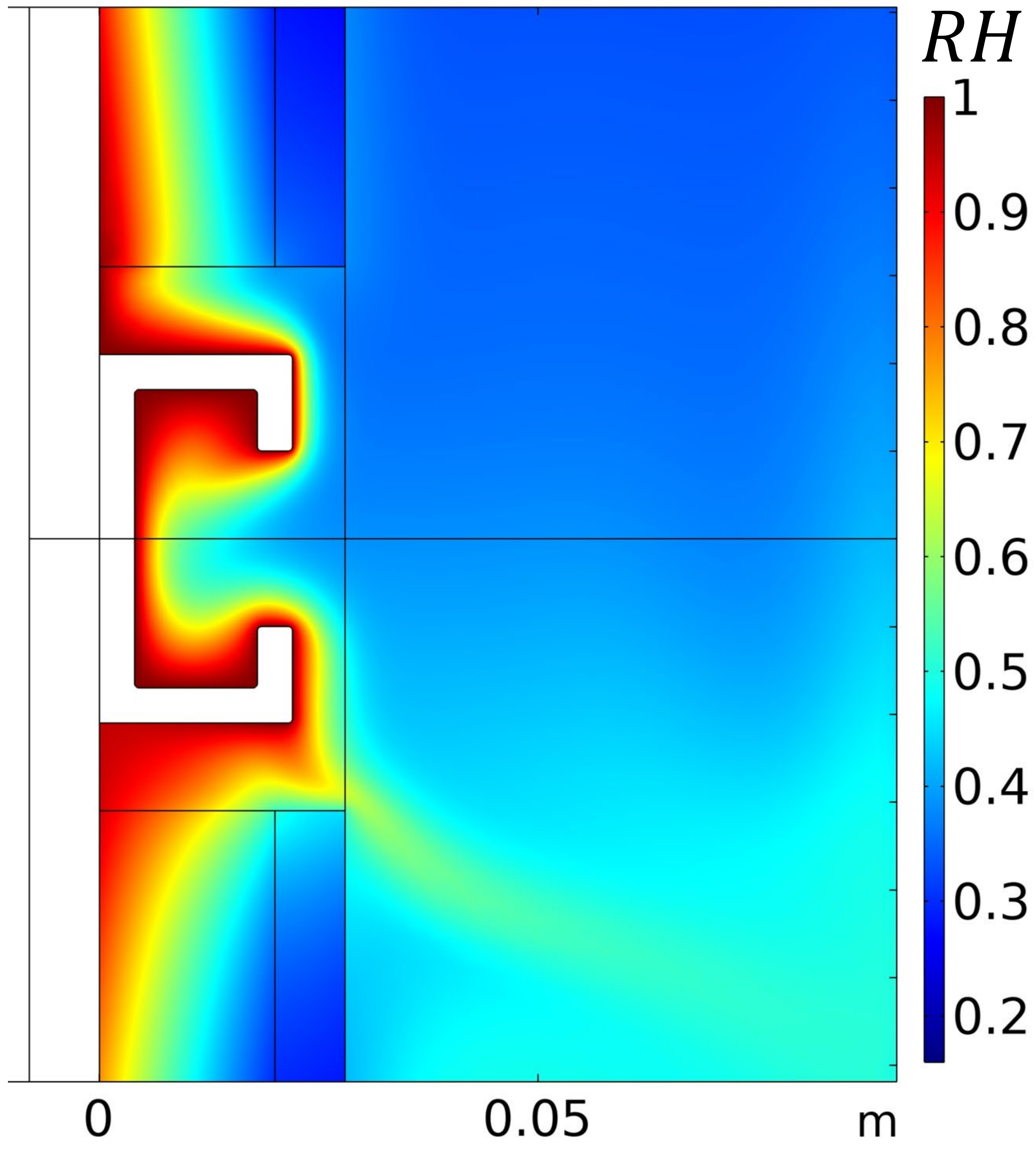
HSLU Lucerne University of Applied Sciences and Arts

Simulating condensate layers on thermal bridges in train walls

Felix Lüönd and Tom Graf



Engineering and Architecture
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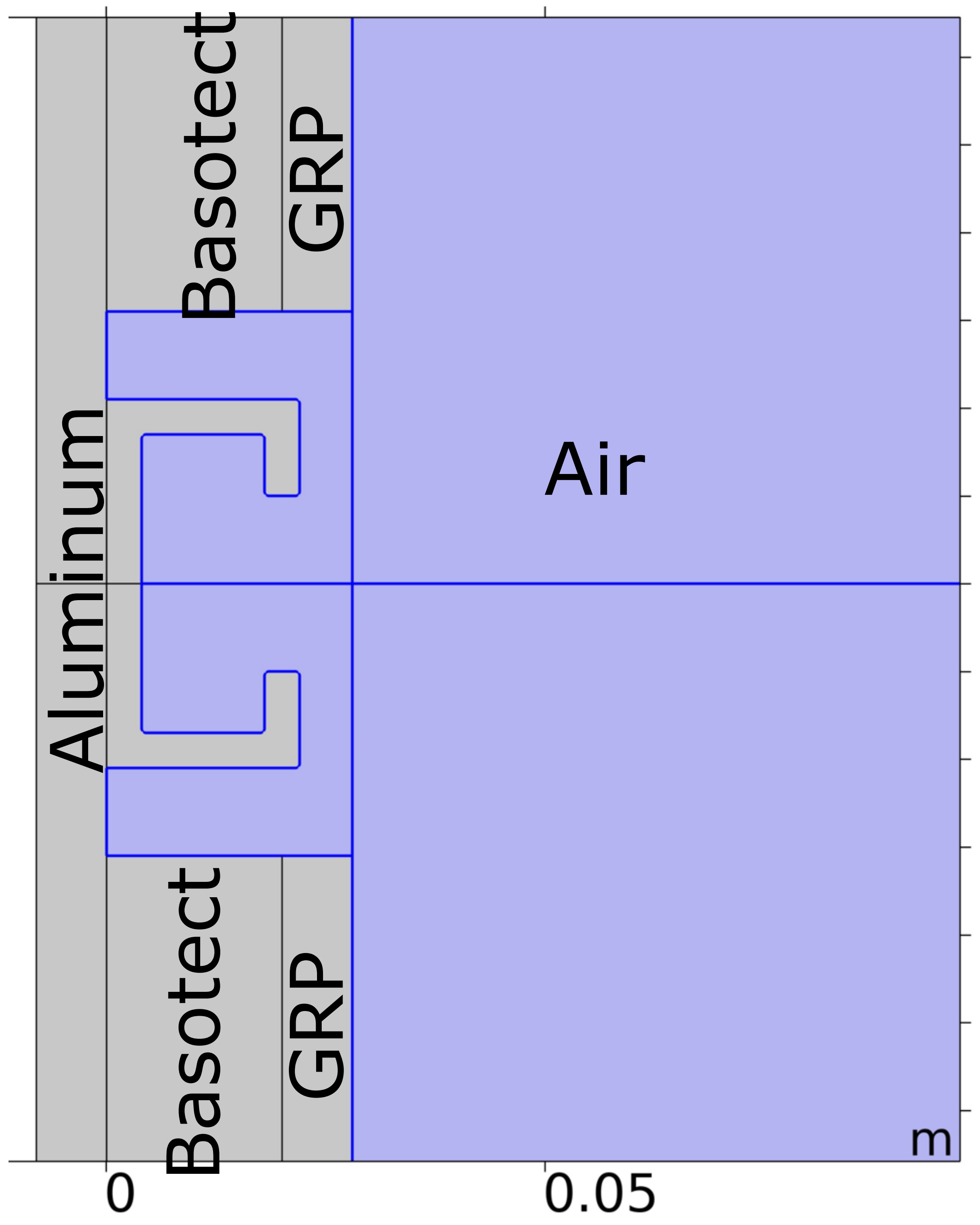
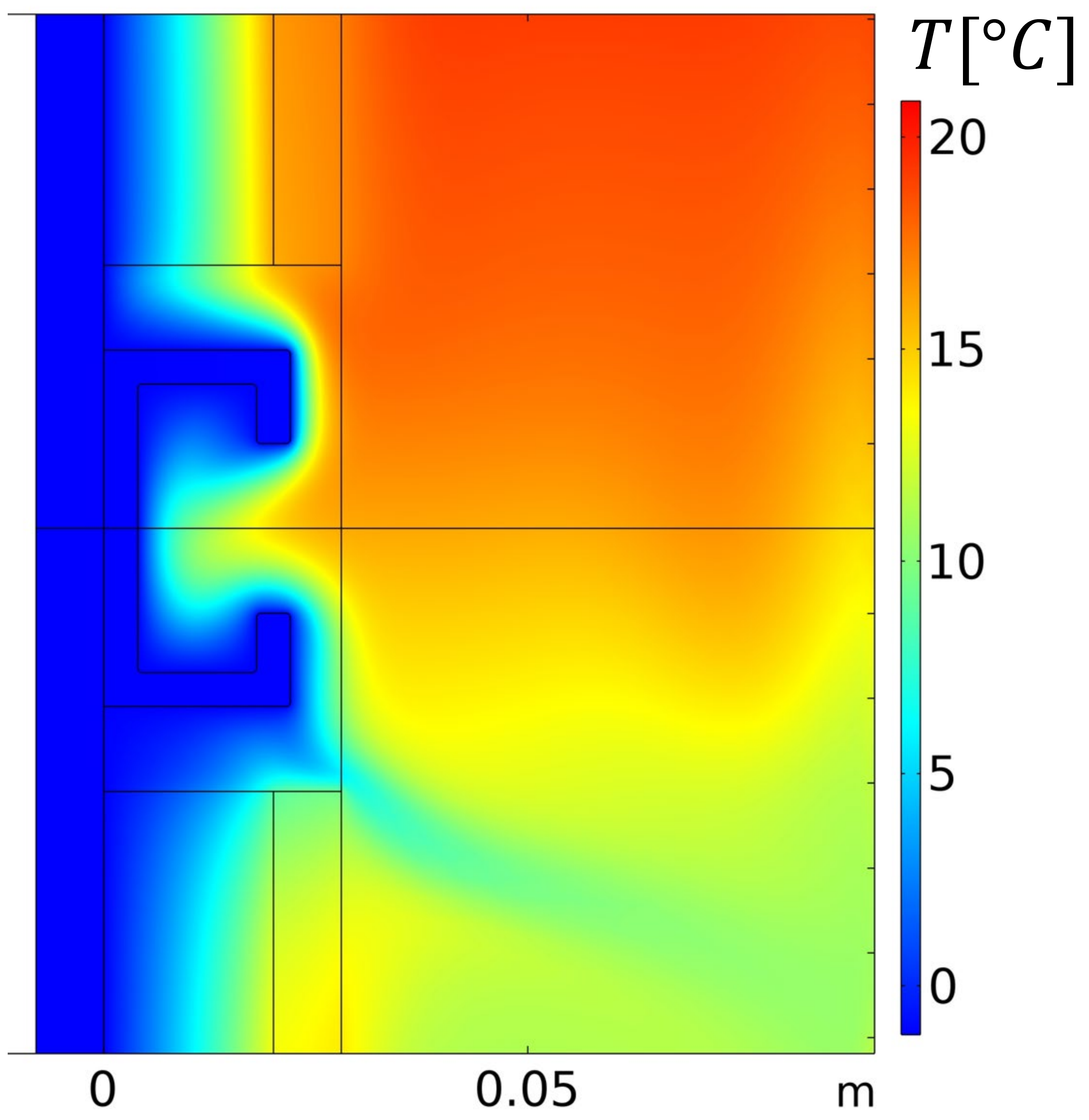
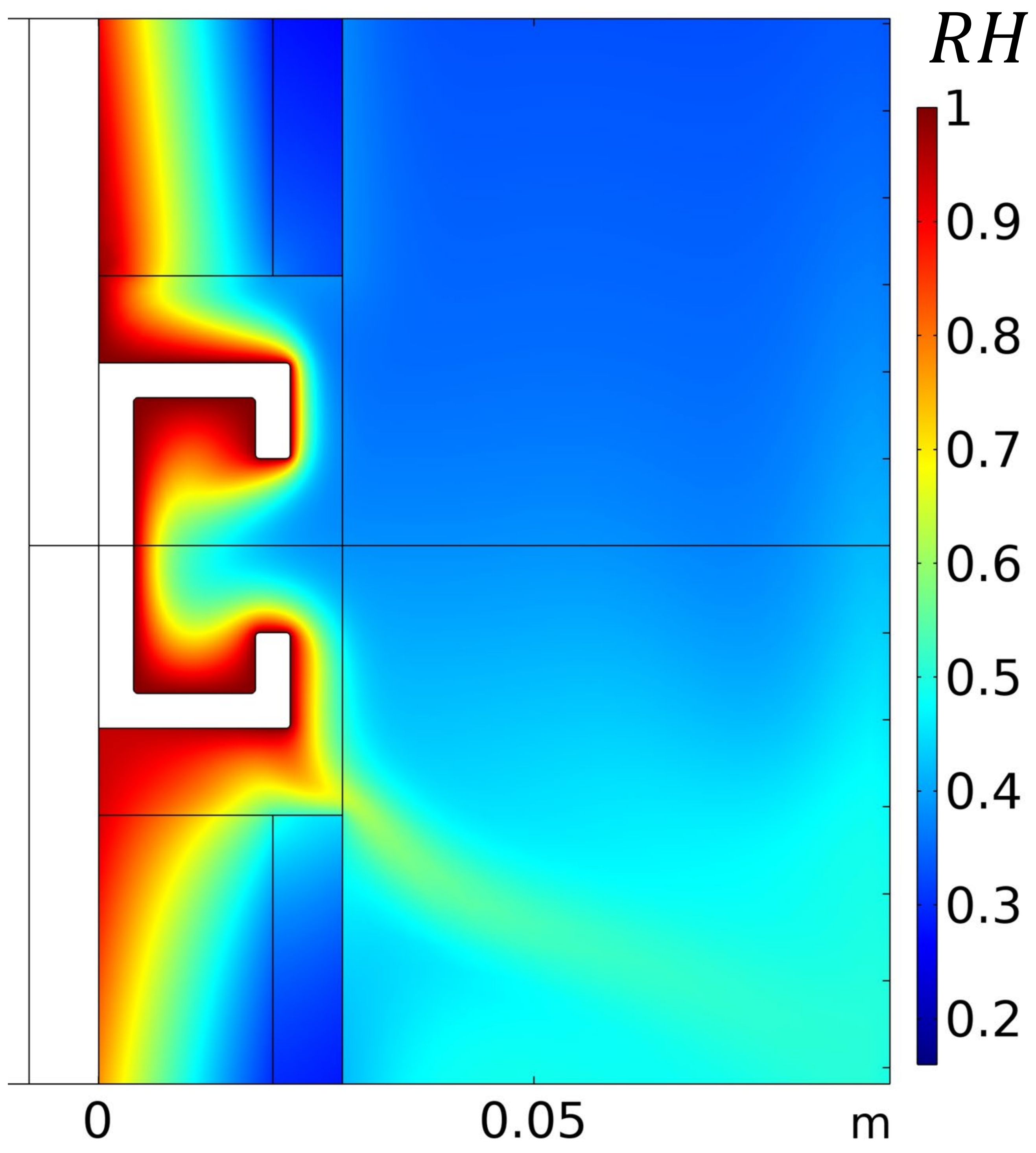
FH Zentralschweiz

Goal: Calculate the amount and distribution of water condensing on the surface of thermal bridges in train walls.

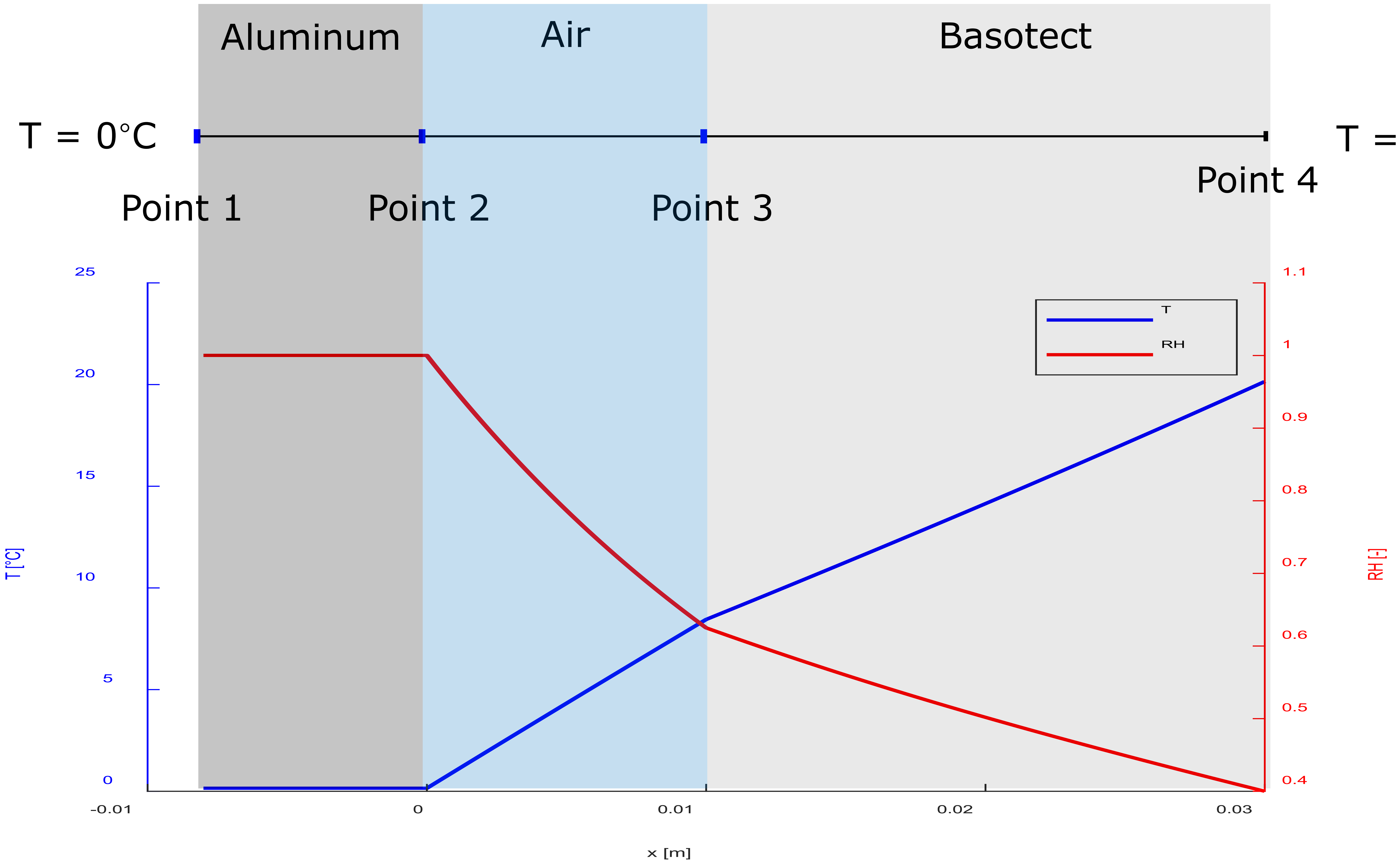






- C-rack as an example of a frequent thermal bridge in train walls
- C-rack usually has direct thermal contact to the metallic hull of the car body
- C-rack is in contact with the warm cabin air
- → high relative humidity and condensing moisture on the surface of the C-rack.

