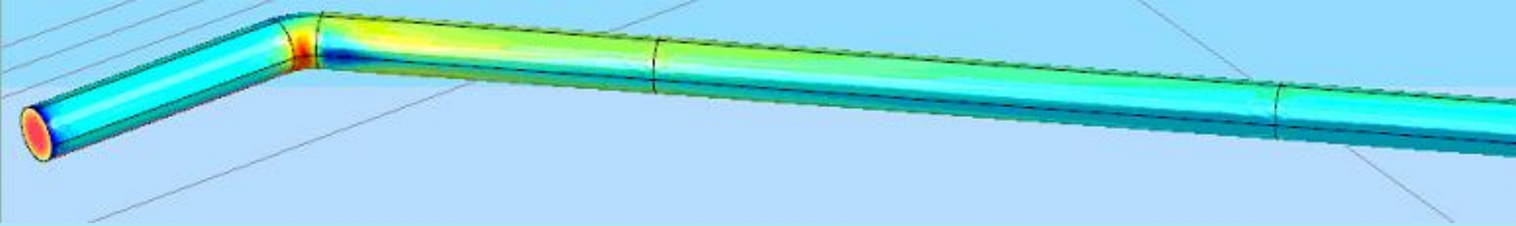

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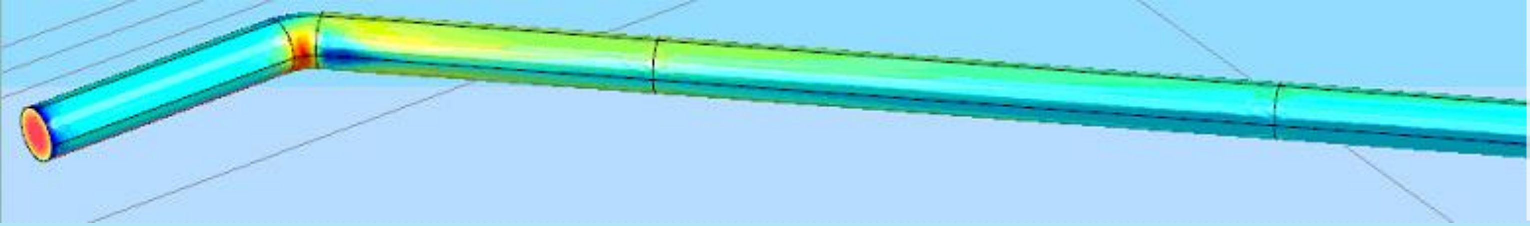
Thermo-Fluid Dynamics of Flue Gas in Heat Accumulation Stoves: Study Cases

Scotton P. – Rossi D.

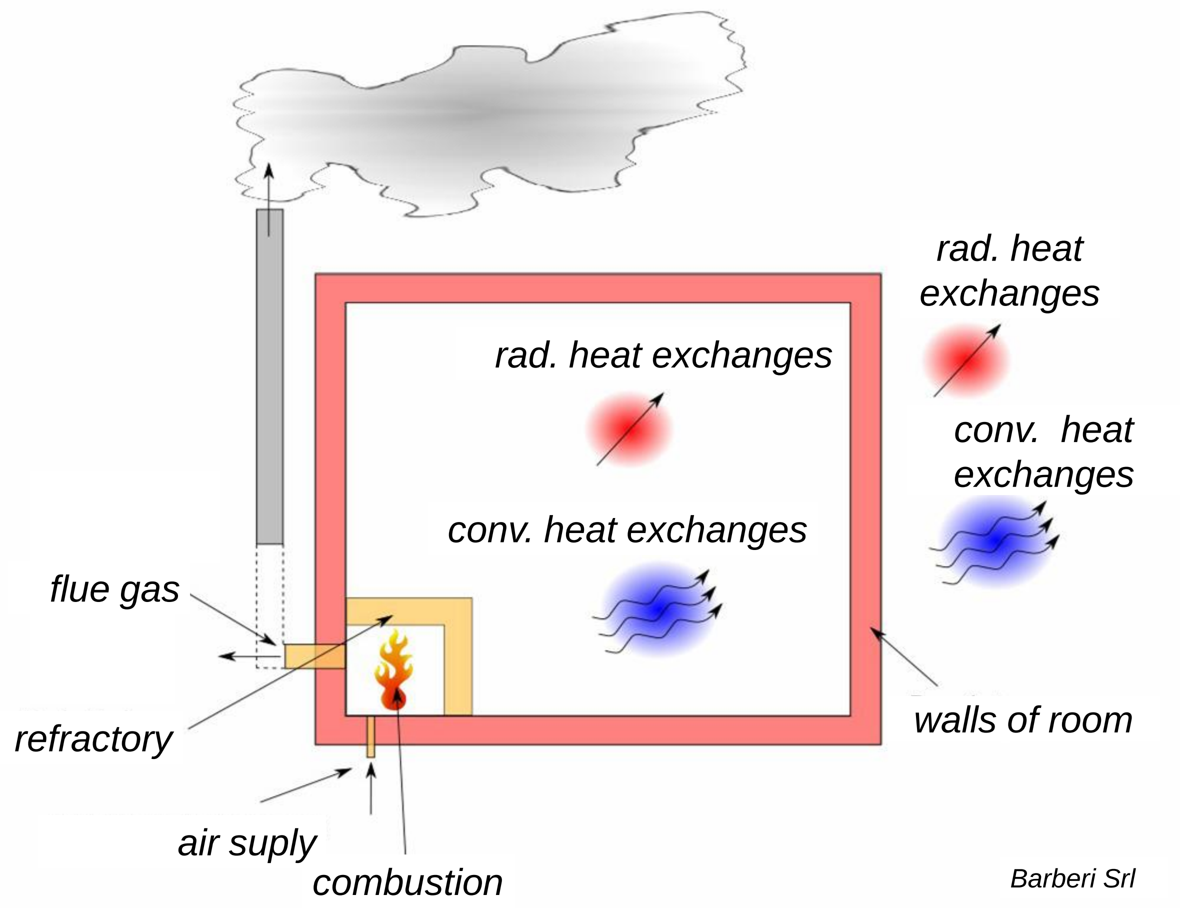
University of Padova, Department of Geosciences

Excerpt from the Proceedings of the 2012 COMSOL Conference in Boston



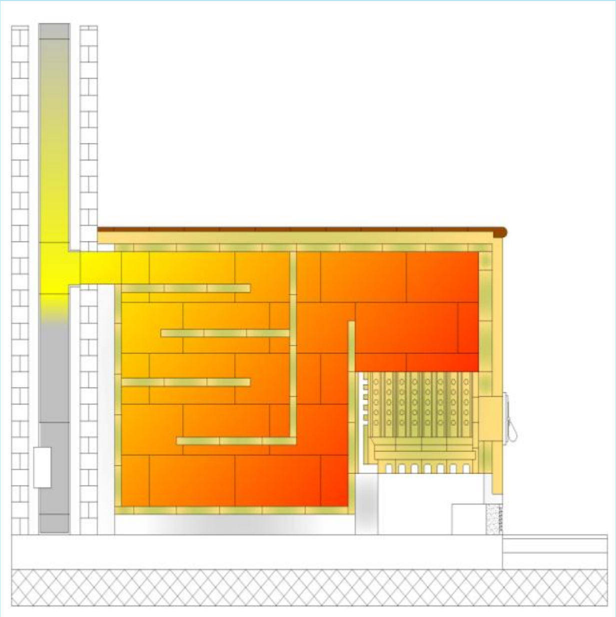
- *Description of Physical Problem;*
- *Theory of Turbulence and of Heat Transfer;*
- *Results of* →
  - *straight steel pipe*
  - *straight refractory pipe*
  - *curved refractory pipe*

Genelar Features of global system



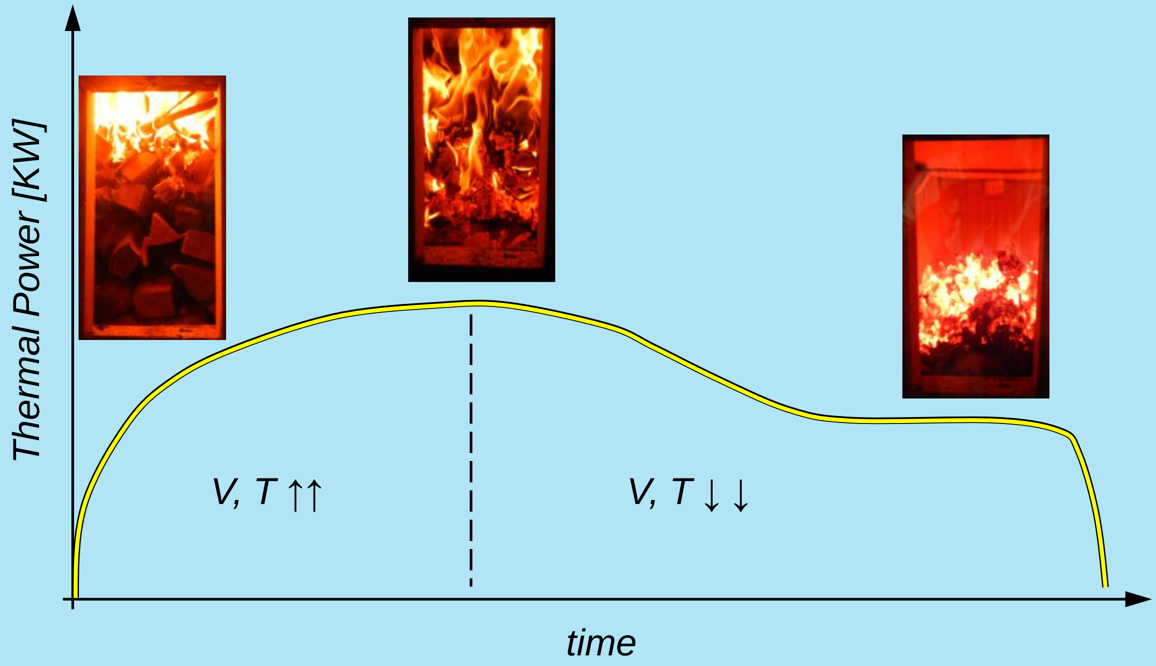
Genelar Features of heat accumulation stoves

historical heat accumulation stove "Sfruz"



a scheme of one modern stove

Burning Process of Woody Material



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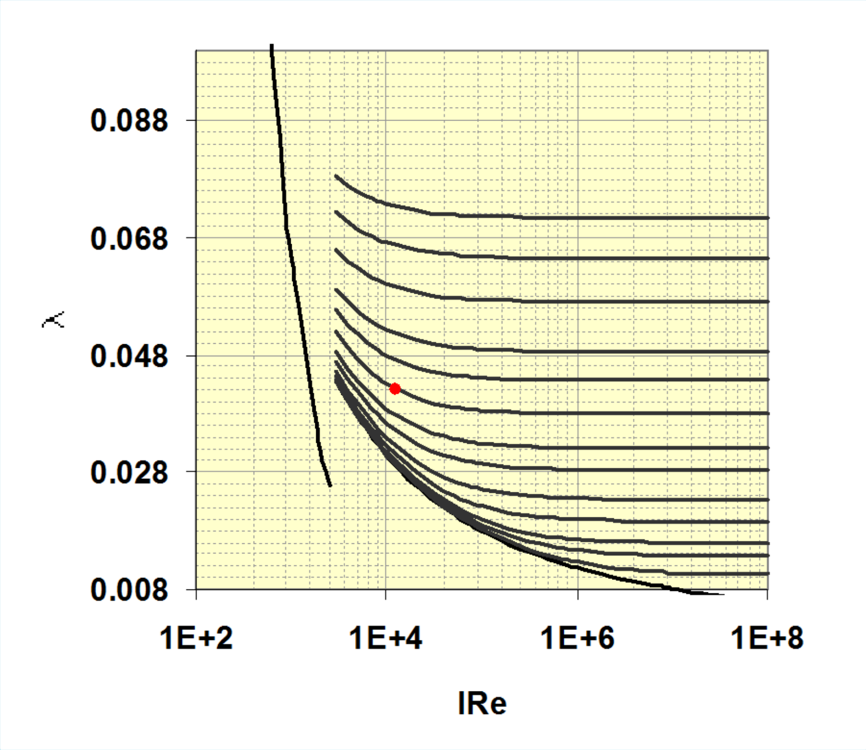
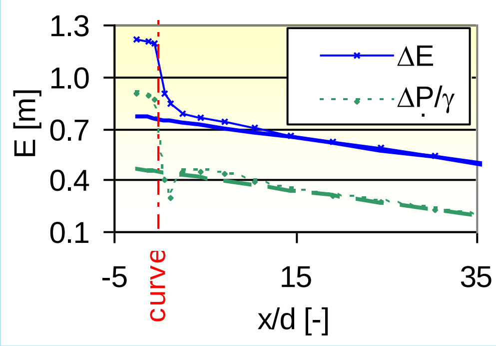
Burning Process of Woody Material

Temporal evolution of Reynolds number



Sharp Curve – Turbulent motion
IRe = 28400 x/D = 1.4



Theory of Turbulence: Transport Equations

Reynolds-averaged Navier–Stokes eq.

$$\begin{cases} \rho \cdot \left(\frac{\partial u}{\partial t} + (u \cdot \nabla) u \right) + \nabla \cdot (\rho u' * u') = \nabla \cdot \left[-p \cdot I + \mu \left(\nabla u + (\nabla u)^T \right) \right] + F \\ \nabla u = 0 \end{cases}$$