1D test case for different models with purely diffusive moisture transport



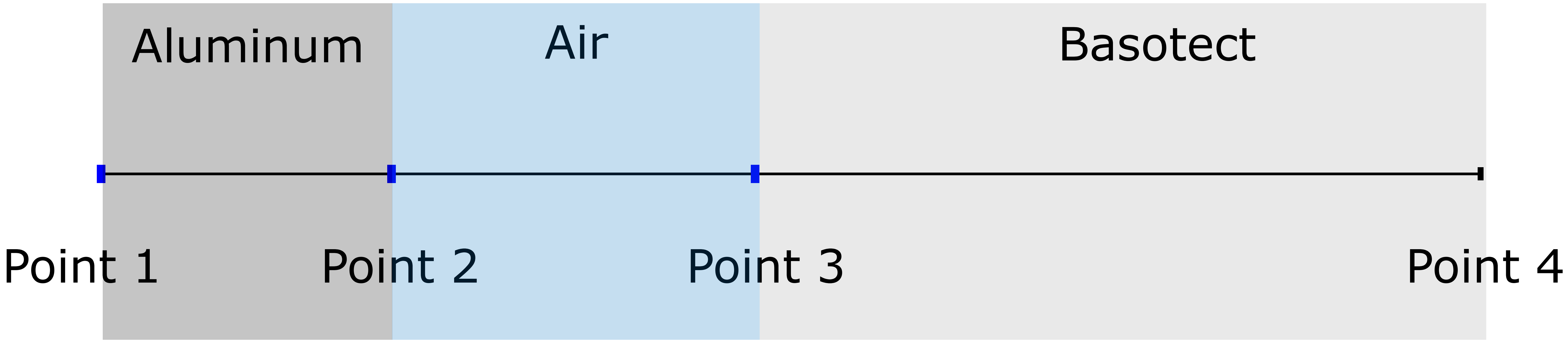
Typical steady state temperature and relative humidity distribution

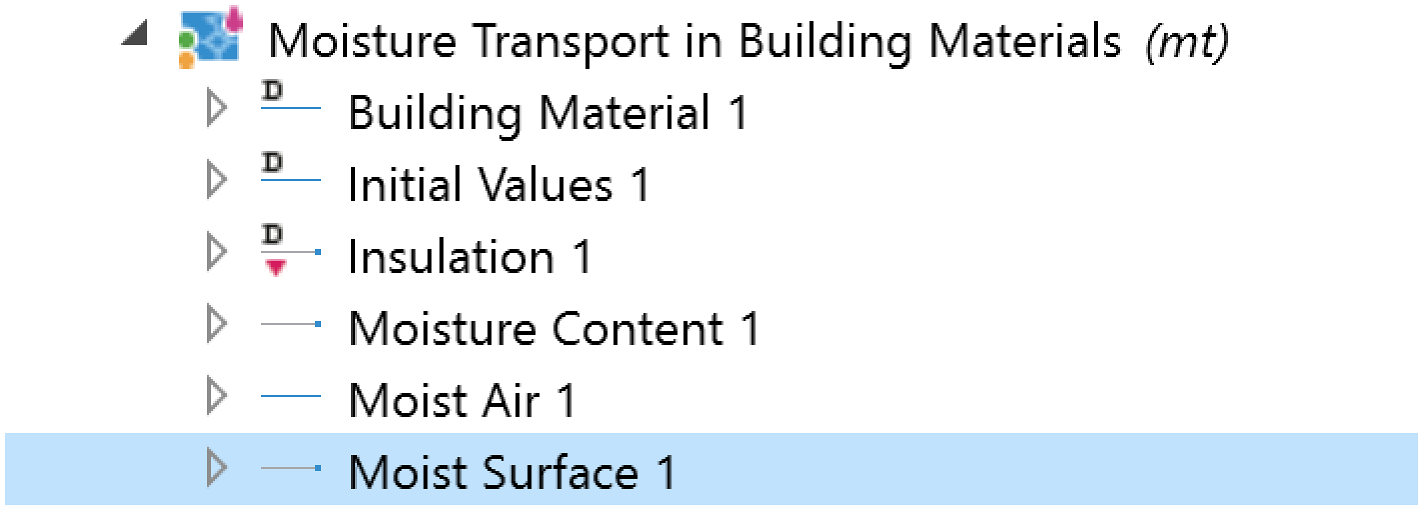
Three ways to implement a condensed water layer in COMSOL

1D test case for different layer growth models: Purely diffusive moisture transport

1. COMSOL provided: «Moist surface» node in the «Moisture Transport in Building Materials» physics
2. Custom model: «Film growth model», calculate condensate layer thickness from vapor flux with custom ODE
3. Custom model: «Sponge model», air layer modeled as a Building material with artificial moisture capacity

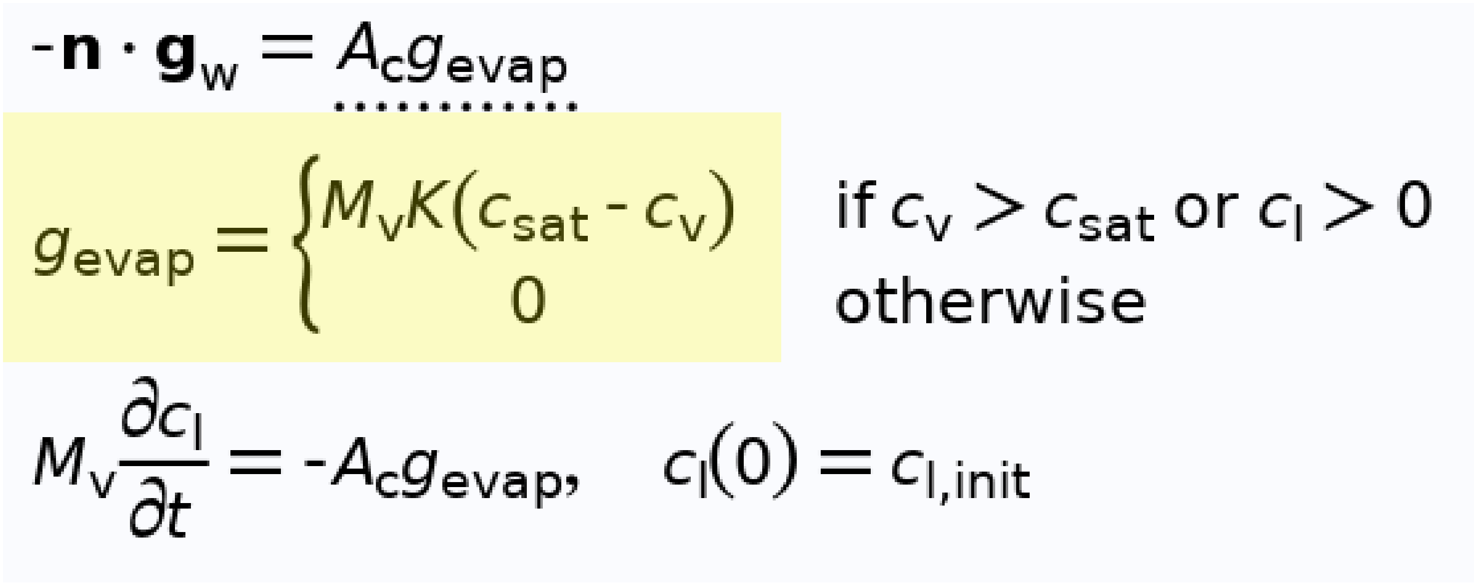
1. COMSOL provided: «Moist surface» node





- Requires a «Moist air» node within «Moisture transport in Building Materials» physics
- Designed to model convective moisture transport
- Evaporation only if a liquid layer exists
- Condensation only for RH>1
- Evaporation rate factor K for diffusive moisture transport:

$$K = -D \cdot \frac{dc_v}{dx} \cdot \frac{1}{c_{sat}(T) - c_v} \rightarrow g_{evap} = -M_v \cdot D \cdot \frac{dc_v}{dx}, \quad \text{if} \quad c_v > c_{sat} \quad \text{or} \quad c_l > 0$$

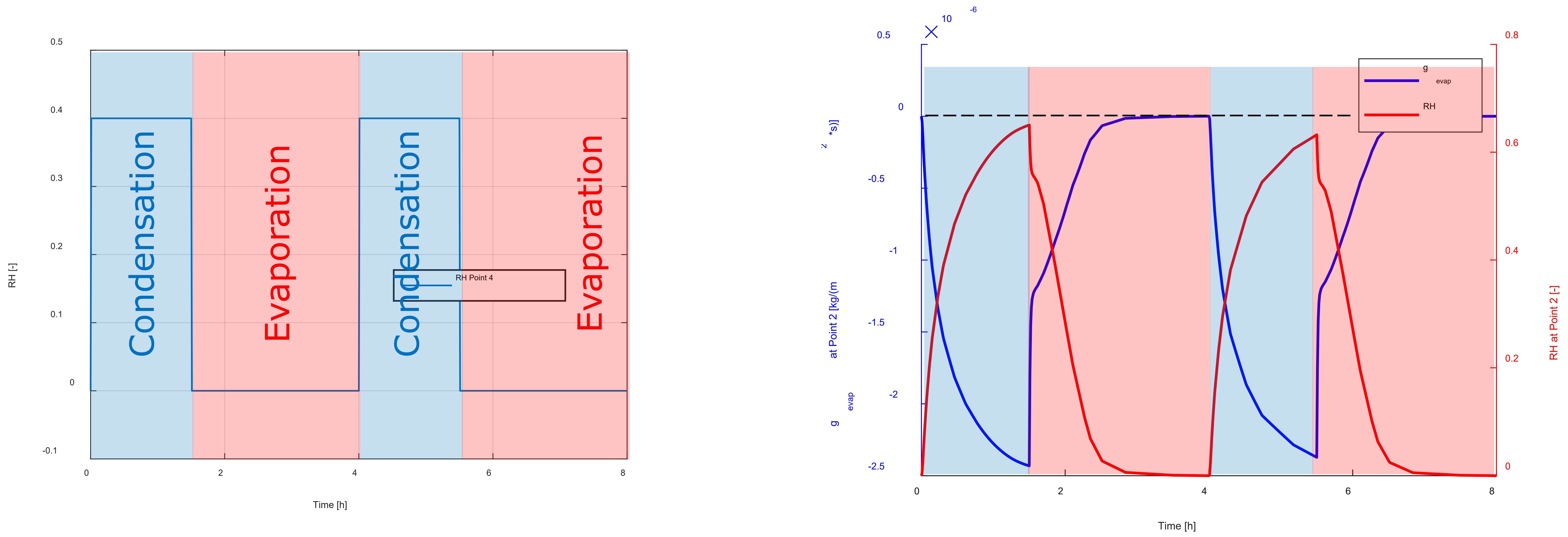


$$-\mathbf{n} \cdot \mathbf{g}_w = A_c g_{evap}$$

$$g_{evap} = \begin{cases} M_v K(c_{sat} - c_v) & \text{if } c_v > c_{sat} \text{ or } c_l > 0 \\ 0 & \text{otherwise} \end{cases}$$

$$M_v \frac{\partial c_l}{\partial t} = -A_c g_{evap}, \quad c_l(0) = c_{l,init}$$

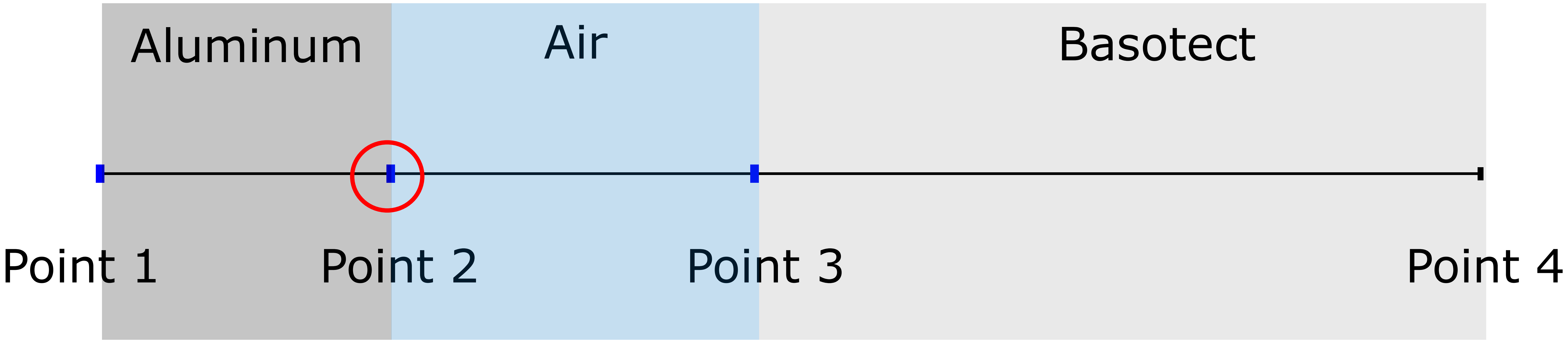
1. «Moist surface» node: Test with condensation/evaporation cycles



- Vapor flux $g_{evap} \neq 0$, despite $RH < 1$ and $c_{liq,ini} = 0$
- Vapor flux $g_{evap} < 0$ always, despite $RH < 1$
- $c_{liq} = 0$, despite $g_{evap} < 0$

Erroneous and inconsistent results?!

2. Custom model: «Film growth model»

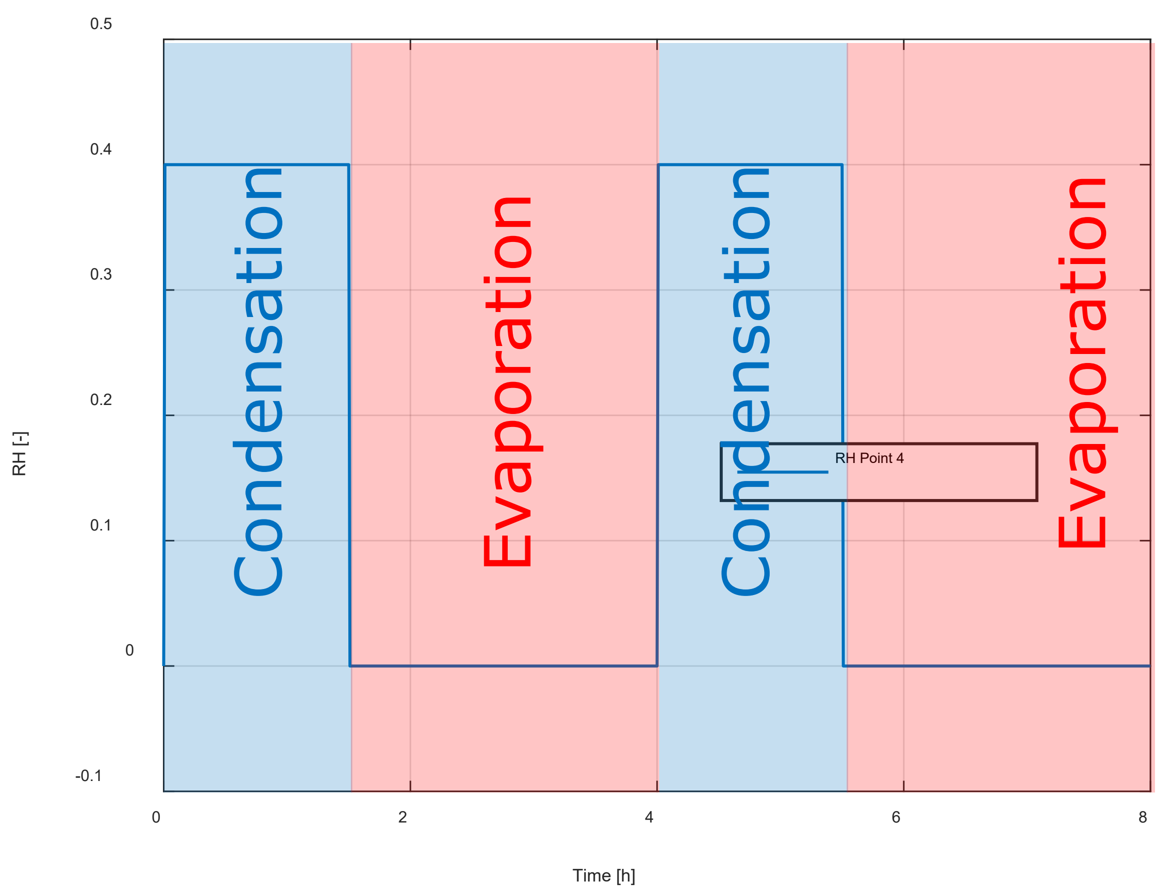


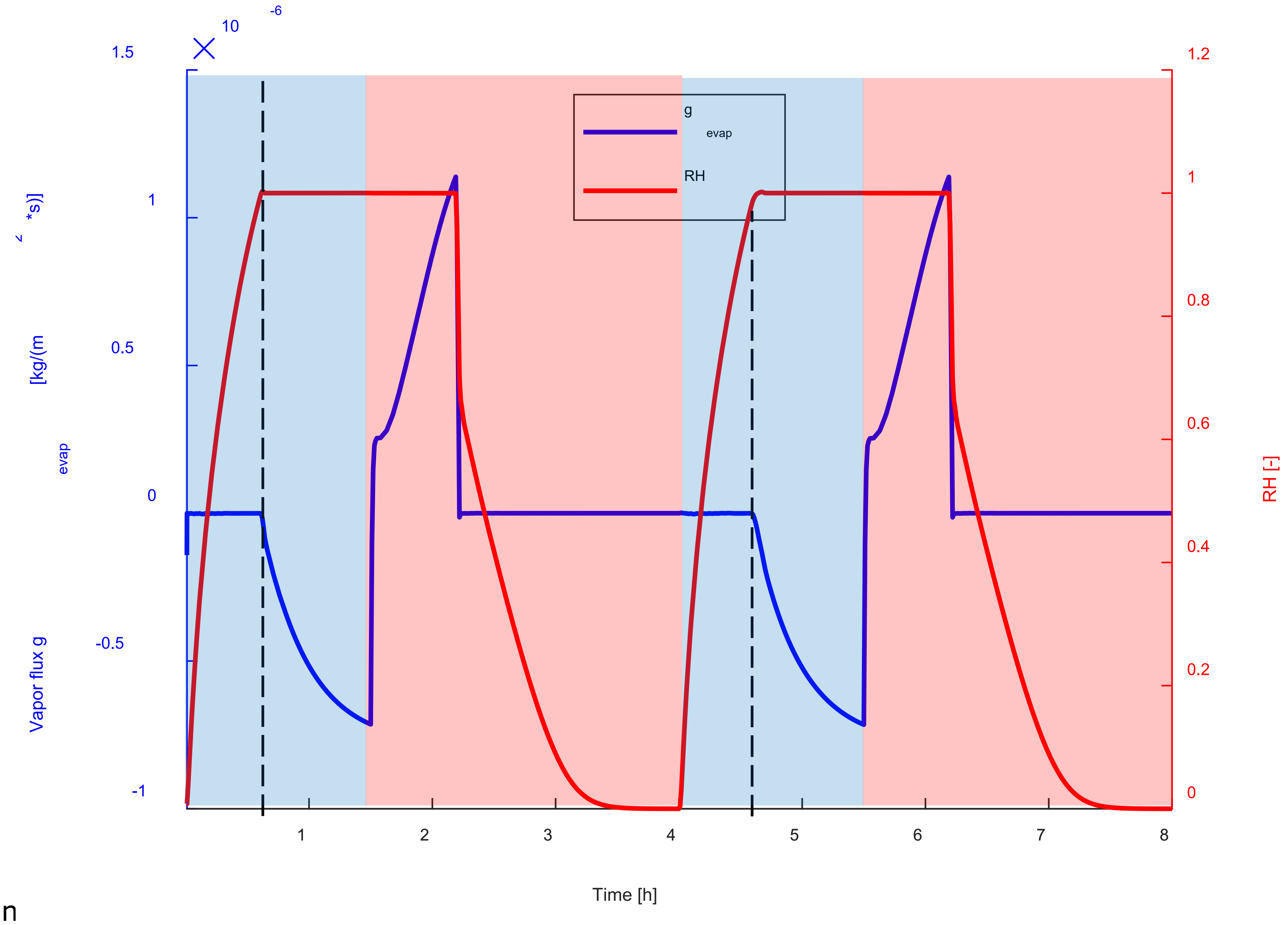
- Water condensation on cold wall boundary (Point 2)
- 2 weak inequality constraints at Point 2 to impose that:
  - $\phi \leq 1$
  - $\phi = 1$ if $d_{film} > d_0$ with $d_0 = 10^{-9} m$
- Model solves for dependent variable d_{film} defined by ODE:

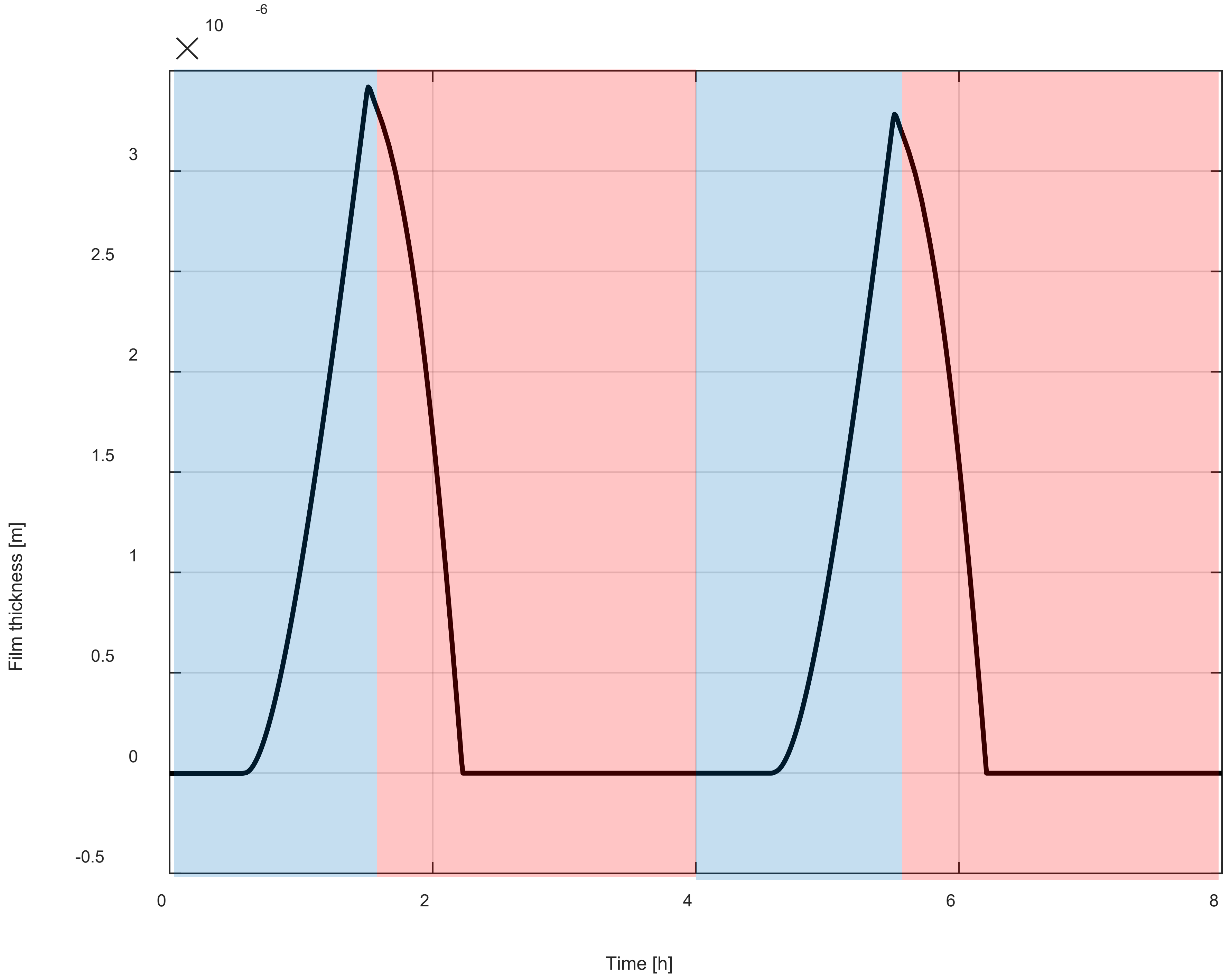
$$\frac{d\, d_{film}}{dt} = \Big(\theta(\phi - 1) + \theta\big(d_{film}\big) \cdot \big(1 - \theta(\phi - 1)\big)\Big) \cdot \frac{1}{\rho}\big(-\delta_p \vec{\nabla} p_v \cdot \vec{n}\big) \qquad \theta(x) = \begin{cases} 0, & x < 0 \\ 1, & x > 0 \end{cases}$$

2. Custom model: «Film growth model»

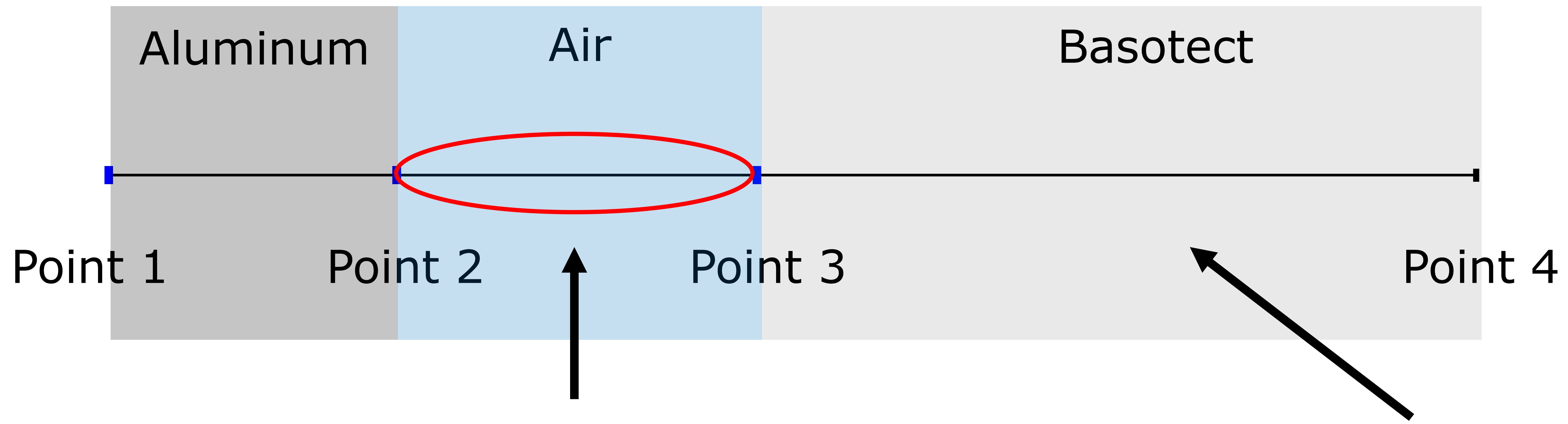
- Relative humidity at point 2 is limited to $RH = 1$, and consistent with the vapor flux.
- $RH = 1$ persists into the evaporating period due to the existence of a liquid film

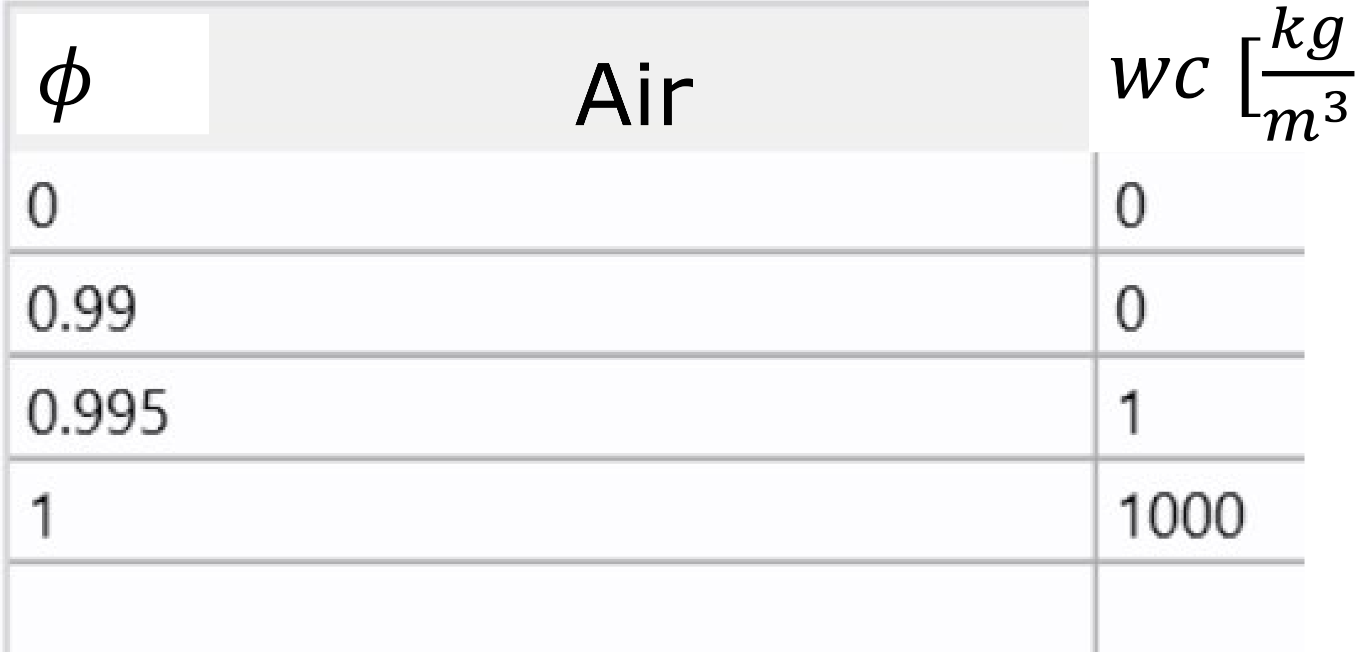






3. Custom built: «Sponge model»





ϕ Air	$wc\ [\frac{kg}{m^3}]$
0	0
0.99	0
0.995	1
1	1000

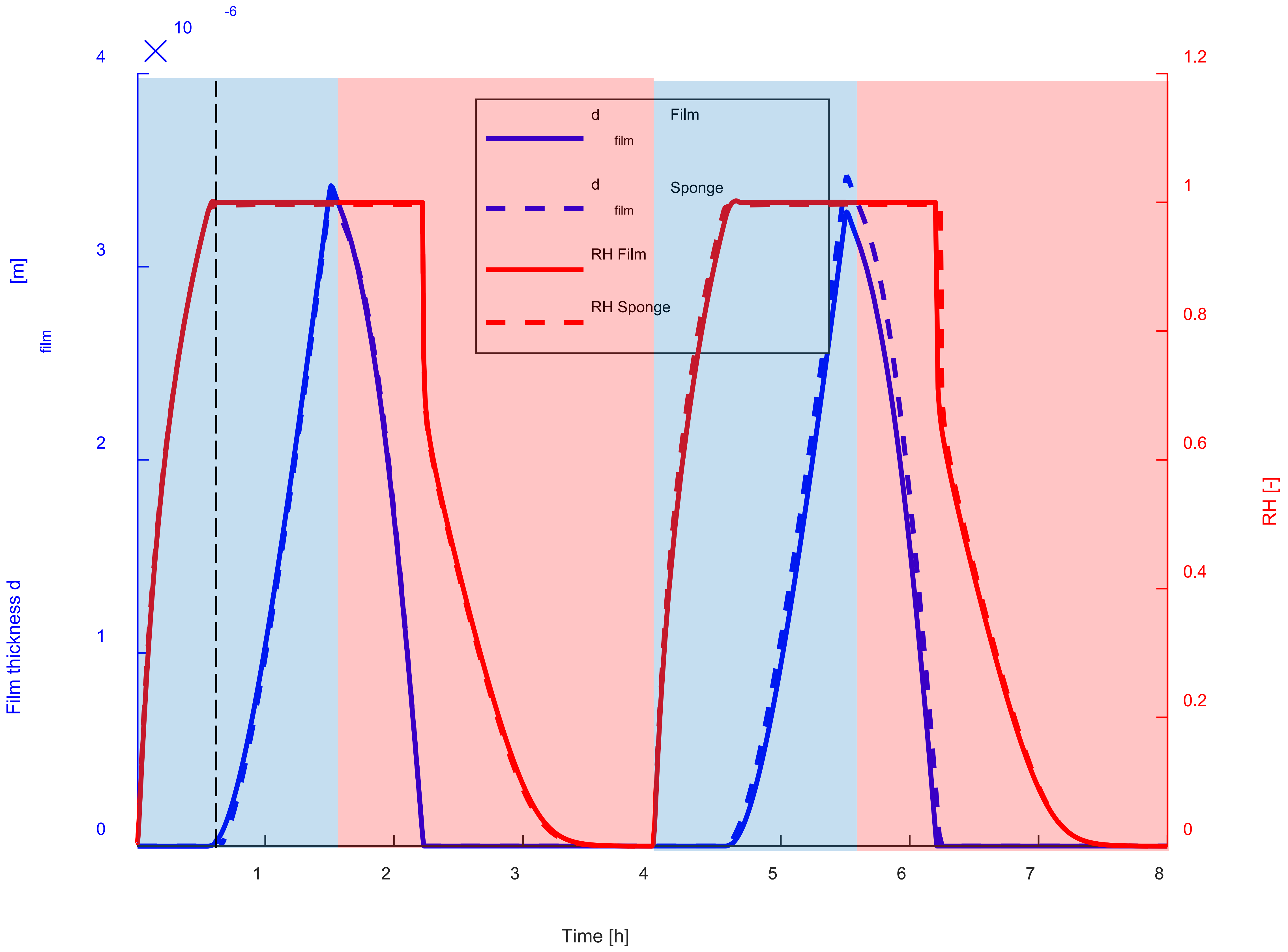
ϕ Basotect	$wc\ [\frac{kg}{m^3}]$
0	0
0.55	1
0.95	2.7
1	100

- Water condensation within the «Air» domain
- Air modeled as a porous building material with an **artificial moisture capacity function** $wc(\phi)$

- Water film thickness d calculated from moisture content $c_w\ [\frac{mol}{m^3}]$:

$$d = \frac{M_w}{\rho}\int_{Point\ 2}^{Point\ 3} c_w(x)\ dx$$

«Film growth model» vs. «Sponge model»



- Good agreement between «Film model» and «Sponge model».
- Both models are able to predict the amount of water condensed on a surface.

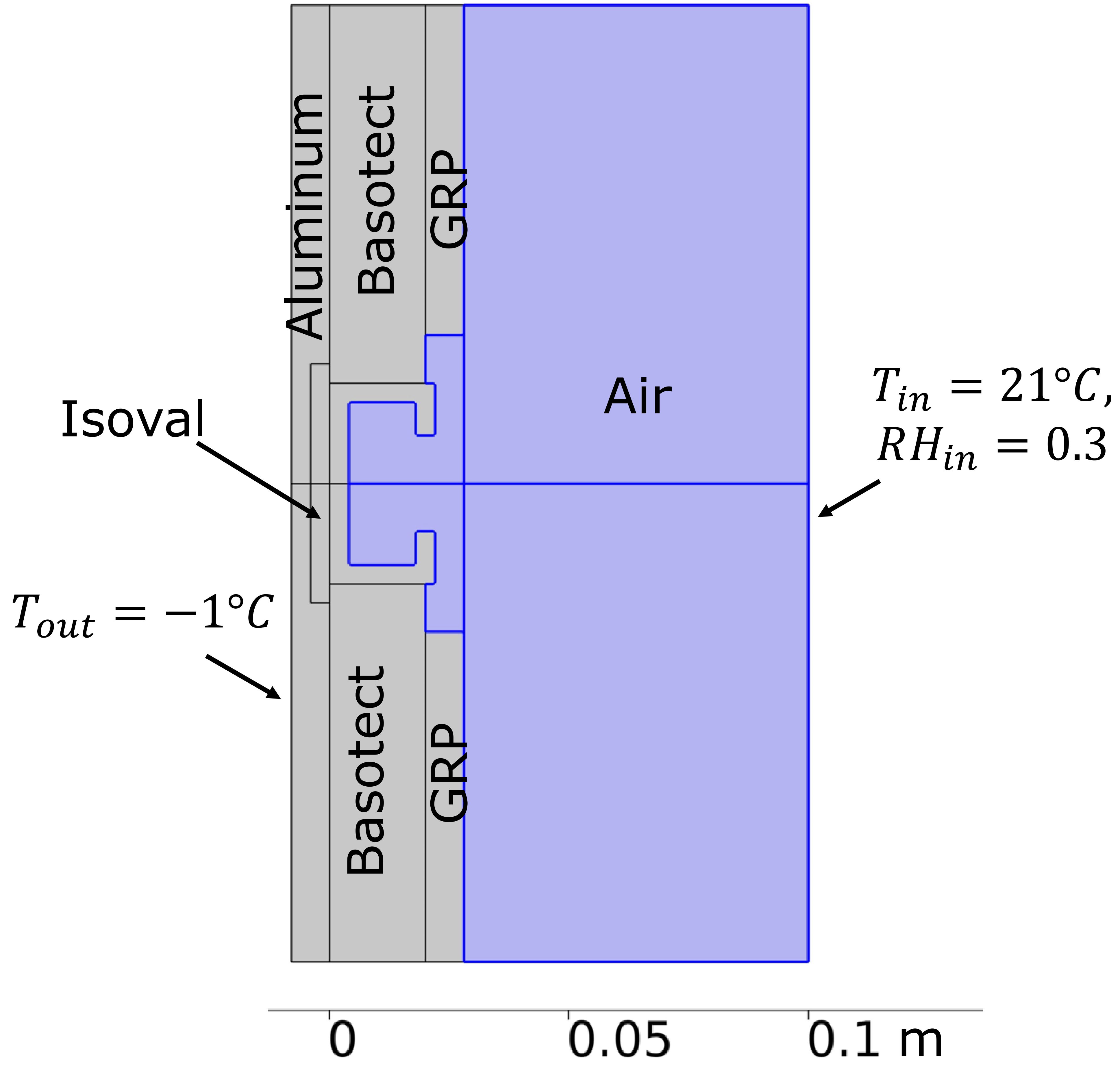
«Film model» vs «Sponge model»