+

Turbulent energy eq.

$$\rho \cdot \frac{\partial k}{\partial t} + \rho u \cdot \nabla k = \nabla \cdot \left[\left(\mu + \frac{\mu_T}{\sigma_k} \right) \nabla k \right] + P_k - \rho \varepsilon$$

Turbulent Dissipation energy eq.

$$\rho \cdot \frac{\partial \varepsilon}{\partial t} + \rho u \cdot \nabla \varepsilon = \nabla \cdot \left[\left(\mu + \frac{\mu_T}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + C_{\varepsilon 1} \frac{\varepsilon}{k} P_k - C_{\varepsilon 2} \rho \frac{\varepsilon^2}{k}$$

where

$$\mu_T = \rho C_\mu \frac{k^2}{\varepsilon}$$

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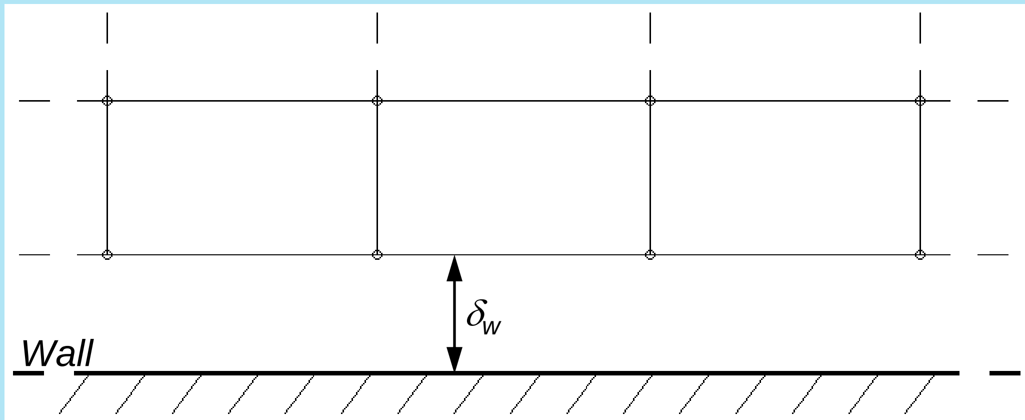
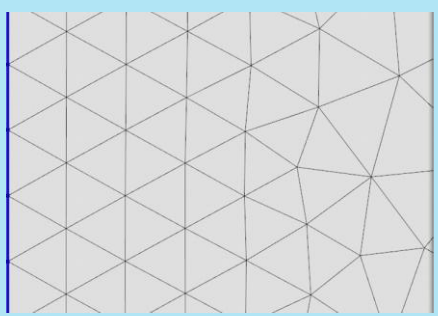
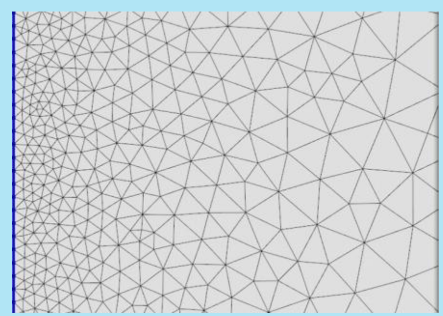
Theory of Turbulence: Wall Functions

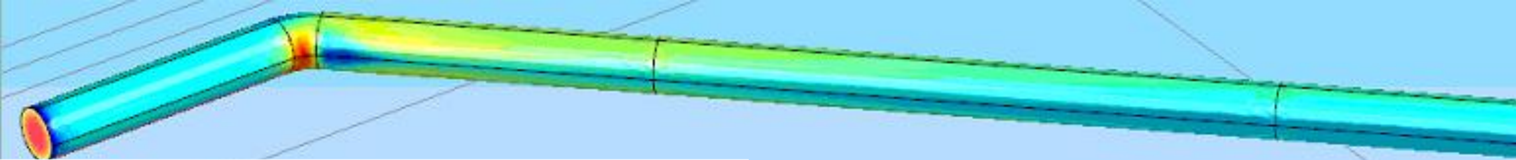


$$\delta_w \to \delta_w^+ = \frac{\rho u_\tau \delta_w}{\mu} \le 11.06$$

we will see also the influence that the choice of mesh can have on the results







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Theory of Heat Transfer

Heat transfer is guaranteed from three terms:

- *conduction* $q_i = -k \frac{\partial T}{\partial x_i}$
- *convection* $q = h A (T_s - T_\infty)$
- *radiation* $q = A \sigma T_s^4$

Equation of heat transfer

$$\rho C_p \left(\frac{\partial T}{\partial t} + (u \cdot \nabla) T \right) = -\nabla \cdot q + \tau : S - \frac{T}{\rho} \frac{\partial \rho}{\partial T}\bigg|_p \left(\frac{\partial p}{\partial t} + (u \cdot \nabla) p \right) + Q$$

heat flux by conduction (q)

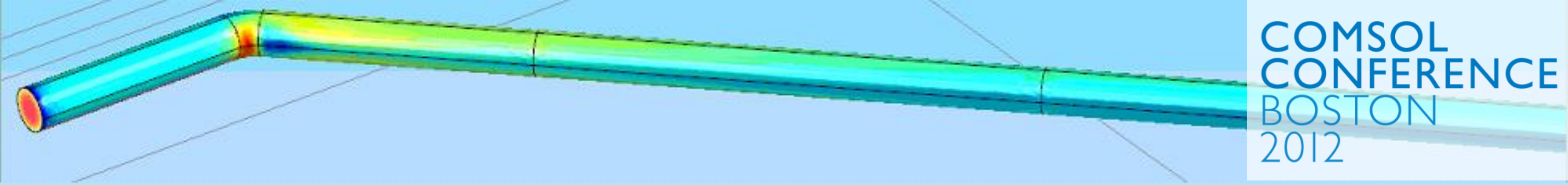
Equation of mass conservation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho v) = 0$$

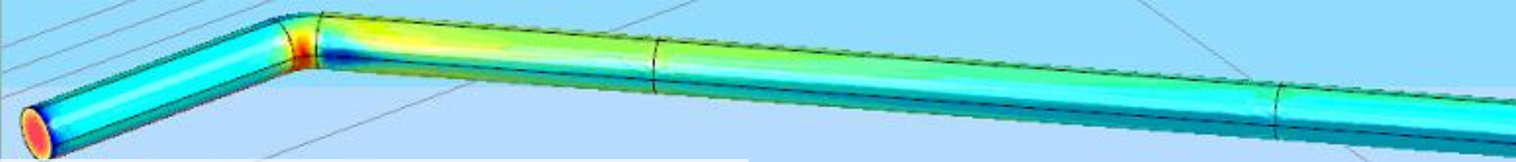
.. the conserved property is the total energy not the heat

$$\rho u (H_0 + \Psi) - k \nabla T + \tau \cdot u + q_r$$

heat flux by radiation (q_r)