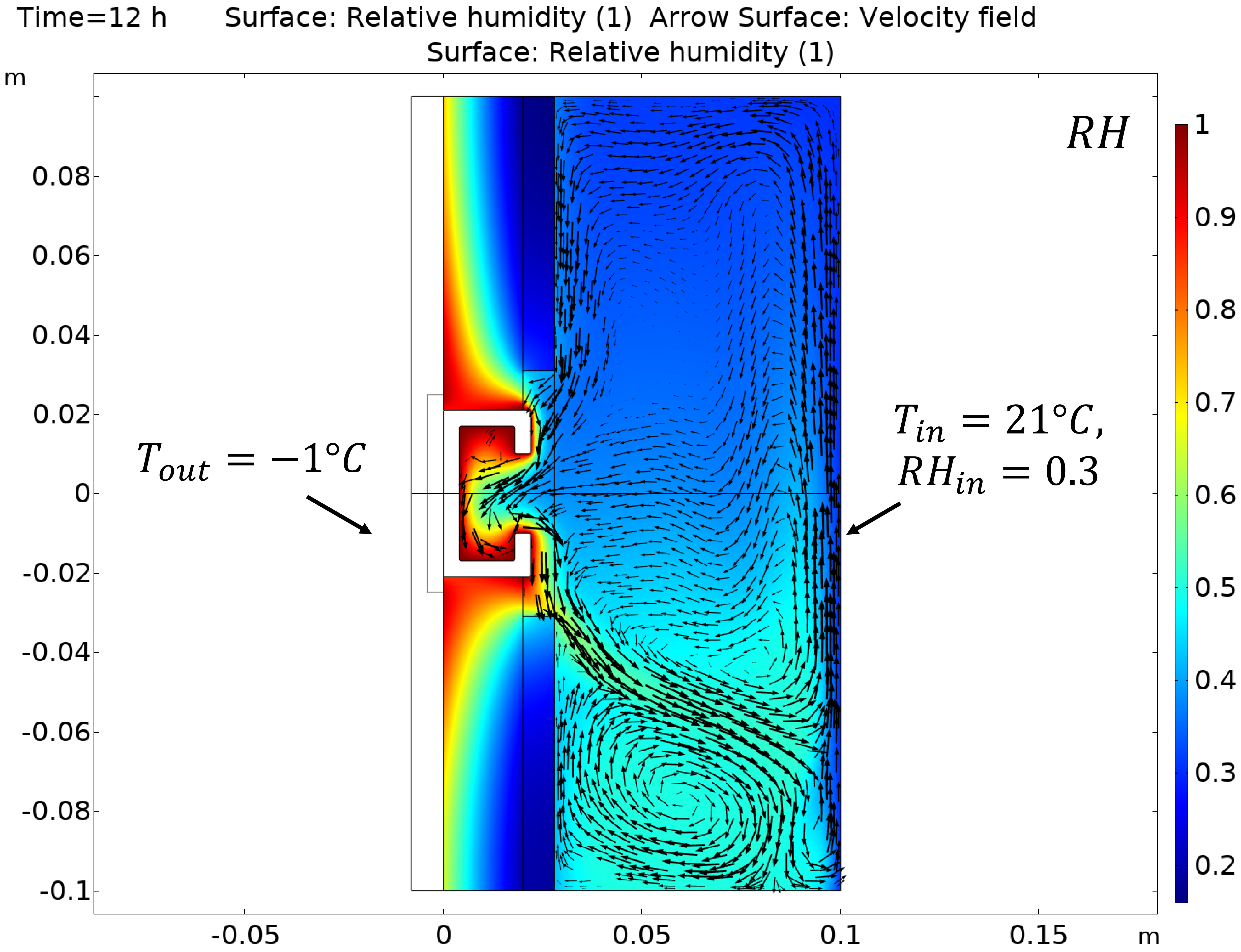
	2. Film growth model	**3. Sponge model**
Advantages	• Direct calculation of d_{film} • close representation of the physics • Possible to combine with transport by convection	• Easy to implement • No constraints necessary
Disadvantages	• Increased model complexity: +1 ODE, +2 constraints • Evaporation requires very small timesteps in 2D • Neglects the influence of condensed moisture on conductive heat transport	• Not possible to combine with transport by convection • d_{film} difficult to calculate in 2D • Sensitivity to the arbitrary definition of $wc(\phi)$ in 2D • Neglects the influence of condensed moisture on conductive heat transport

➡ Film growth model has been selected to calculate condensation on thermal bridges in 2D and 3D.

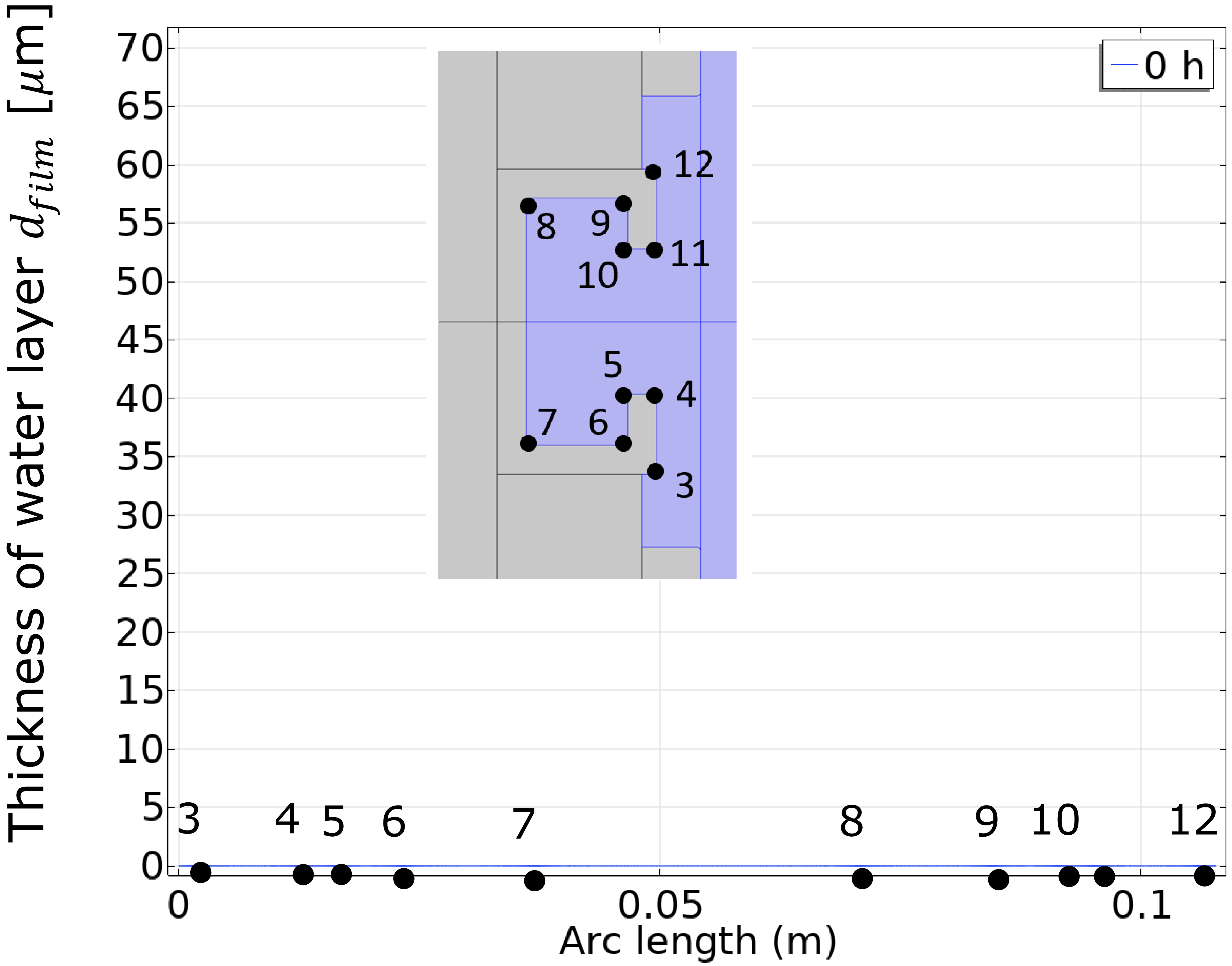
2D Sim. of condensation on a thermal bridge in train walls using «Film growth model»

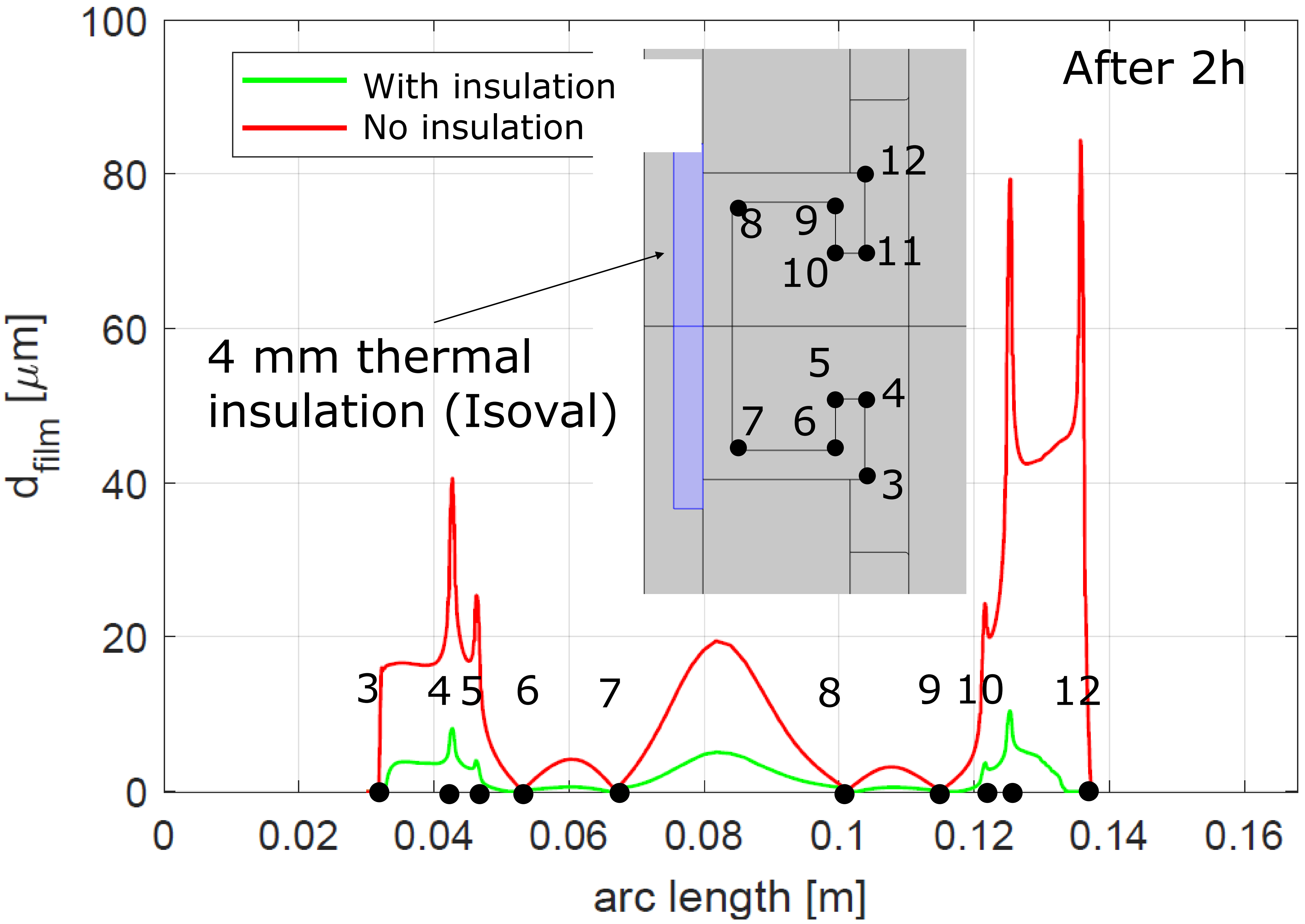
Winter scenario with constant boundary conditions: $T_{out} = -1°C$, $T_{in} = 21°C, RH_{in} = 30\%$





2D Sim. of condensation on a thermal bridge in train walls using «Film growth model»





4 mm thermal insulation (Isoval)

After 2h

- Convection dominates water vapor transport over diffusion.
- Droplets running from the surface of a thermal bridge have to be expected after ~10 h in a winter scenario.
- Thermal insulation of the thermal bridge substantially reduces the amount of condensing water.
- Condensation is most pronounced at protruding edges and corners.

Conclusions

- The COMSOL preset «Moist surface» node has not yielded plausible results for the growth and evaporation of a condensate layer.
- Two different custom approaches («Film growth model» and «Sponge model») have been successfully tested in 1D with nearly identical results for the resulting condensate film thickness.
- The «Film model» has been coupled to CFD to include convective moisture transport in 2D simulations of condensation on a thermal bridge in train walls:
    - Convection dominates water vapor transport over diffusion.
    - Droplets running from the surface of a thermal bridge have to be expected after ~10 h in a winter scenario.
    - Thermal insulation of the thermal bridge substantially reduces the amount of condensing water.
- In 2D, the «Film model» is computationally very expensive in cases of evaporation, due to rapid changes in the ODE when $d_{film} \rightarrow 0$.

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Thank you!

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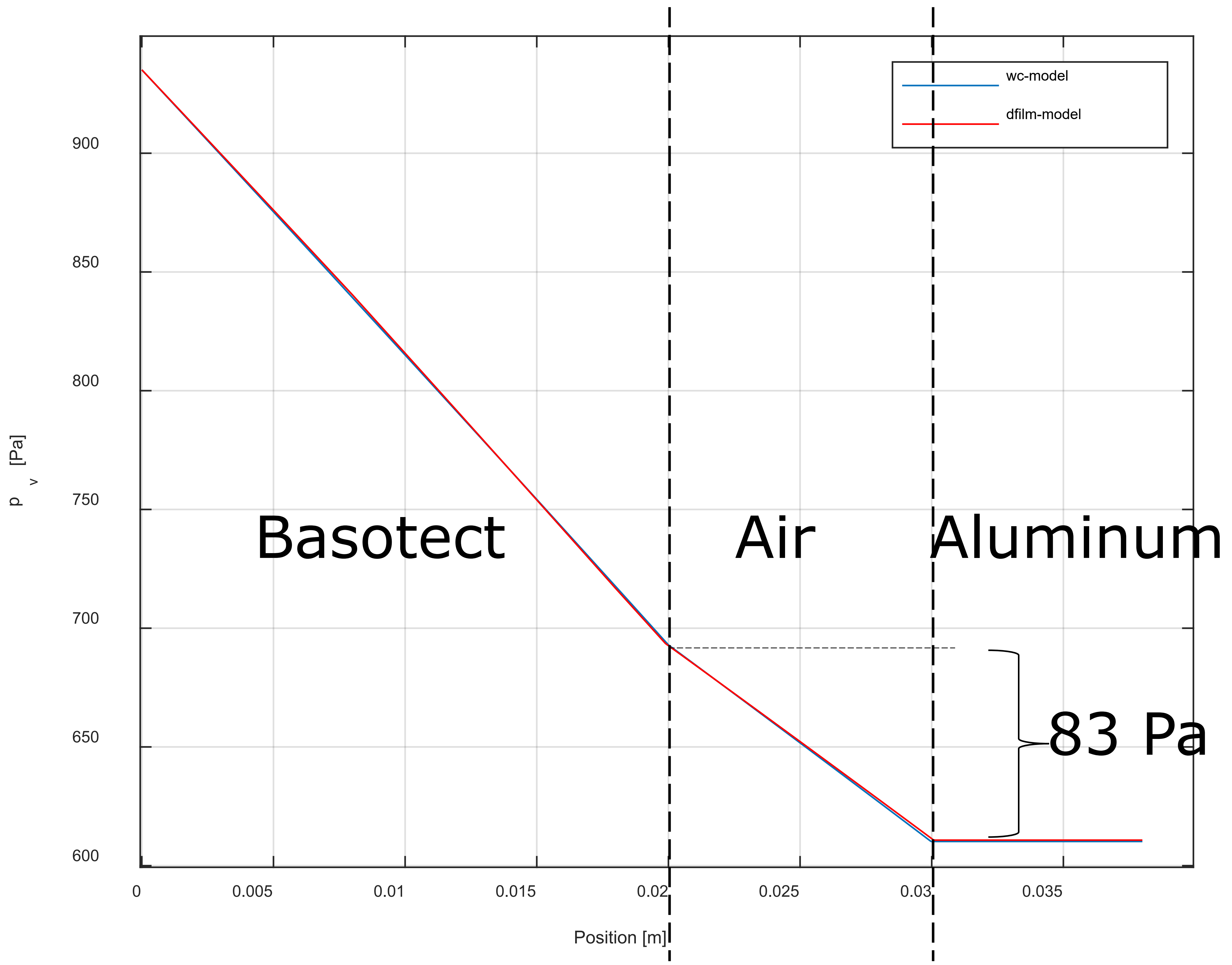
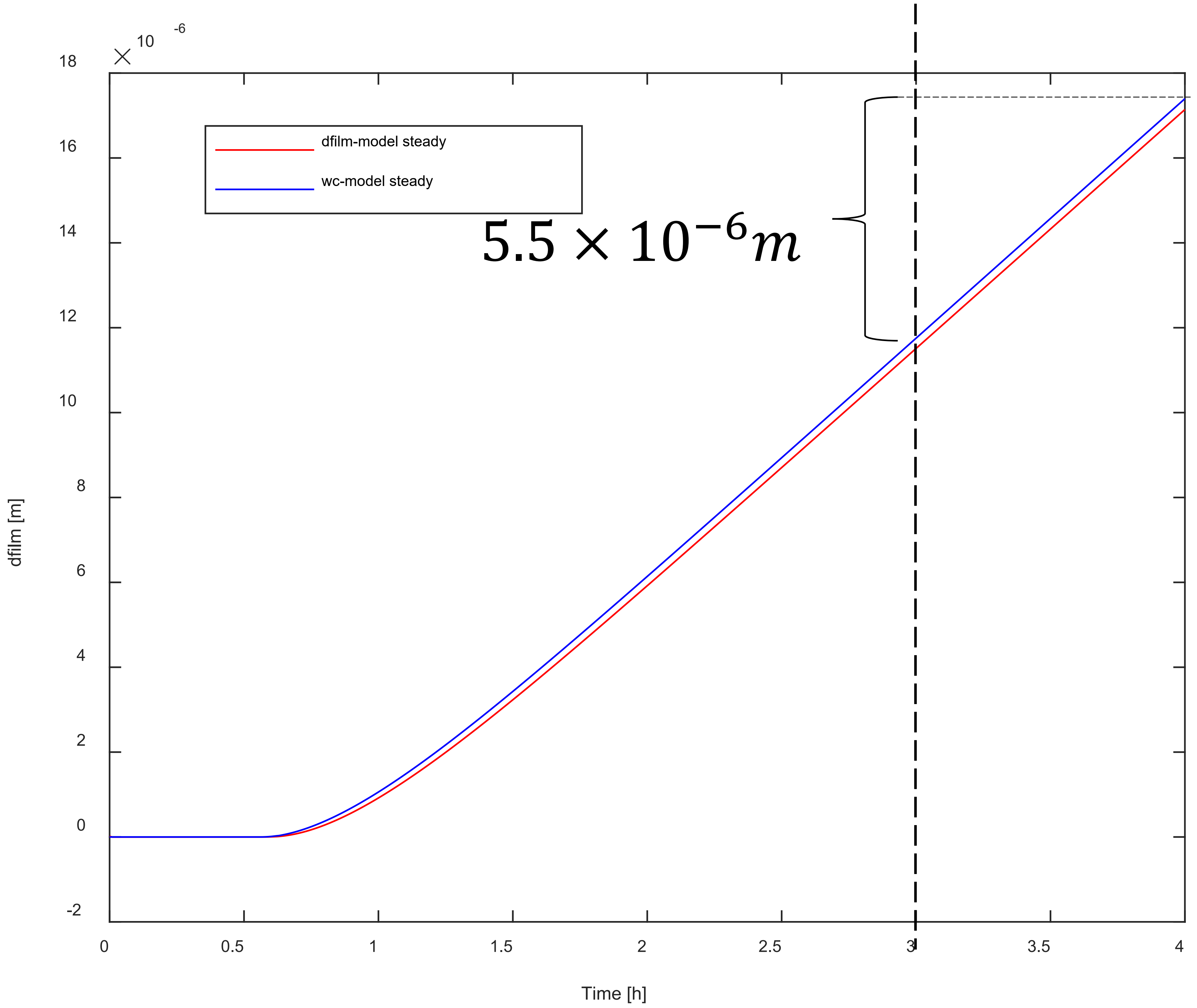
Annex