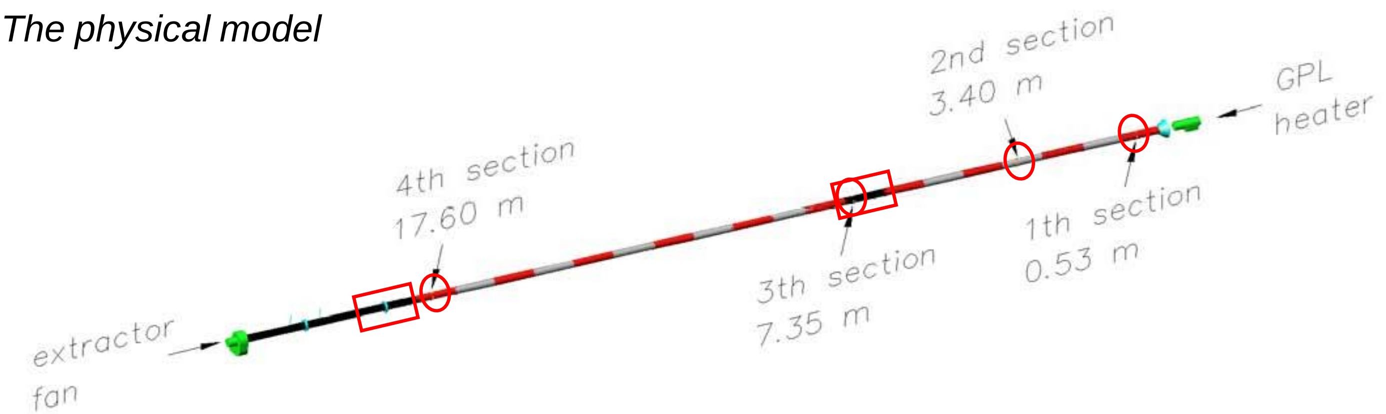
Study Case One: Straight Steel Pipe



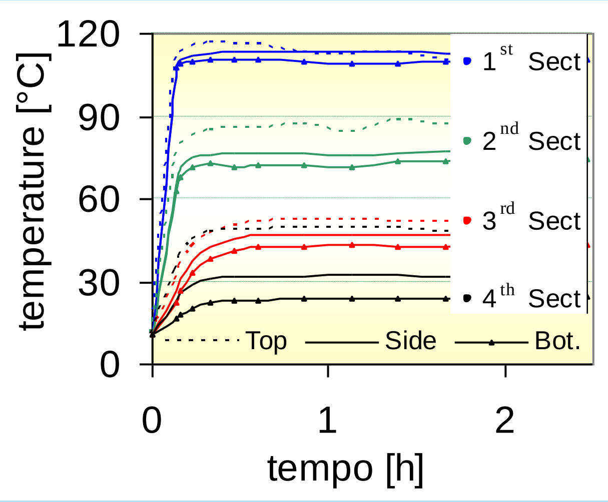
- Result: straight steel pipe

Straight steel pipe: physical model

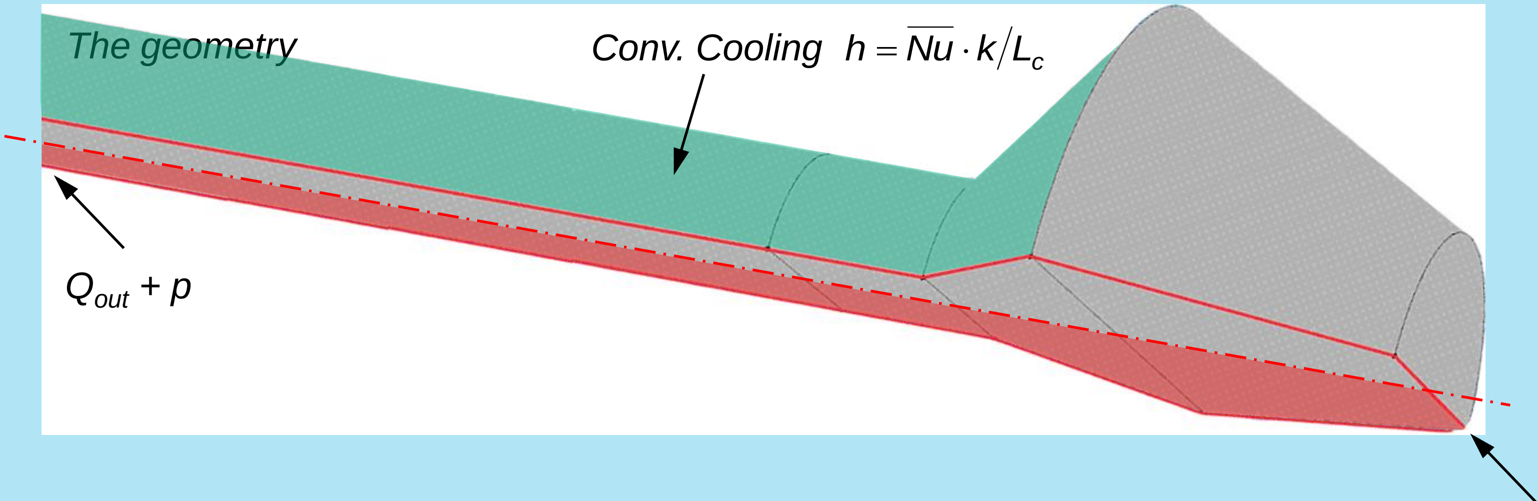
The physical model



Thermotechnical characteristics		
	stainless steel	*black steel*
thickness [mm]	*0.2*	*2.0*
emissivity [-]	*0.1*	*0.95*
conductivity [W/mK]	*17*	*50*



Straight steel pipe: 2D axial symmetry model

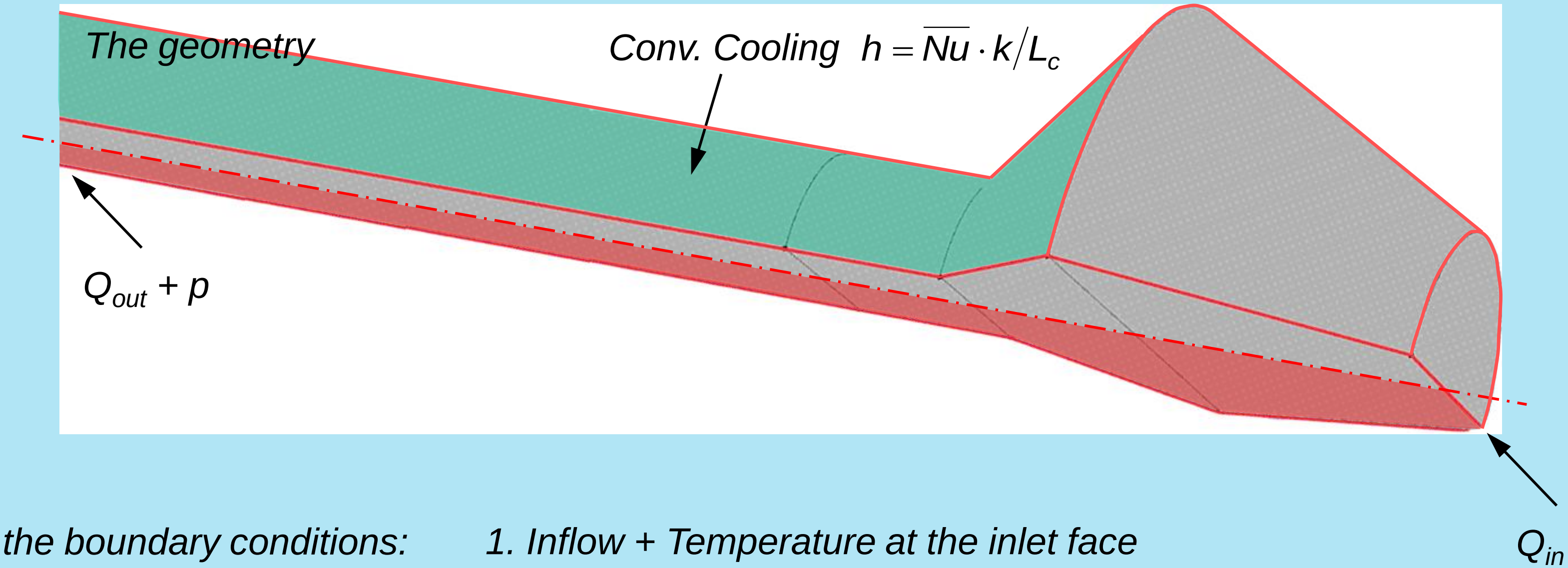


with the boundary conditions:

1. *Inflow + Temperature at the inlet face*
2. *Pressure + Outflow at the outlet face*
3. *Convective cooling on the outer surface using the coefficient*

$$h = \frac{\overline{Nu} \cdot k}{L_c} \quad \text{where} \quad \overline{Nu} = \left\{ 0.6 + 0.387 \cdot Ra^{\frac{1}{6}} \Bigg/ \left[1 + \left(\frac{0.559}{\mathrm{Pr}} \right)^{\frac{9}{16}} \right]^{\frac{8}{27}} \right\}^2$$

Straight steel pipe: 3D model

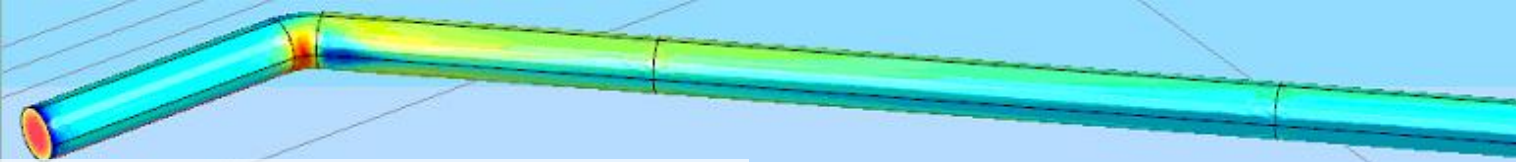


with the boundary conditions:

1. *Inflow + Temperature at the inlet face*
2. *Pressure + Outflow at the outlet face*
3. *Convective cooling on the outer surface*

+

4. *Buoyancy forces:* $F = \rho_R \cdot g \cdot \beta \cdot (T - T_R)$



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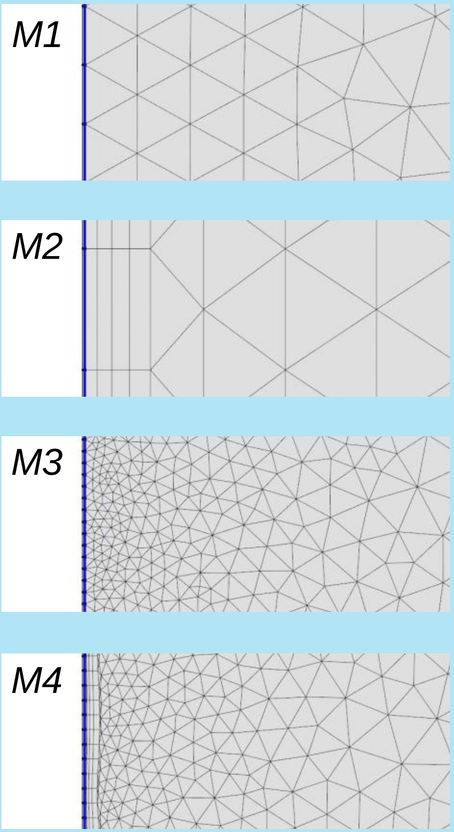
Straight steel pipe: 2D results

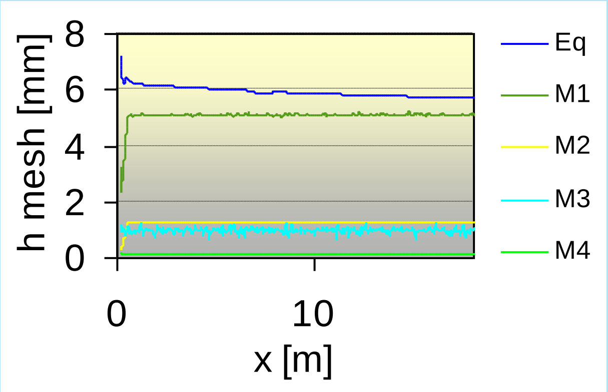


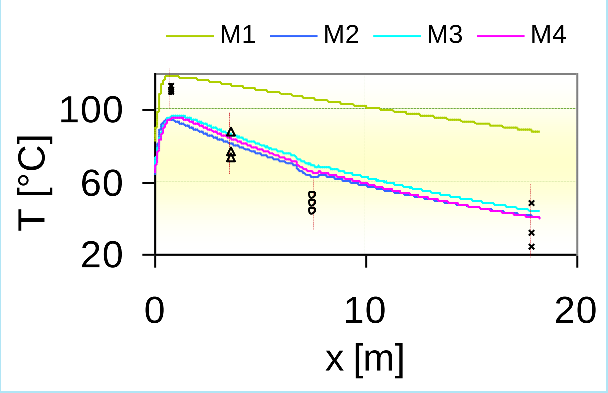
The choice of the mesh dimensions are fundamental for the quality of the results.