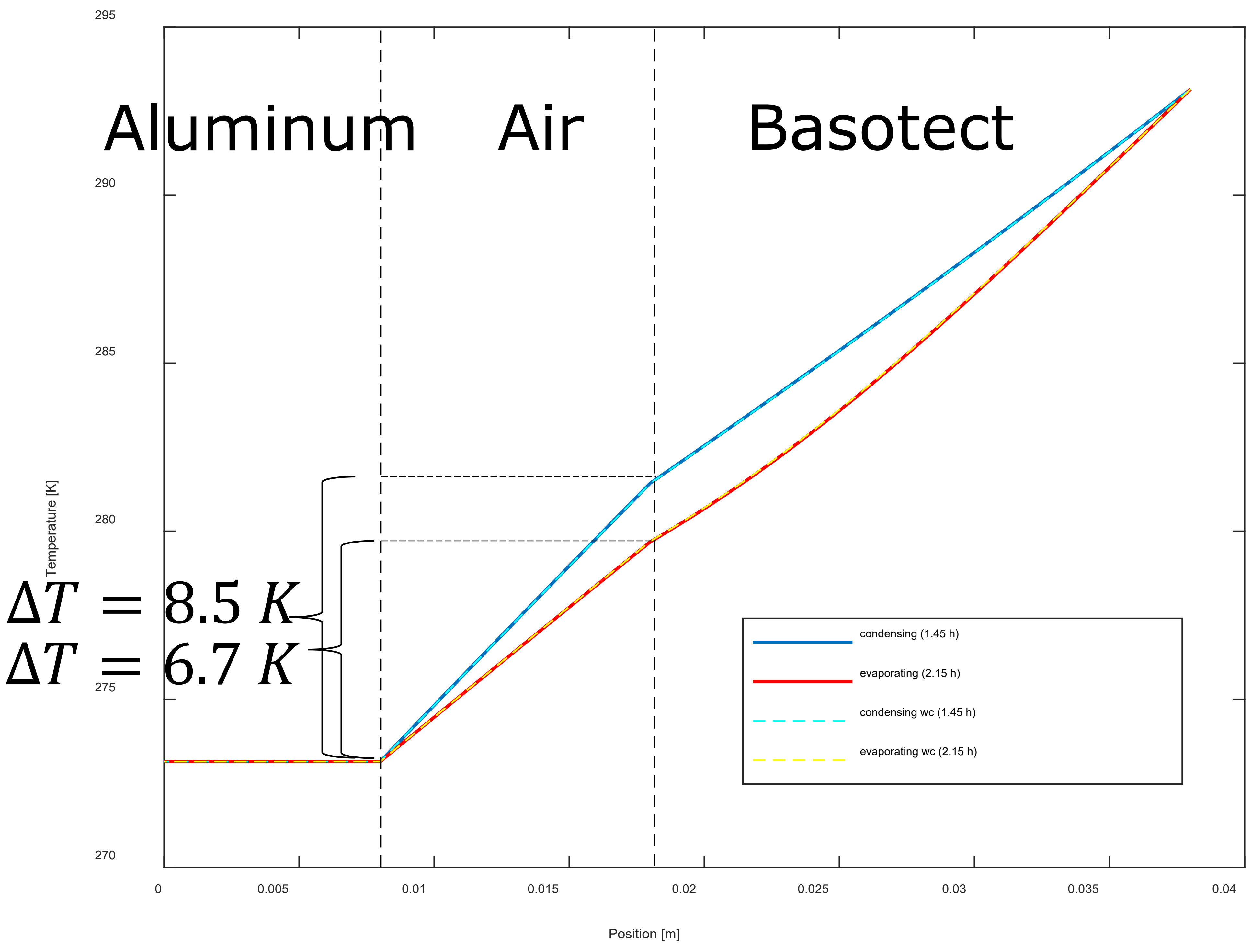


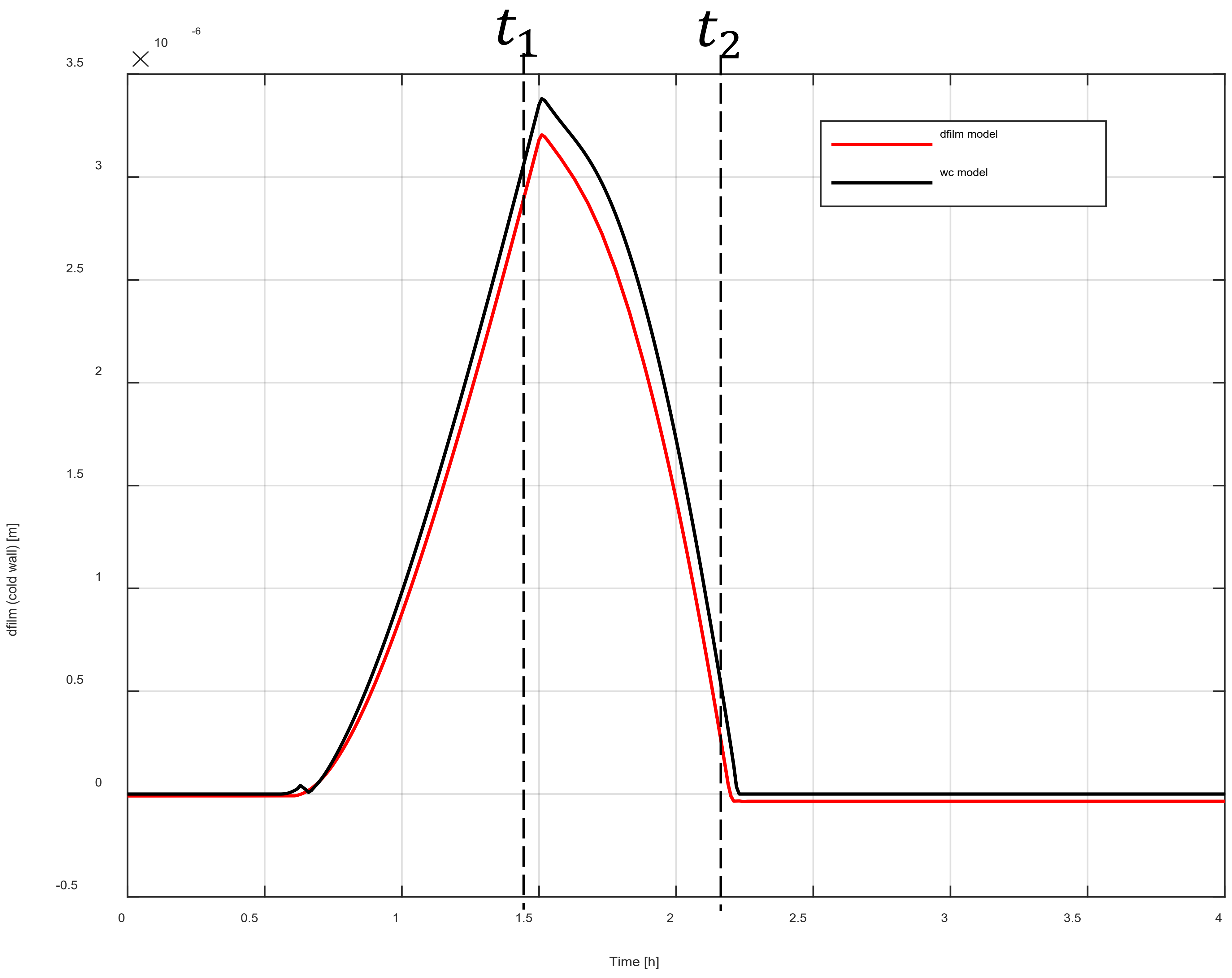


Validation of the growth rate of the water film for **steady** 40% RH on warm side (Point 4):

- Calculated from water vapor pressure gradient: $\frac{d}{dt}d_{film} = -\frac{\delta_p \cdot \frac{d}{dx}mt.pv}{\rho} = -\frac{1.9\times10^{-10}s\cdot\left(-8300\frac{Pa}{m}\right)}{1000\frac{kg}{m^3}} = 1.57\times10^{-9}\frac{m}{s}$

  where $\delta_p = 2.01\times10^{-7}s\cdot T^{0.81}\cdot\frac{1}{p_{atm}} = 1.89\times10^{-10}s$ … $1.93\times10^{-10}s$ for T between $273\,K$ … $281\,K$
- Calculated from line integral of total water content (wc-model) or dependent variable directly (dfilm-growth-model): $\frac{d}{dt}d_{film} = \frac{5.5\times10^{-6}m}{3600\,s} = 1.53\times10^{-9}\frac{m}{s}$





Consistency of heat flux at cold wall (Point 2) **for on-off** 40% RH on warm side (Point 4):

Condensing ($t_1 = 1.45\,h$):

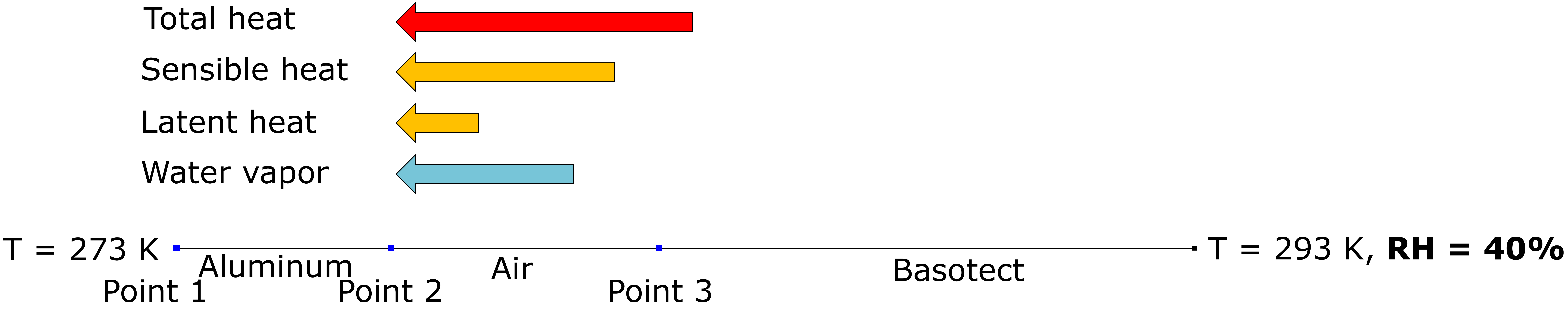
- $\nabla T = 850\,\frac{K}{m} \;\rightarrow\; q_{sens} = -k \cdot \nabla T = -19.6\,\frac{W}{m^2}$
- Total heat flux reported at point 2: $q_{tot} = -23.8\,\frac{W}{m^2}$
- Latent heat flux: $q_{lat} = q_{tot} - q_{sens} = -4.2\,\frac{W}{m^2}$, in negative x-direction, released at Point 2 (as vapor cannot penetrate Aluminum)
- Latent heat release expected from growth of water film: $q_{lat,exp} = \frac{d}{dt} d_{film} \cdot \rho_w \cdot L_v = 3.5\,\frac{W}{m^2}$

Evaporating: ($t_2 = 2.15\,h$):

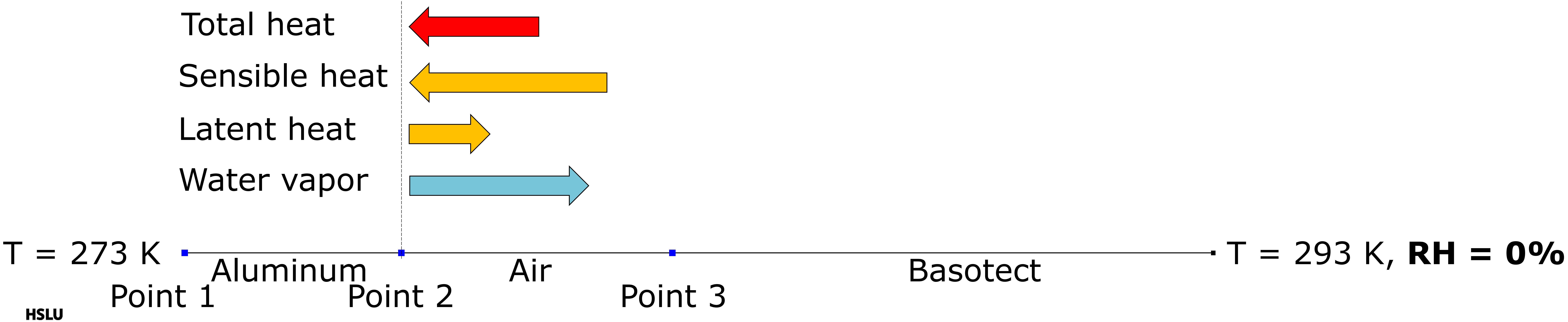
- $\nabla T = 670\,\frac{K}{m} \rightarrow\ q_{sens} = -k \cdot \nabla T = -15.4\,\frac{W}{m^2}$
- Total heat flux reported at point 2: $q_{tot} = -10.5\,\frac{W}{m^2}$
- Latent heat flux: $q_{lat} = q_{tot} - q_{sens} = +4.9\,\frac{W}{m^2}$ in positive x-direction (away from cold wall)
- Latent heat release expected from growth of water film: $q_{lat,exp} = \frac{d}{dt} d_{film} \cdot \rho_w \cdot L_v = 5.5\,\frac{W}{m^2}$

Point 2	**Condensing ($t_1 = 1.45\,h$)**	**Evaporating ($t_1 = 2.15\,h$)**
∇T air	$850\,\frac{K}{m}$	$670\,\frac{K}{m}$
$q_{sens} = -k \cdot \nabla T$	$-19.6\,\frac{W}{m^2}$	$-15.4\,\frac{W}{m^2}$
q_{tot} reported	$-23.8\,\frac{W}{m^2}$	$-10.5\,\frac{W}{m^2}$
$-q_{lat} = -(q_{tot} - q_{sens})$	$4.2\,\frac{W}{m^2}$	$-4.9\,\frac{W}{m^2}$
$q_{lat,exp} = \frac{d}{dt} d_{film} \cdot \rho_w \cdot L_v$	$3.5\,\frac{W}{m^2}$	$-5.5\,\frac{W}{m^2}$

Condensing $(t_1 = 1.45\ h)$

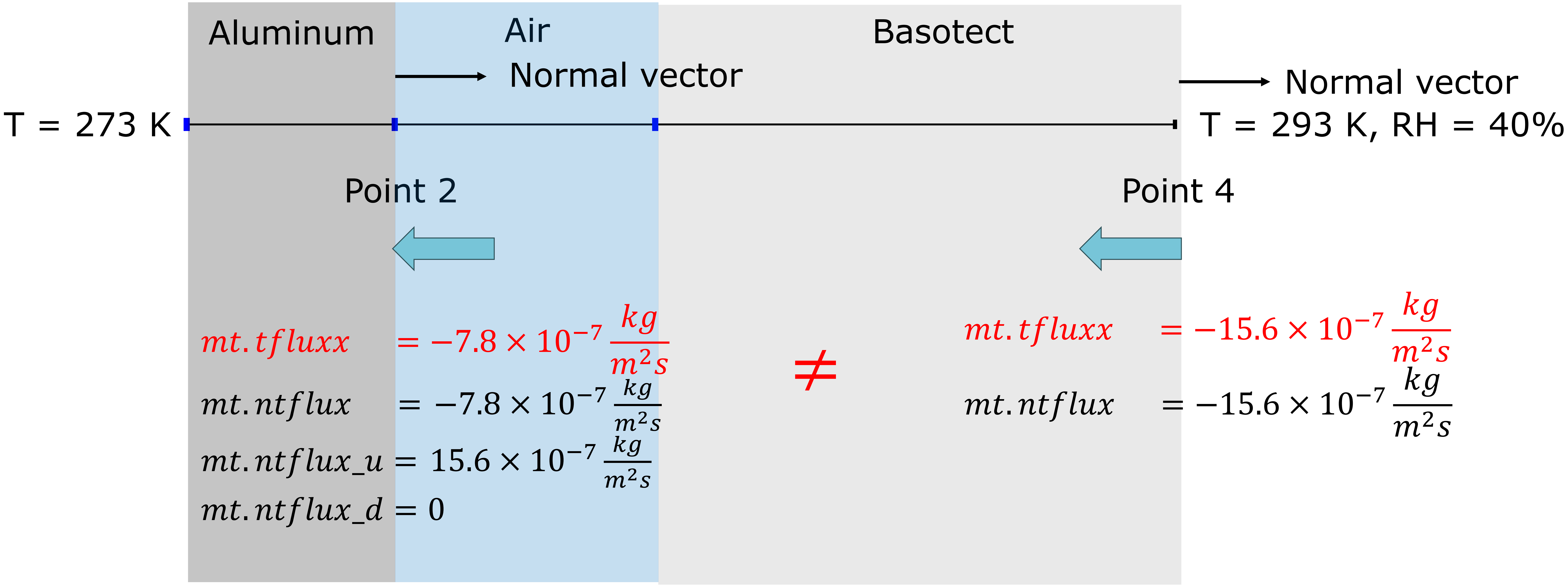


Evaporating $(t_2 = 2.15\ h)$



2. Film growth model: Using the correct vapor flux quantity

Comparison between vapor flux quantities for steady-state condensation, after 4h:



Aluminum

Air

Basotect

Normal vector

Normal vector

T = 273 K

T = 293 K, RH = 40%

Point 2

$$mt.tfluxx = -7.8 \times 10^{-7}\,\frac{kg}{m^2 s}$$

$$mt.ntflux = -7.8 \times 10^{-7}\,\frac{kg}{m^2 s}$$

$$mt.ntflux\_u = 15.6 \times 10^{-7}\,\frac{kg}{m^2 s}$$

$$mt.ntflux\_d = 0$$

$$\neq$$

Point 4

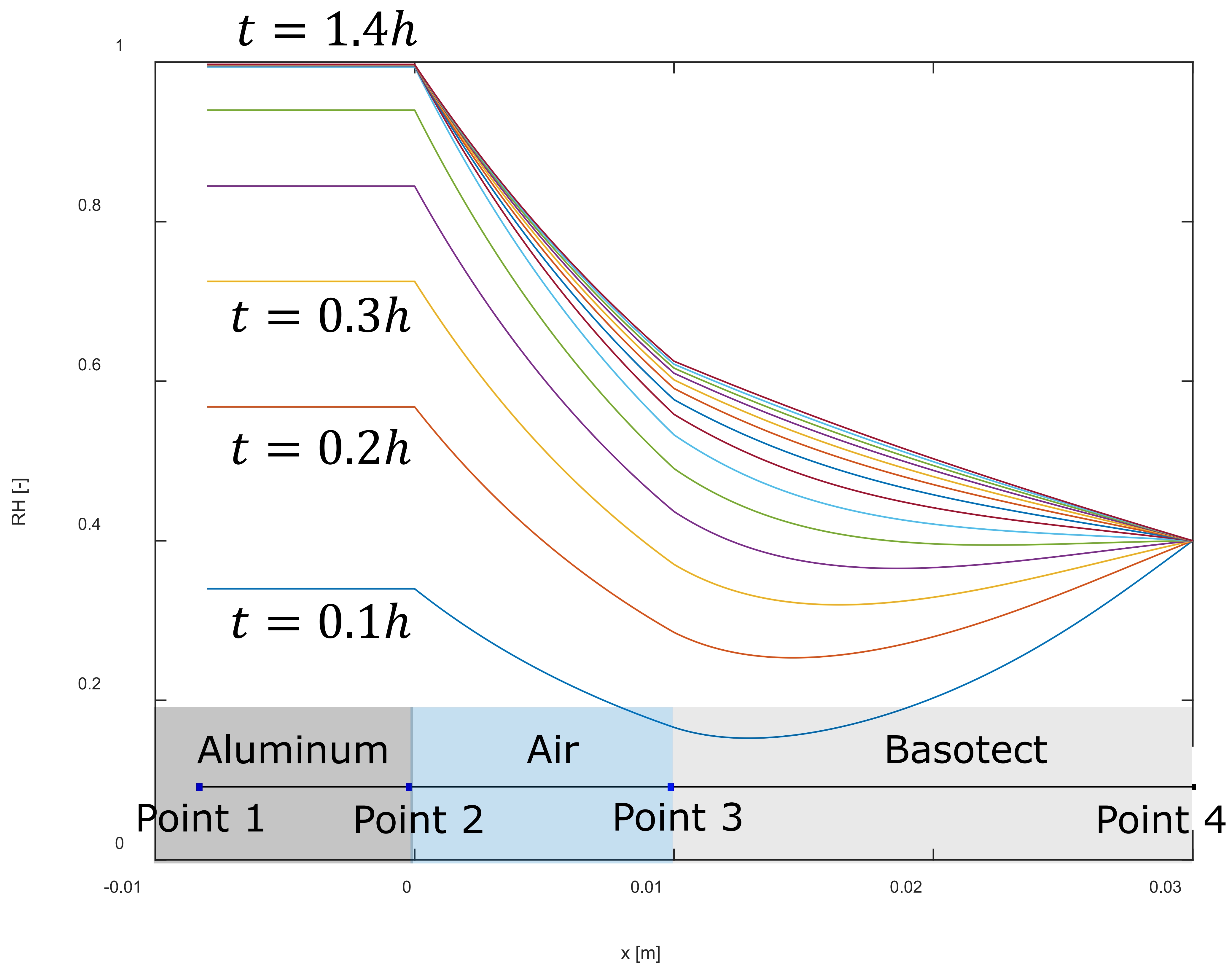
$$mt.tfluxx = -15.6 \times 10^{-7}\,\frac{kg}{m^2 s}$$

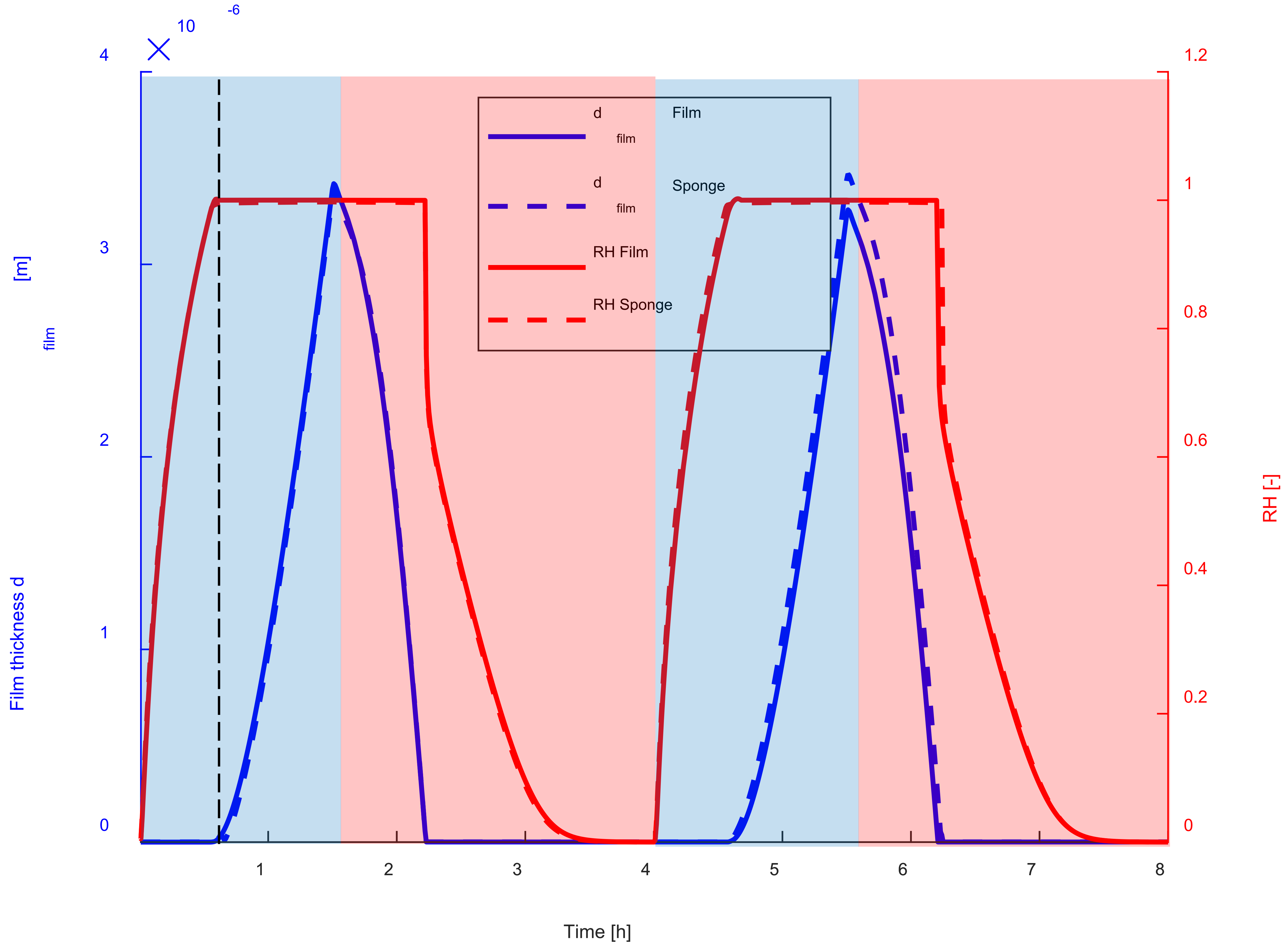
$$mt.ntflux = -15.6 \times 10^{-7}\,\frac{kg}{m^2 s}$$

$$-\delta_p \cdot \frac{d}{dx} mt.pv = -15.7 \times 10^{-7}\,\frac{kg}{m^2 s} \quad \Rightarrow$$

Use $2 \cdot mt.tfluxx$ to calculate the condensing mass??

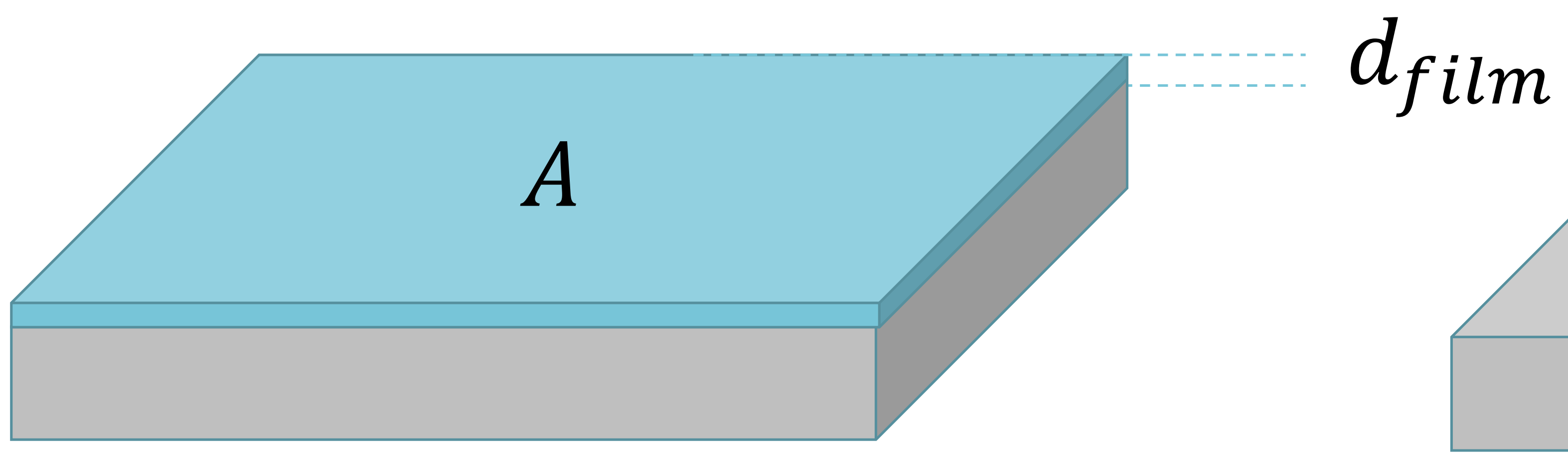
3. Custom built: «Sponge model»

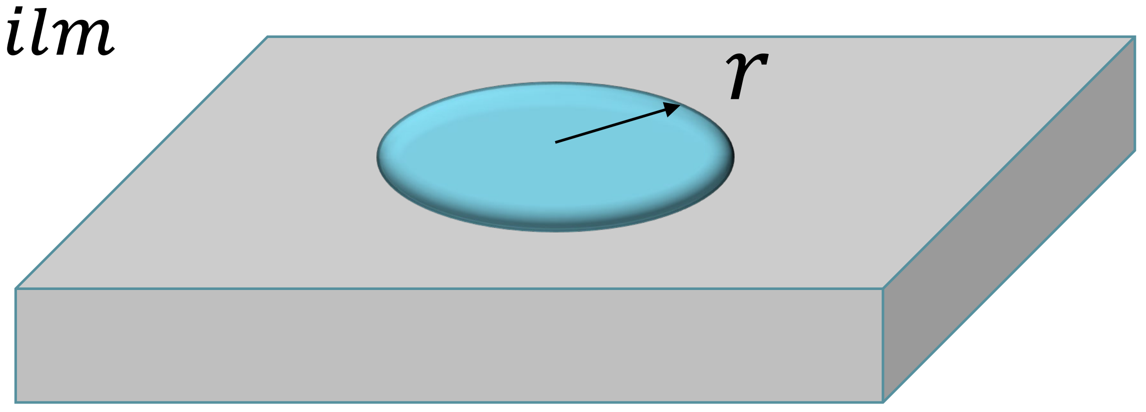




- Good agreement between «Film model» and «Sponge model».
- Both models are able to predict the amount of water condensed on a surface.

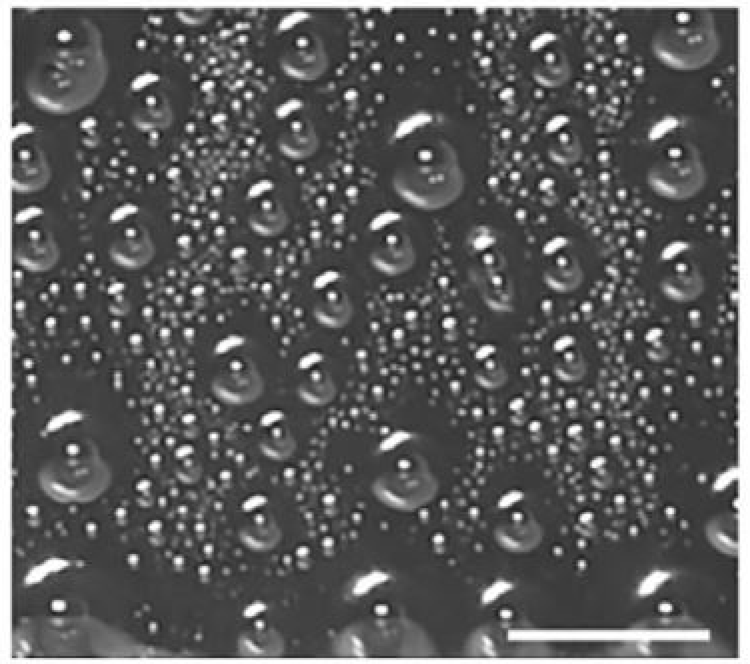
Will running drops form?

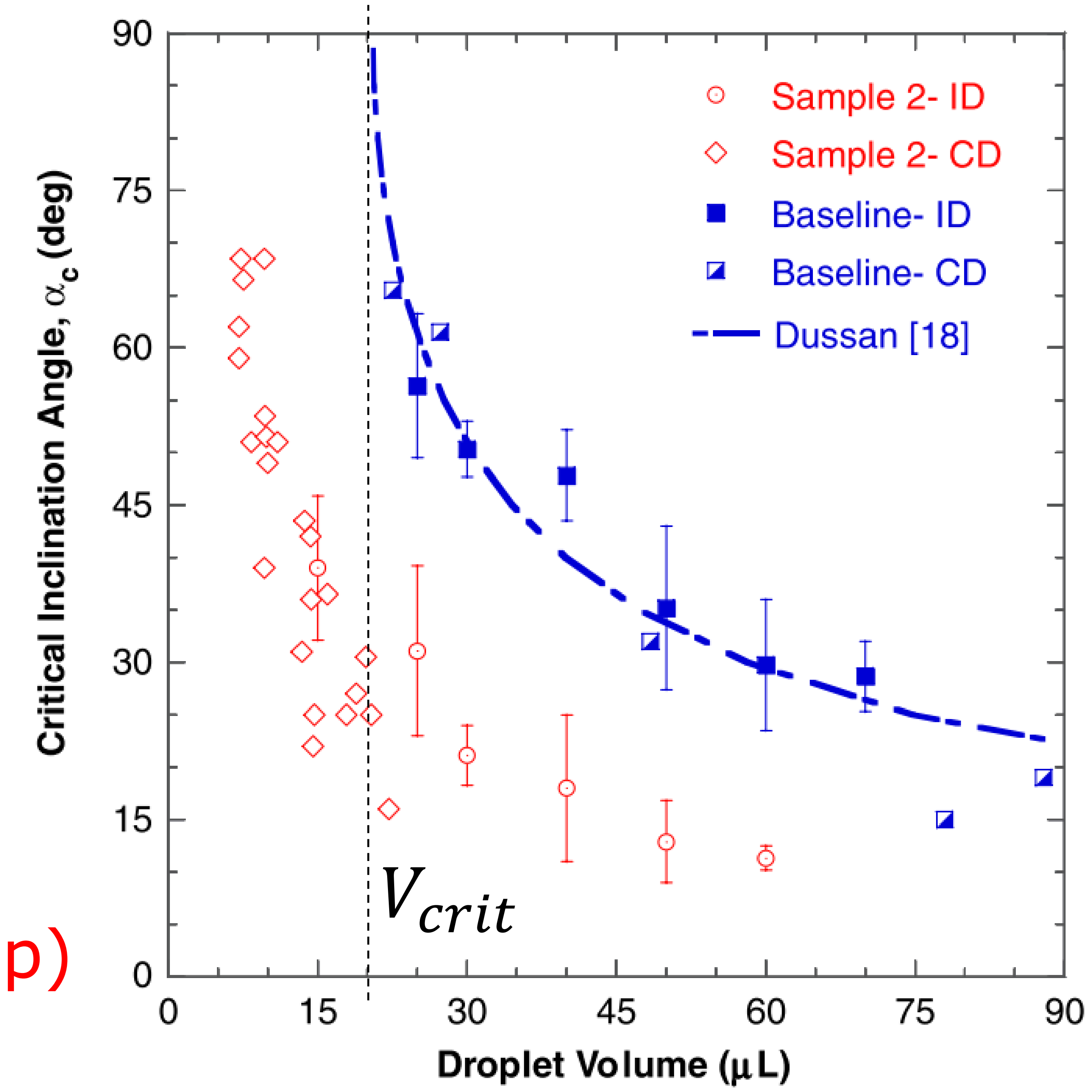




Integrate d_{film} over surface area $A \approx 1\, cm^2$ → Water volume V

- Concentrate V in one hemispherical drop
- Compare V to critical droplet volume V_{crit} for 90° inclination angle (vertical wall)
- $V \geq V_{crit}$: Running droplet
- $V < V_{crit}$: Sessile droplet





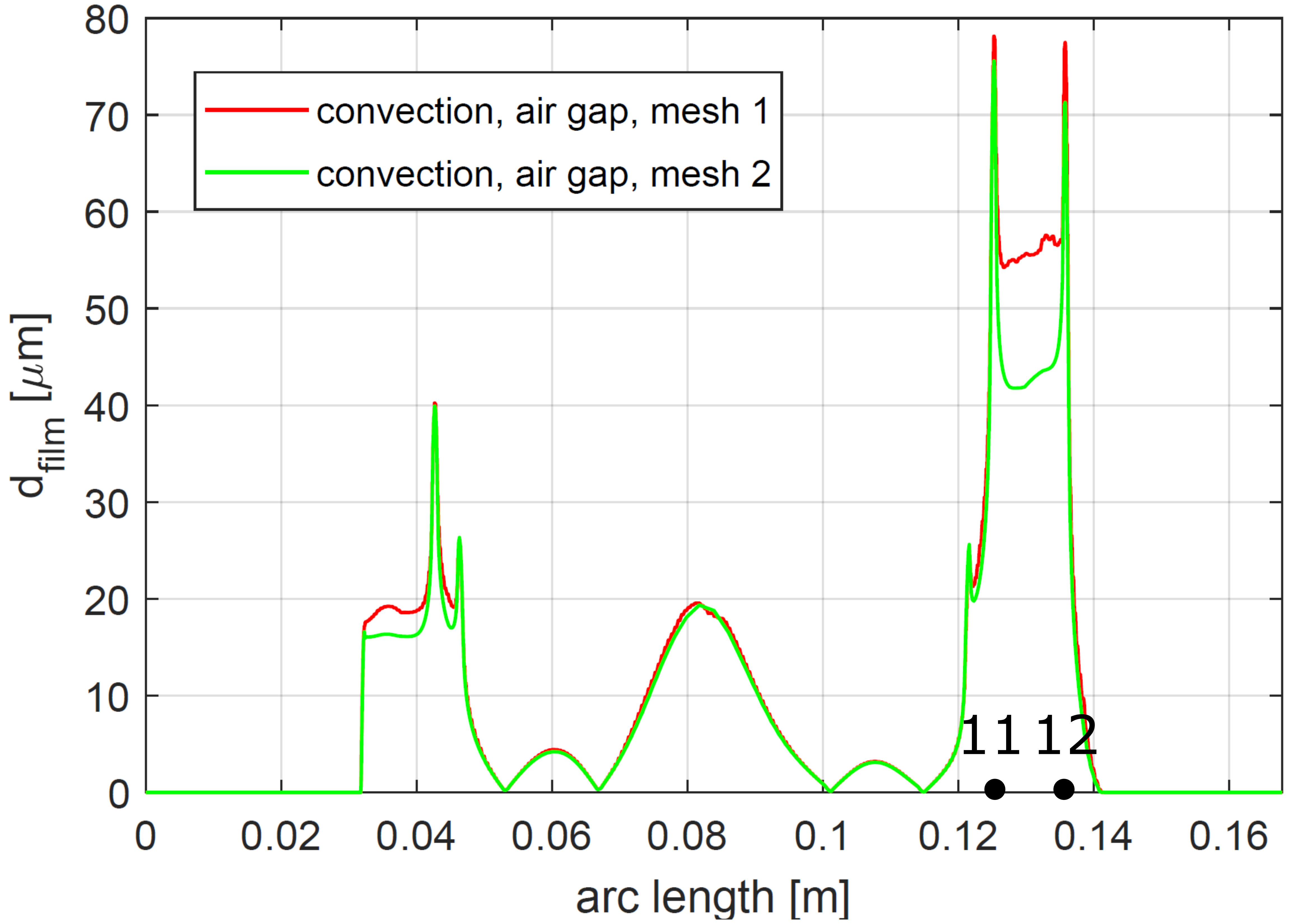
Critical inclination angle of water droplets on aluminum surface. Sommers et al., 2006