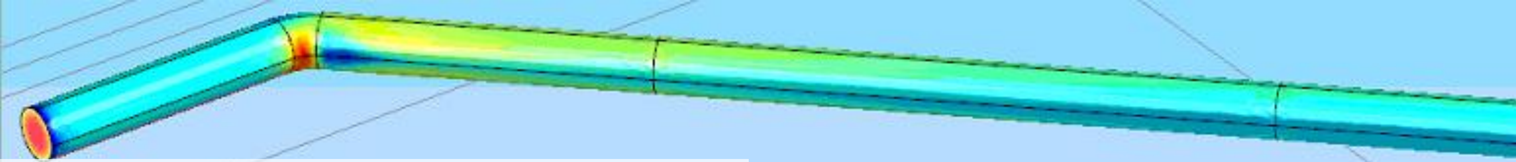
first of all, we have estimated the thickness of the first layer of cells adjacent to the wall with the equation:

$$h \leq 2\frac{11.06\nu}{u_\tau}$$

	Boundary Layer [mm]	Free Triangular [mm]	D.O.F 10^6
M1	No	$h_{BC} \leq 5.5$	0.245
M2	$h_{FL} \approx 1.0$	$h_{BC} \approx 11.0$	0.201
M3	No	$h_{BC} \approx 1.0$	1.722
M4	$h_{FL} \approx 0.25$	$h_{BC} \approx 1.0$	1.713





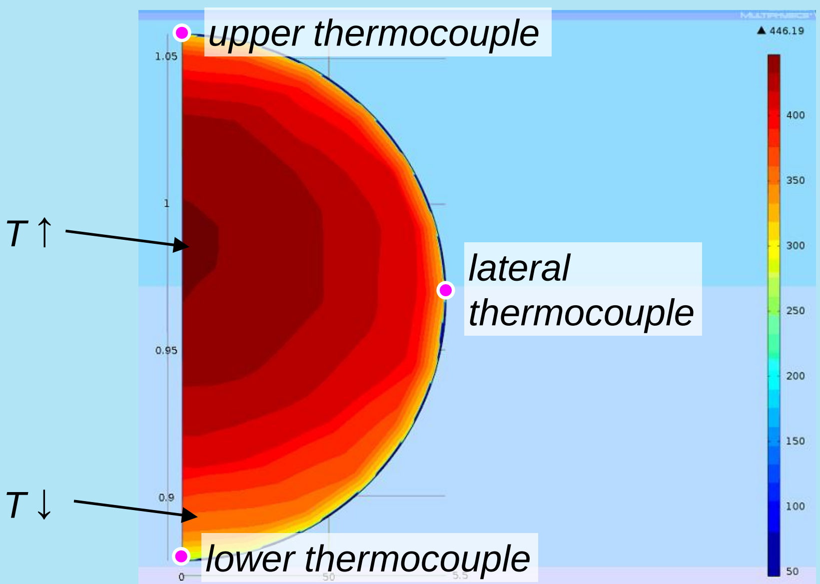


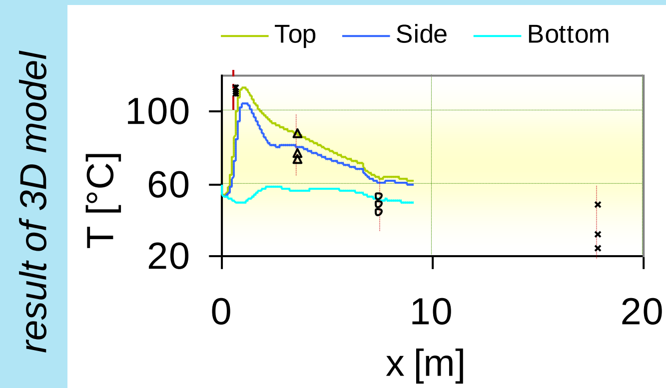
Straight steel pipe: 3D results

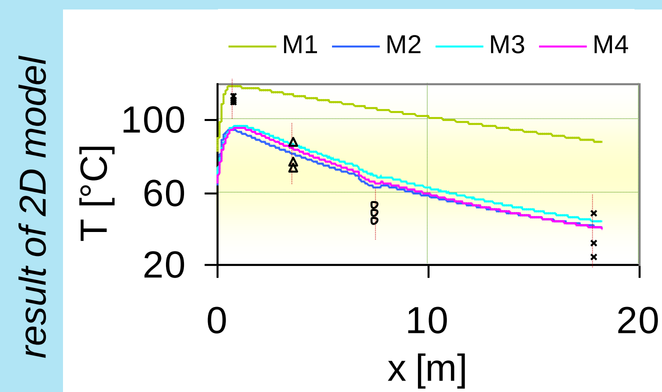
In the 3D model the buoyancy forces have been also considered:

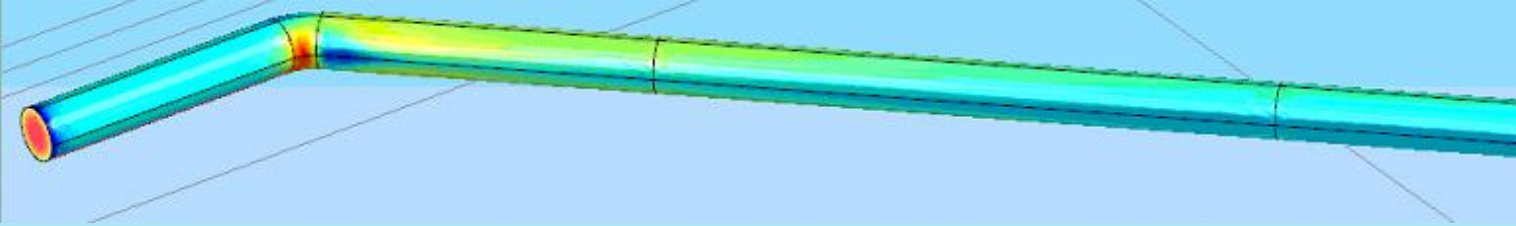
$$F = \rho_R \cdot g \cdot \beta \cdot (T - T_R)$$

It was possible to highlight the temperature differences at different positions in the cross section

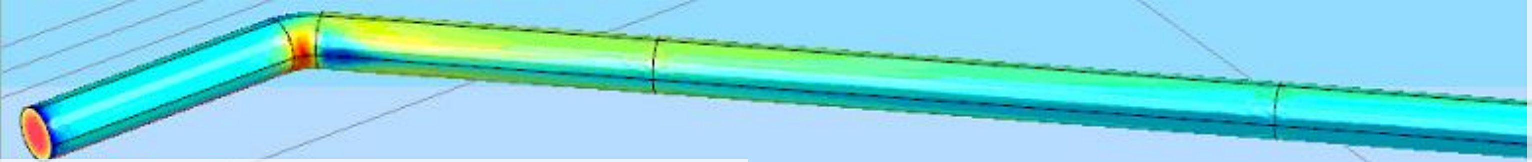








Study Case Two: Refractory Pipes



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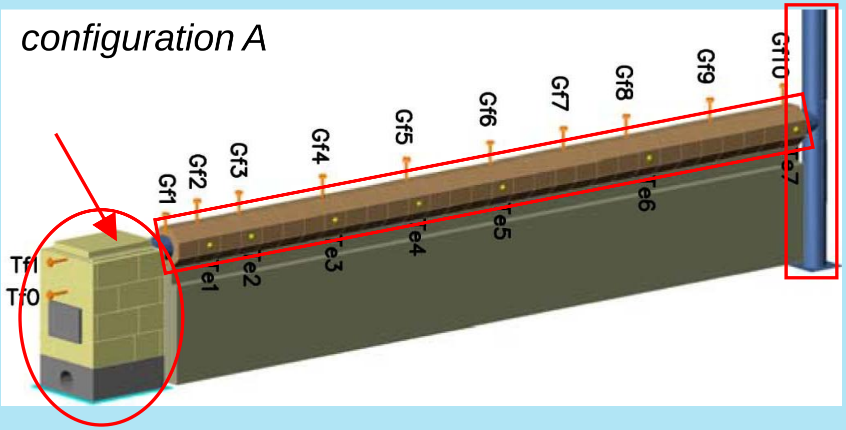
- Result: refractory pipes

Refractory Pipes: physical models



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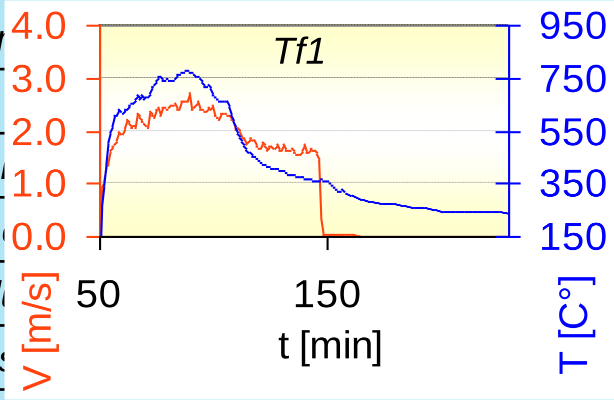
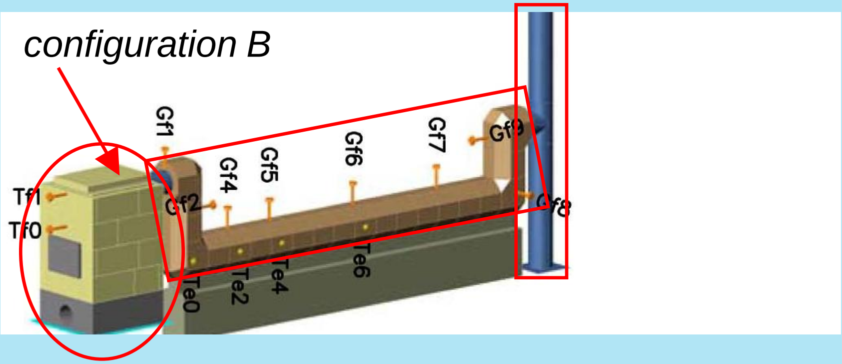
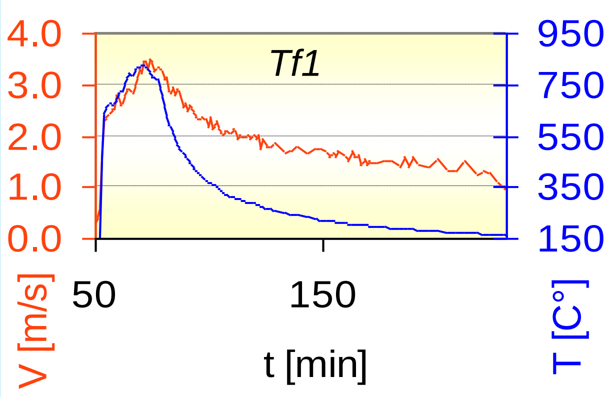
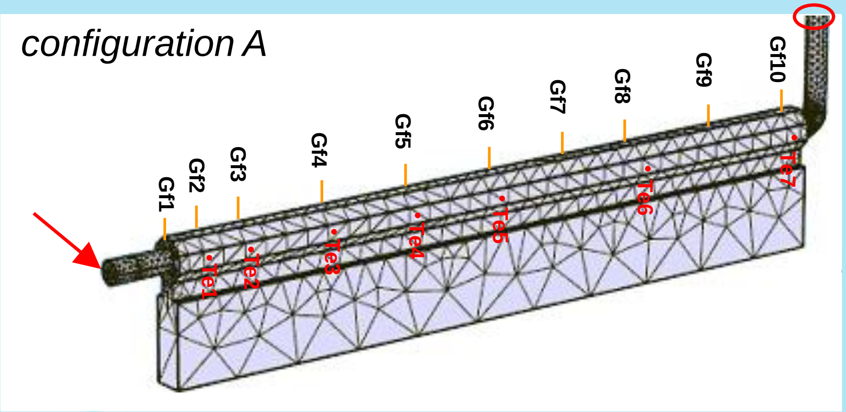
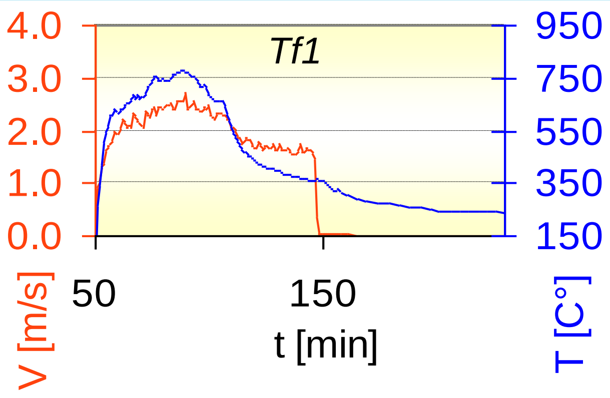