→ Running droplets predicted earlier than in reality (Concentration in 1 drop)

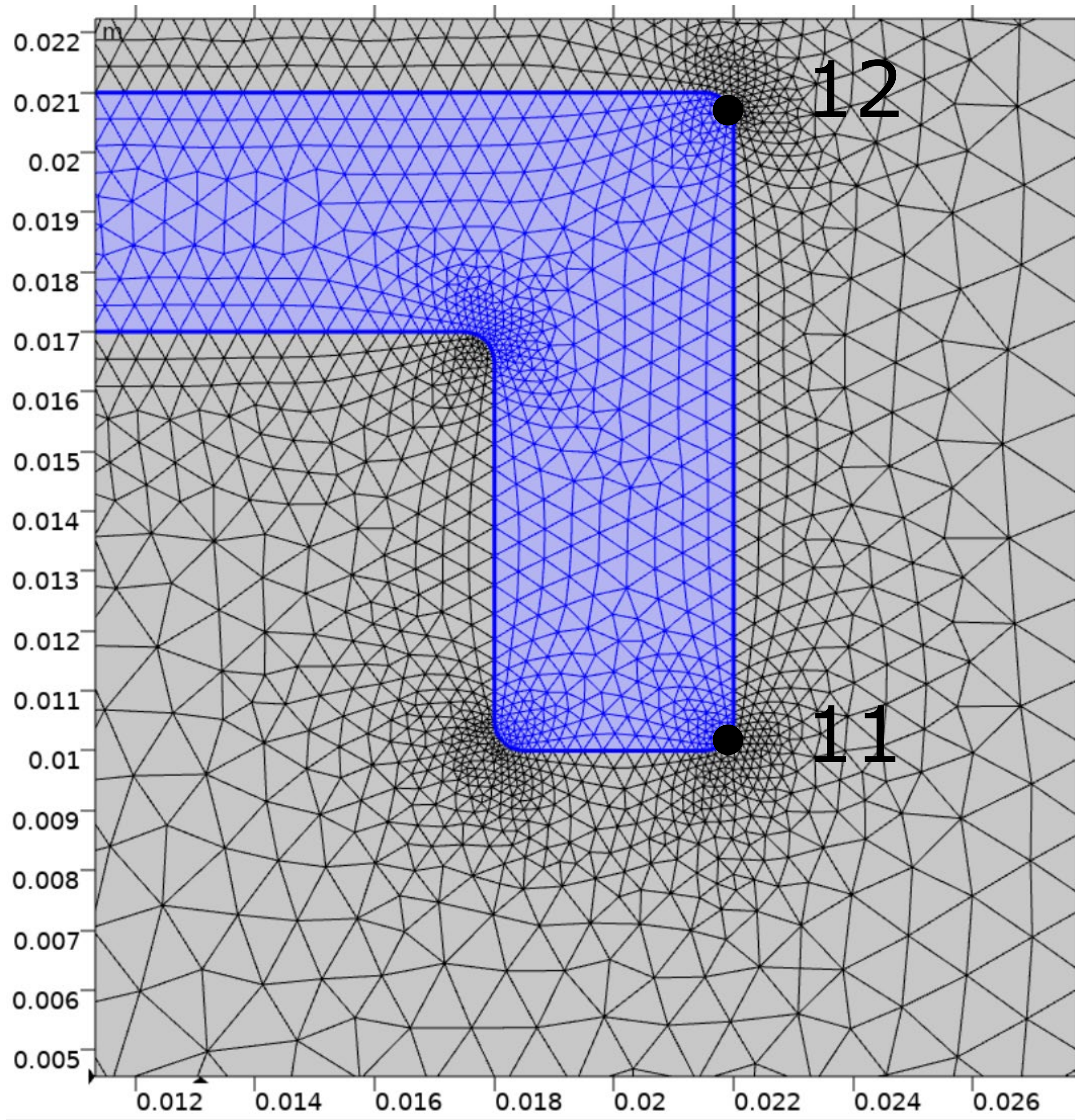
→ But: Vibrations may reduce V_{crit}

→ No precise predictions are possible about the time when droplets start running off.

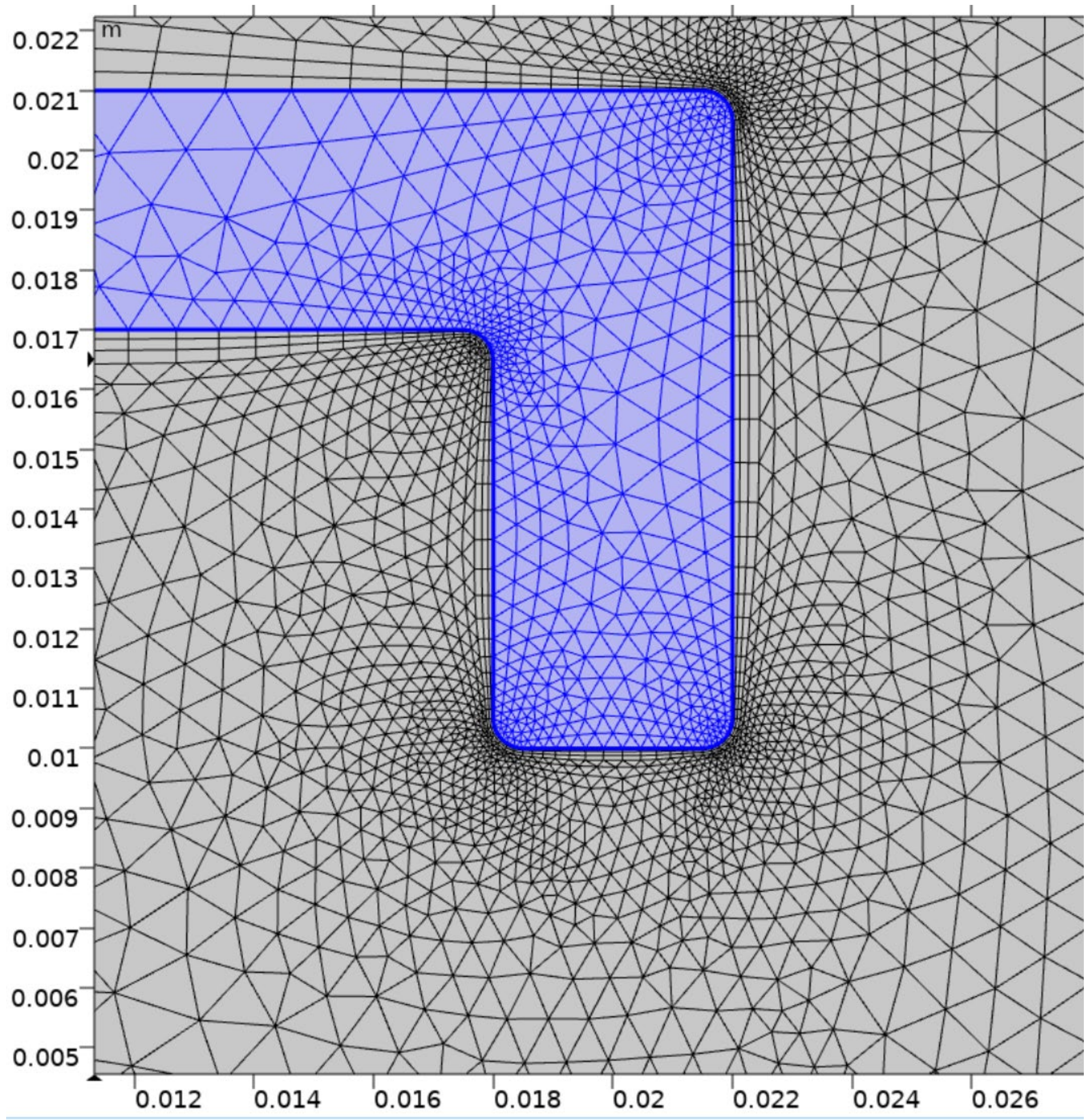
Annex: Mesh sensitivity of dfilm for Geometry 2 (2D)