Refractory Pipes: numerical models

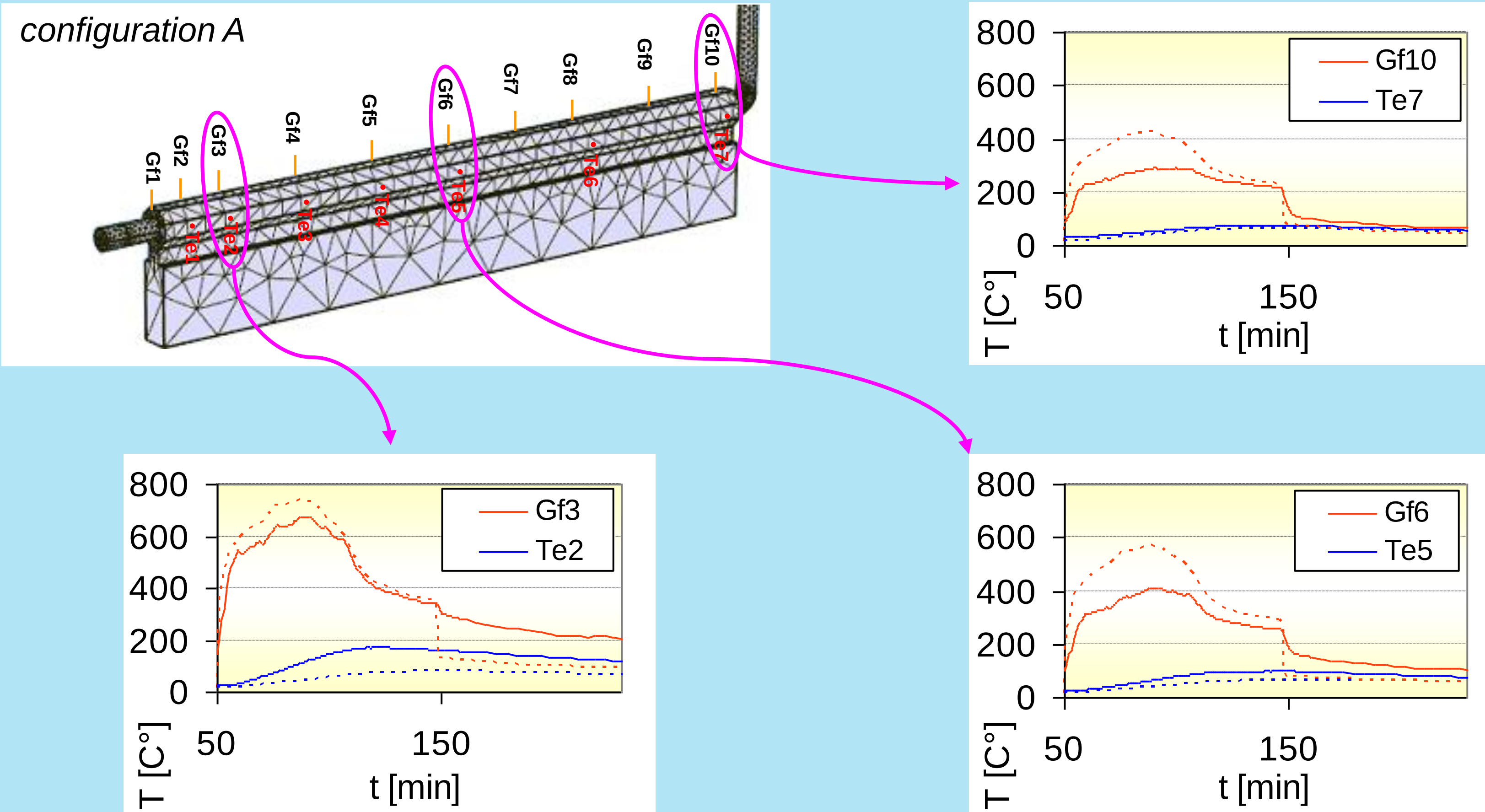




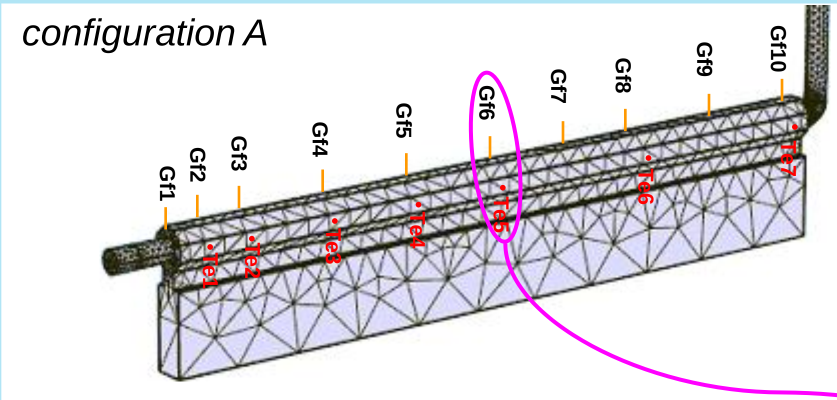
with the boundary conditions:

1. *Inflow + Temperature at the inlet face*
2. *Pressure + Outflow at the outlet face*
3. *Convective cooling on the outer surface*
4. *Buoyancy forces:* $F = -(\rho - \rho_R) \cdot g$

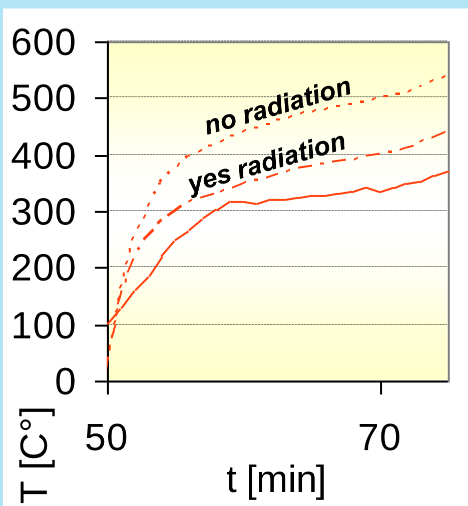
Refractory Pipes: numerical models

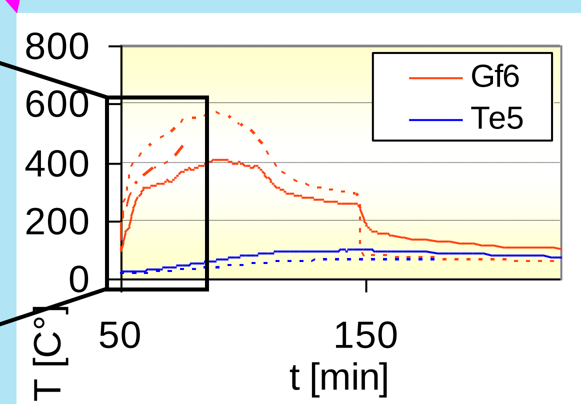


Refractory Pipes: numerical models



Those differences are significantly reduced introducing the radiation in participating media.





Conclusions

The very challenging experimental conditions (very high temperatures, very low pressures) induce to accept high errors of the order of 20%

The choice of the mesh influences, in an important way, the numerical solution, in particular, near a wall, the choice of the mesh could influence, importantly, the heat transfer through the wall

An important contribute on heat transport could be given from radiation absorption and emission phaenomena of particle fraction of the flue gas

THANKS FOR YOUR ATTENTION