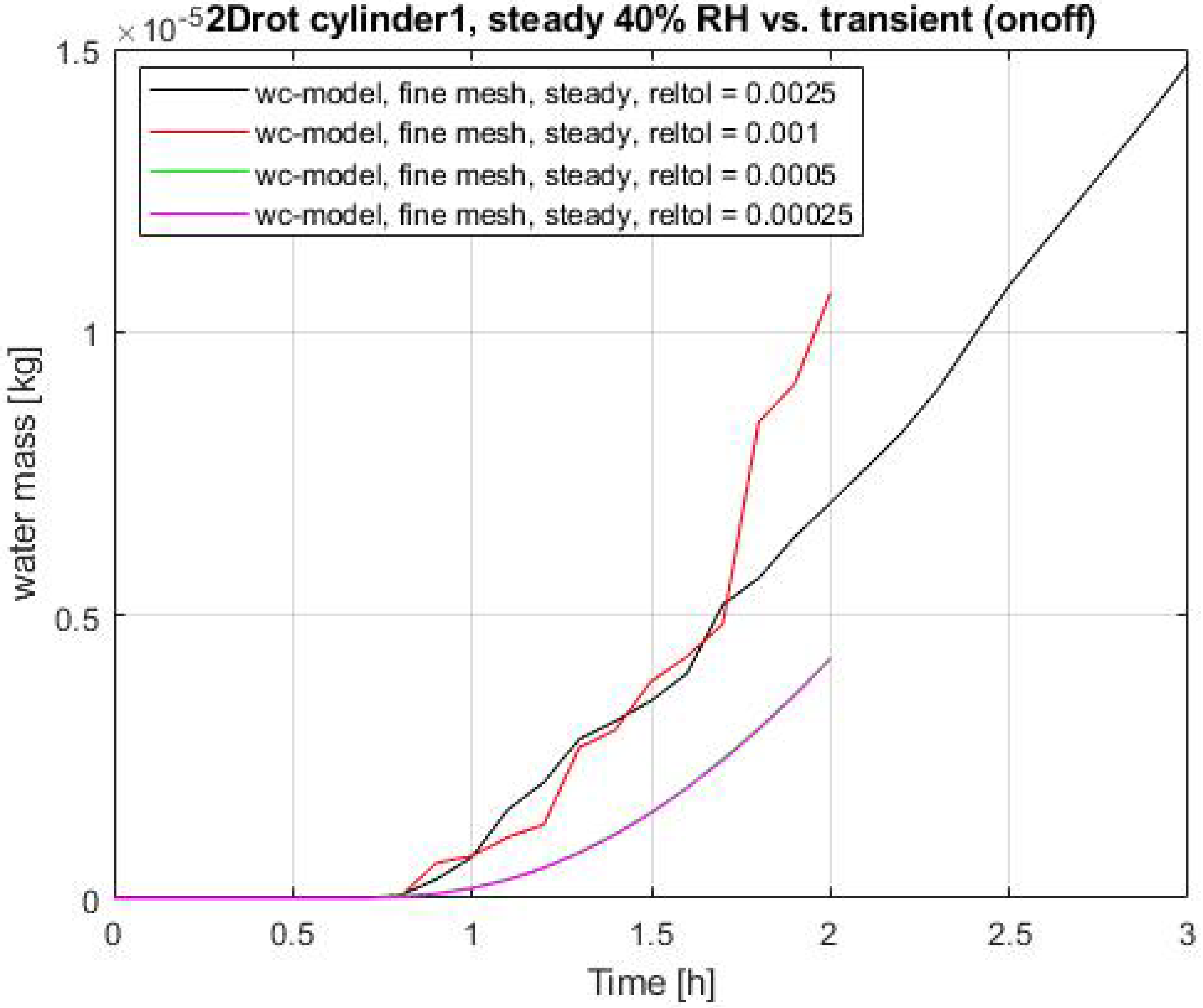
Mesh 1

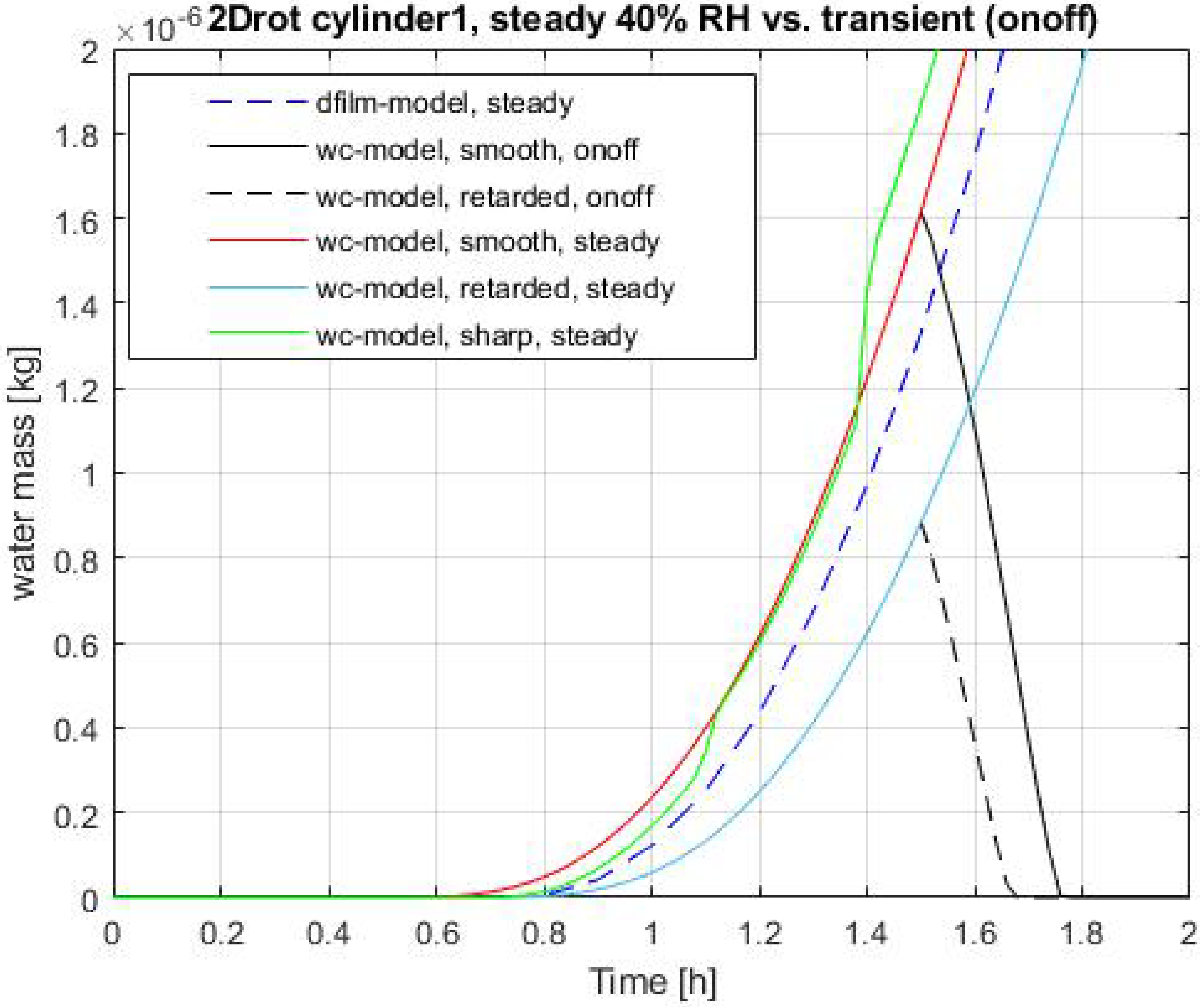


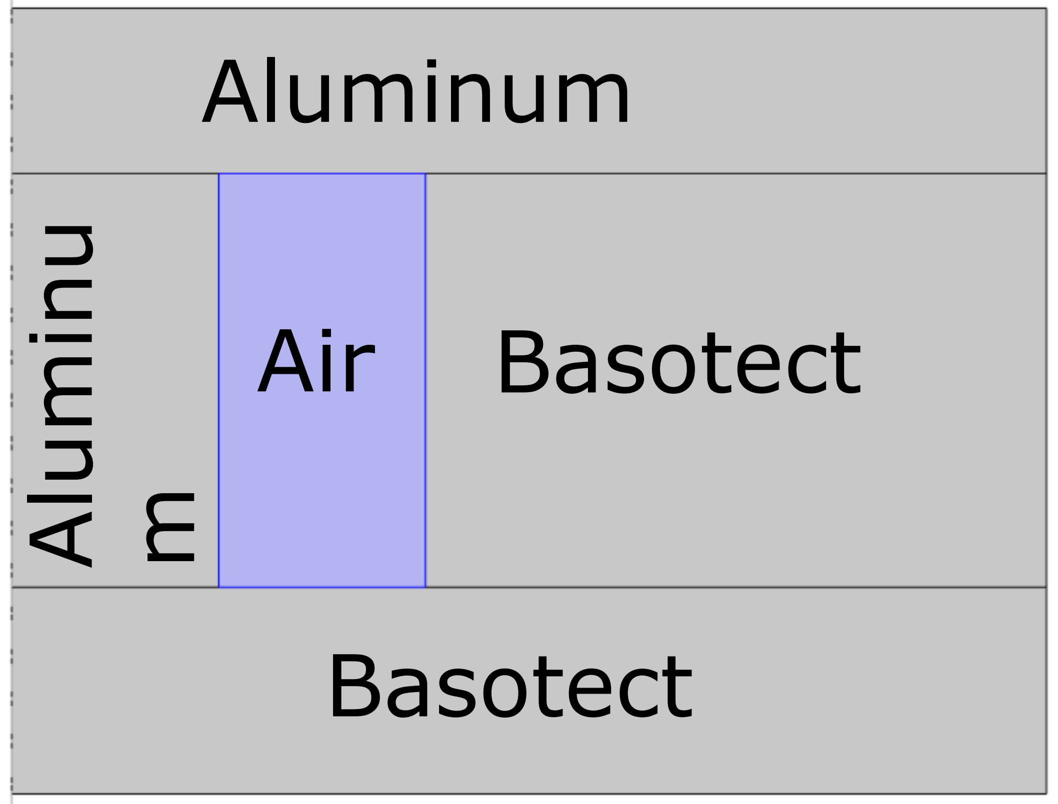
Mesh 2



Annex: Sponge-Model: Sensitivity to relative tolerance and $wc(\phi)$

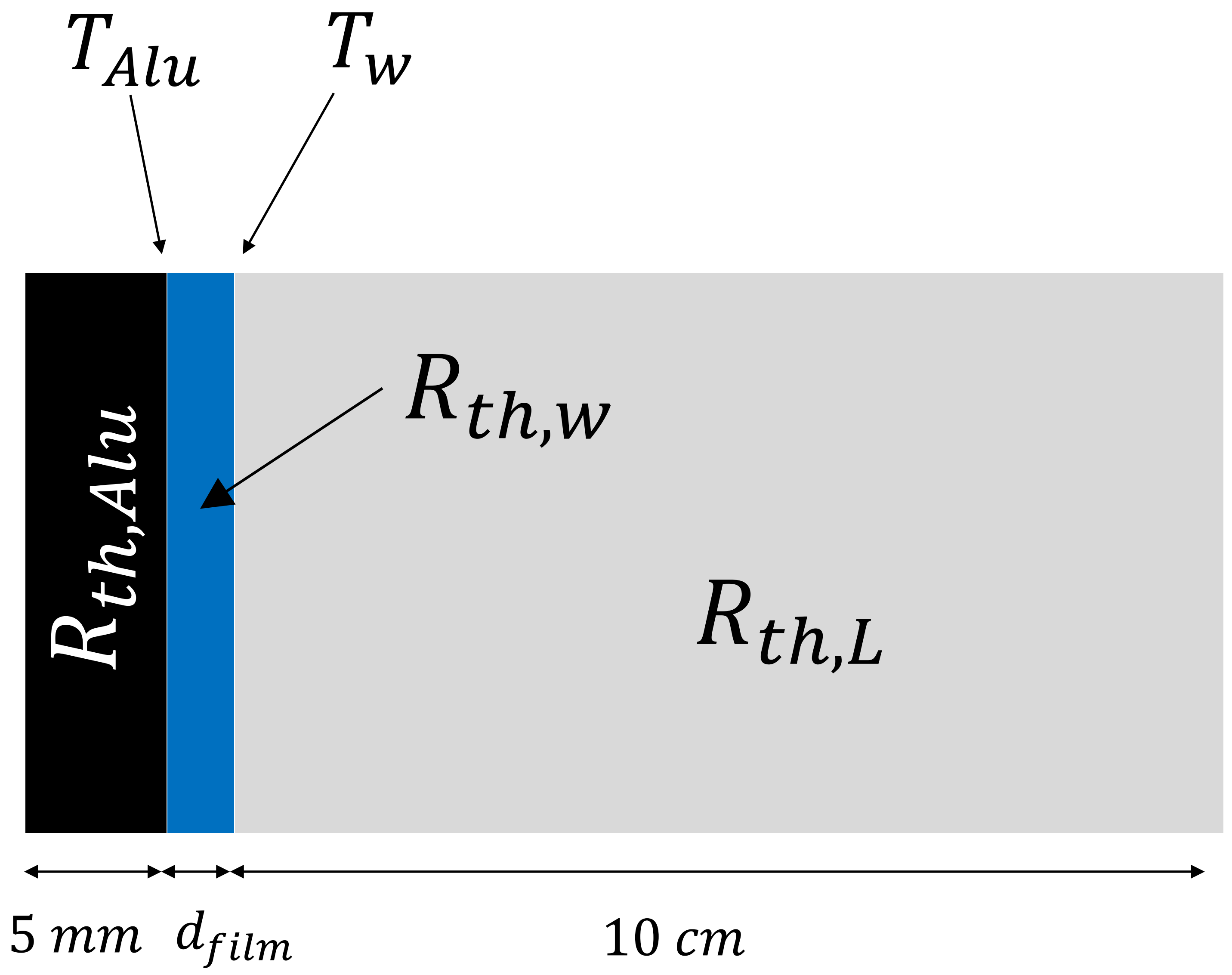






- Wc-model: Relative tolerance 0.00025 is required, even 0.0001 if evaporation takes place
- Even with low tolerance, sharp $wc(\phi)$ function leads to erroneus results. (→ sharp corner?)
- Smoothing $wc(\phi)$ leads to an earlier onset of condensation compared to dfilm-model

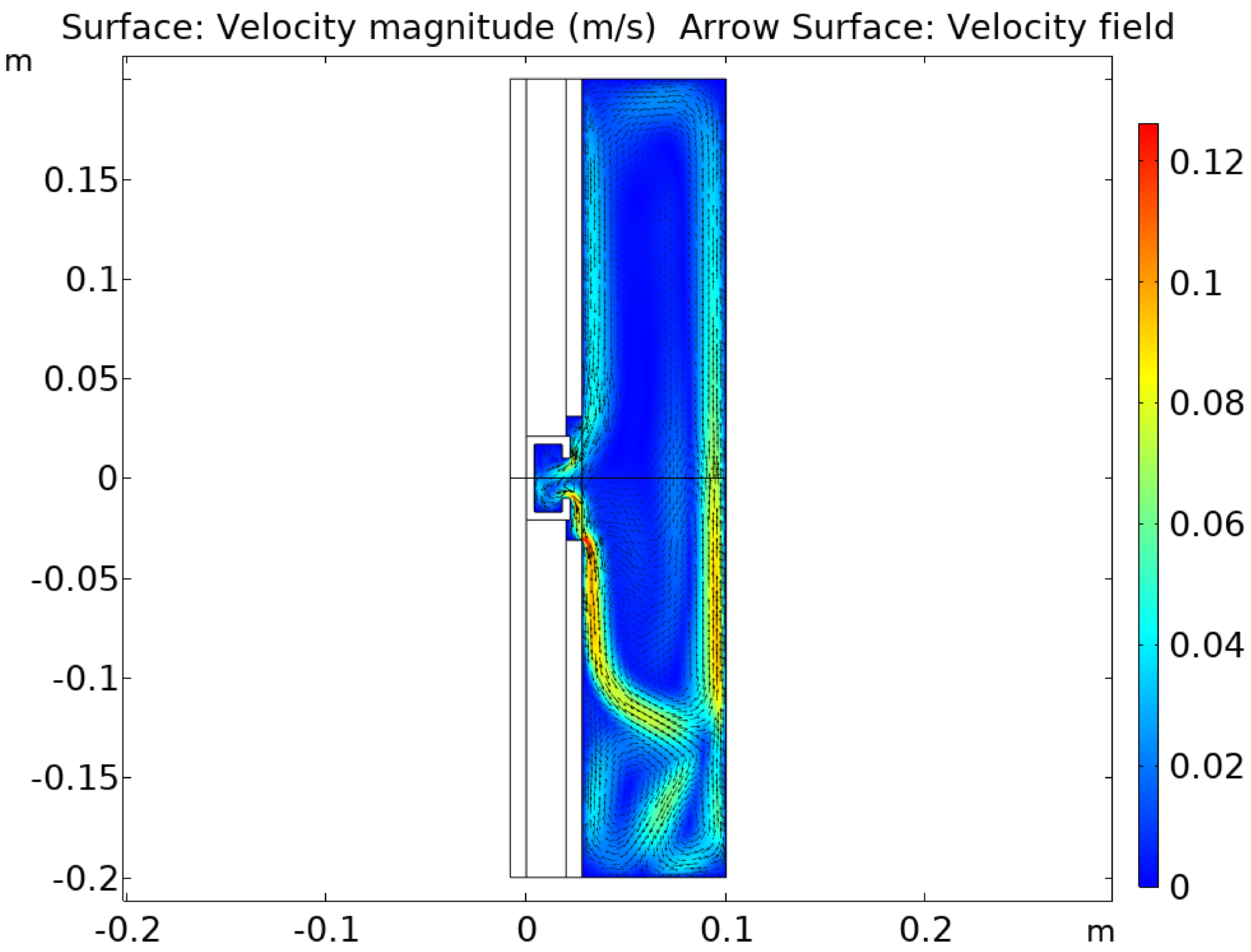
Annex: Influence of a condensed water film on heat transport?



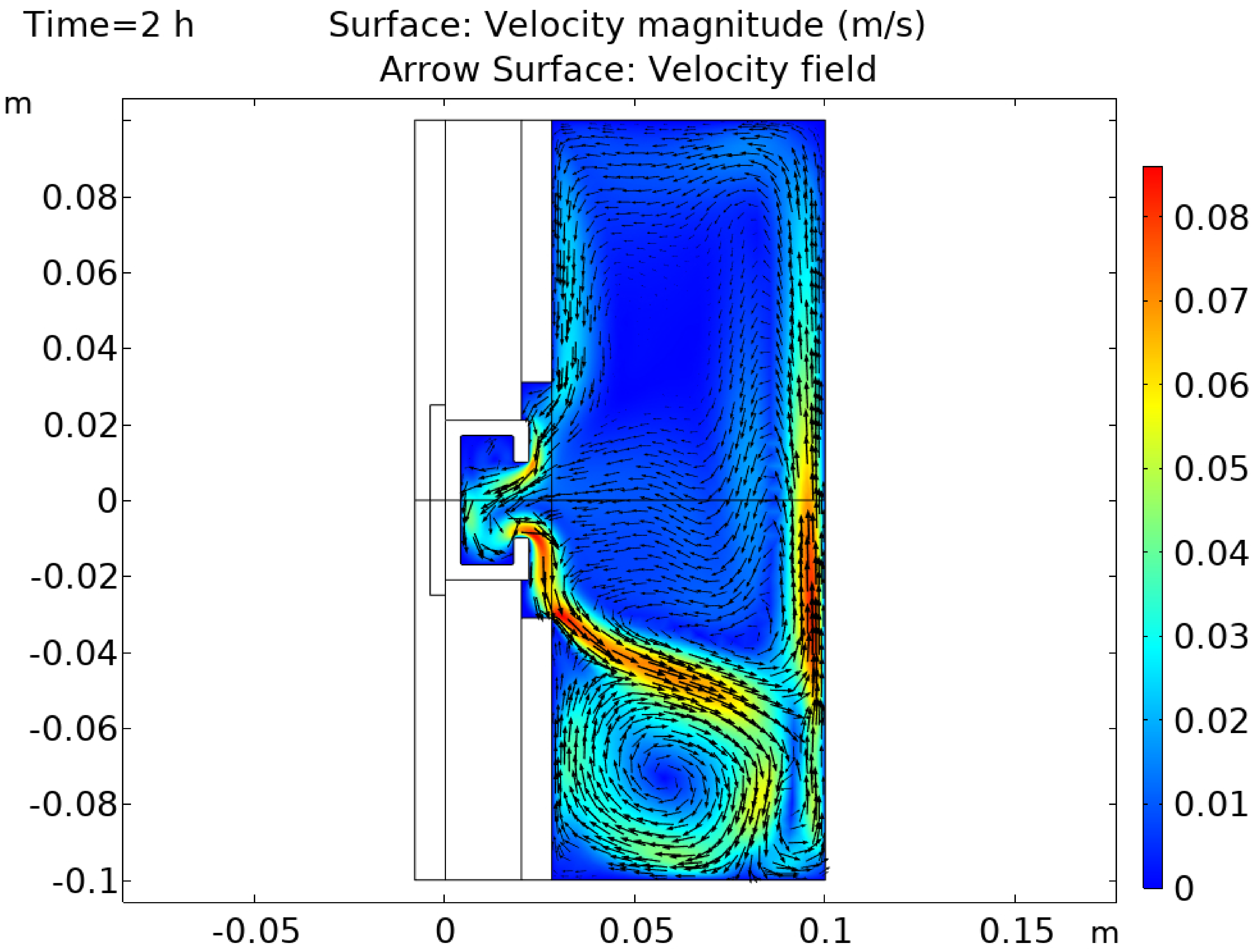
- $T_w = T_{Alu} + I_{therm} \cdot R_{th,w} > T_{Alu}$
- Model assumes $T_w = T_{Alu}$ → condensation rate overestimated
- $R_{th,w} \sim d_{film}$: Temperature difference $\Delta T = T_w - T_{Alu}$ increases with increasing d_{film}
- $\Delta T \approx 1\ K$ for $d_{film} \approx 1\ mm$
- Most simulated d_{film} are on the order max. $0.1\ mm$

→ **Most simulated water films are thin enough that $T_w = T_{Alu}$ is a valid assumption.**

Annex: Sensitivity of dfilm to vertical length for Geometry 2 (2D)

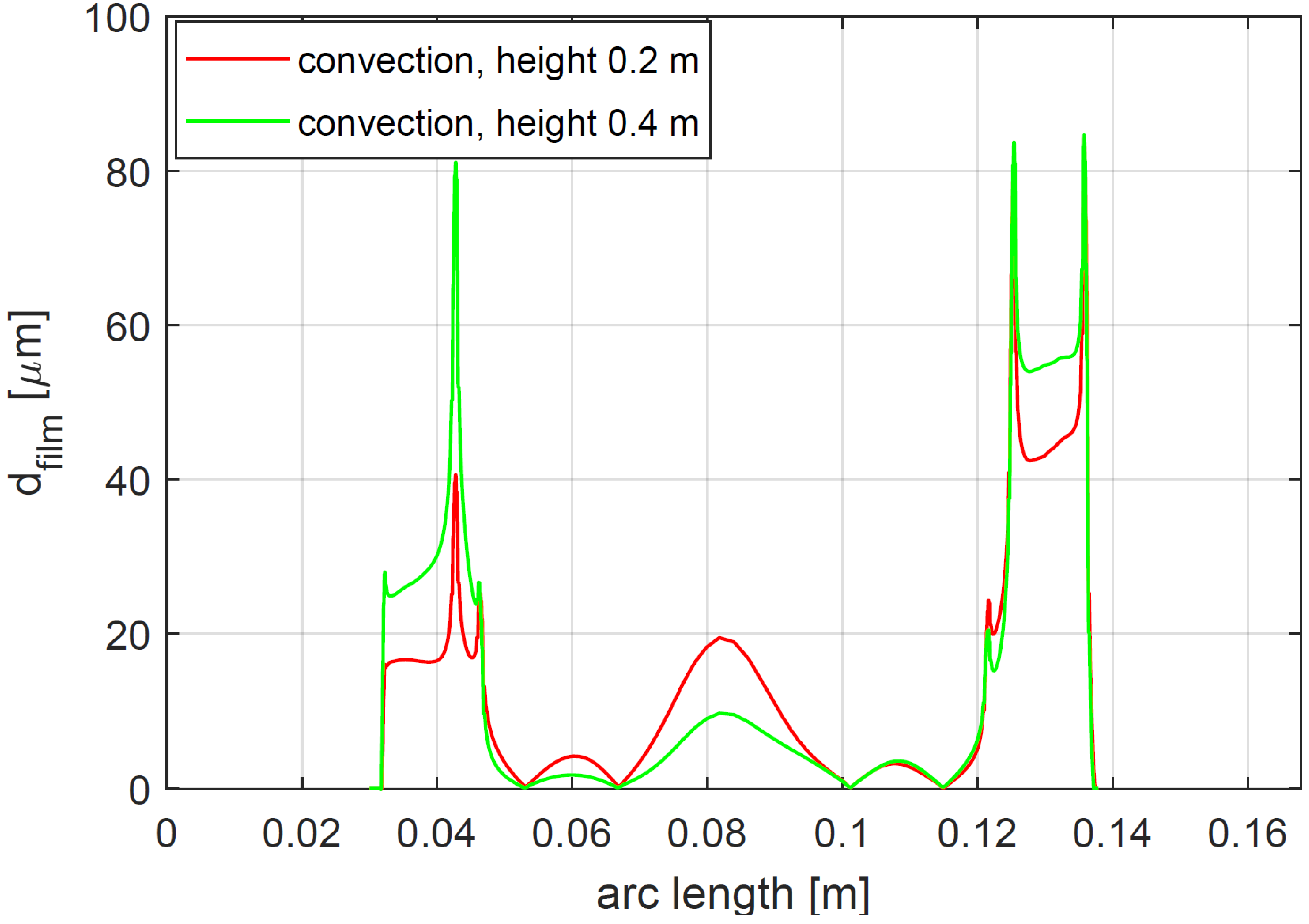


0.4 m vertical length: Max. downward velocity 13 cm/s

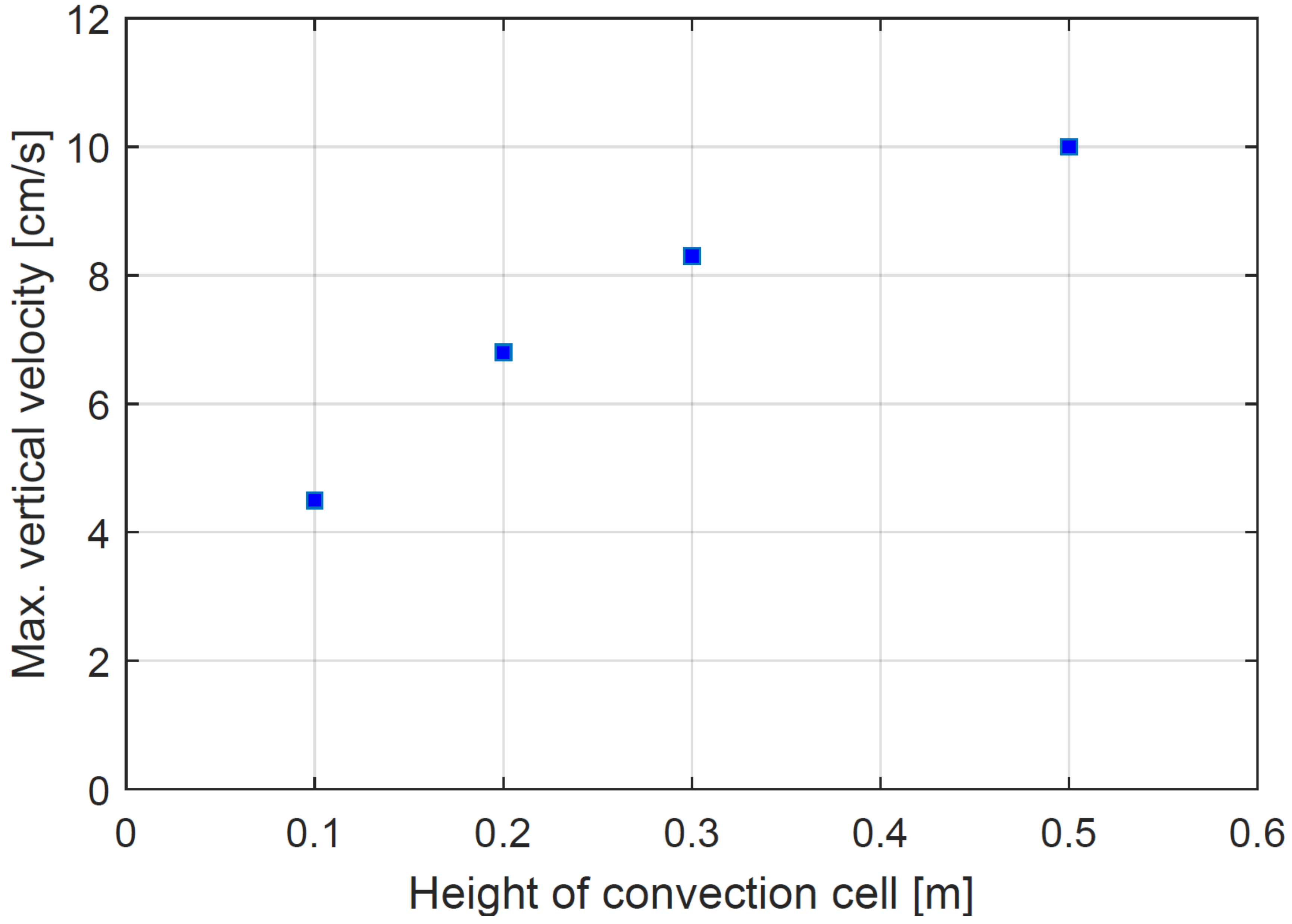


0.2 m vertical length: Max. downward velocity 10.5 cm/s

Annex: Sensitivity of dfilm to vertical length for Geometry 2 (2D)



Sensitivity of max. vertical velocity to vertical length in rectangular, homogeneous test cell ($\Delta T = 10\ K$):



- Exact amount of water condensed on alu depends on size of the convection cell
- Larger cell: More condensation on protruding surfaces, less condensation within C-profile

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