Comparison between honeycomb and fin heat exchangers

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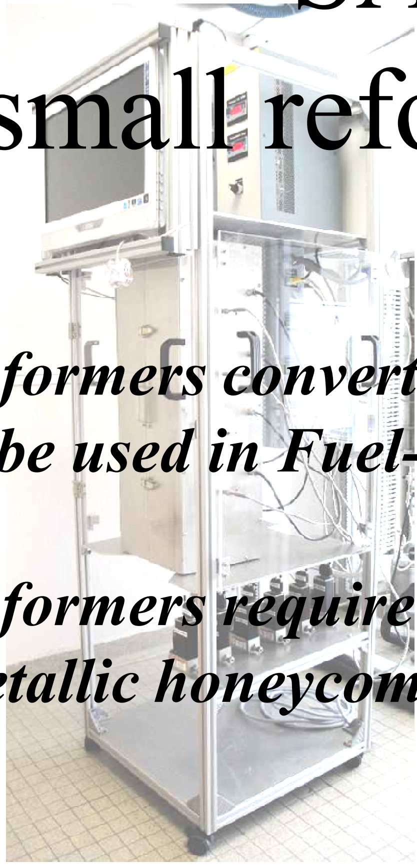
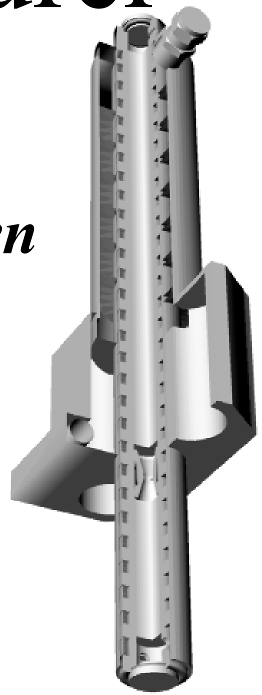
SAS SYNGAS :
small reformers manufacturer





Reformers convert hydrocarbon fuels in hydrogen to be used in Fuel-Cell Stacks

Reformers require catalysts supported, e.g.,on metallic honeycombs

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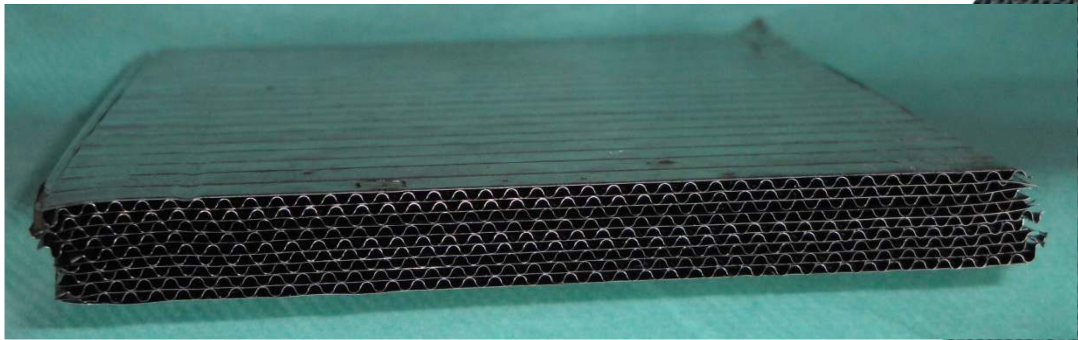
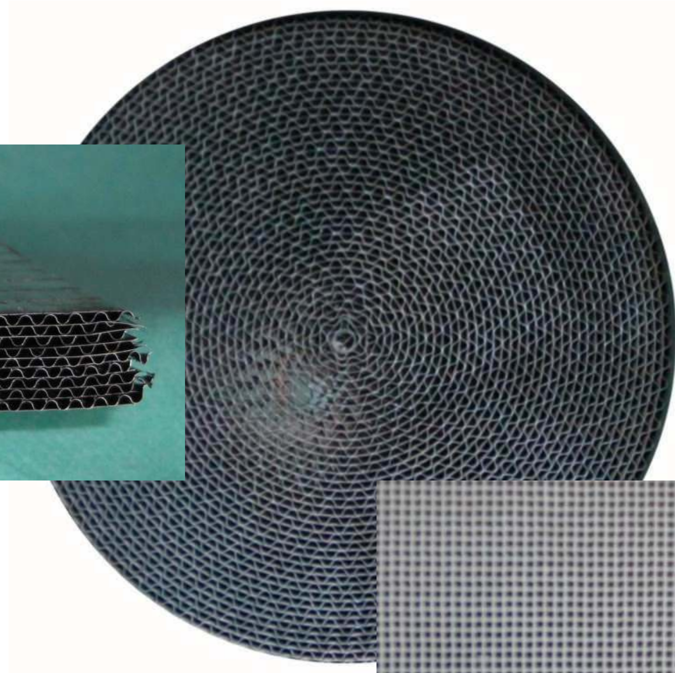
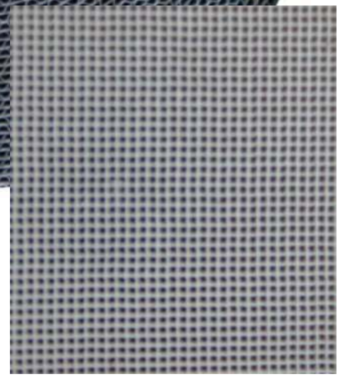
Honeycombs





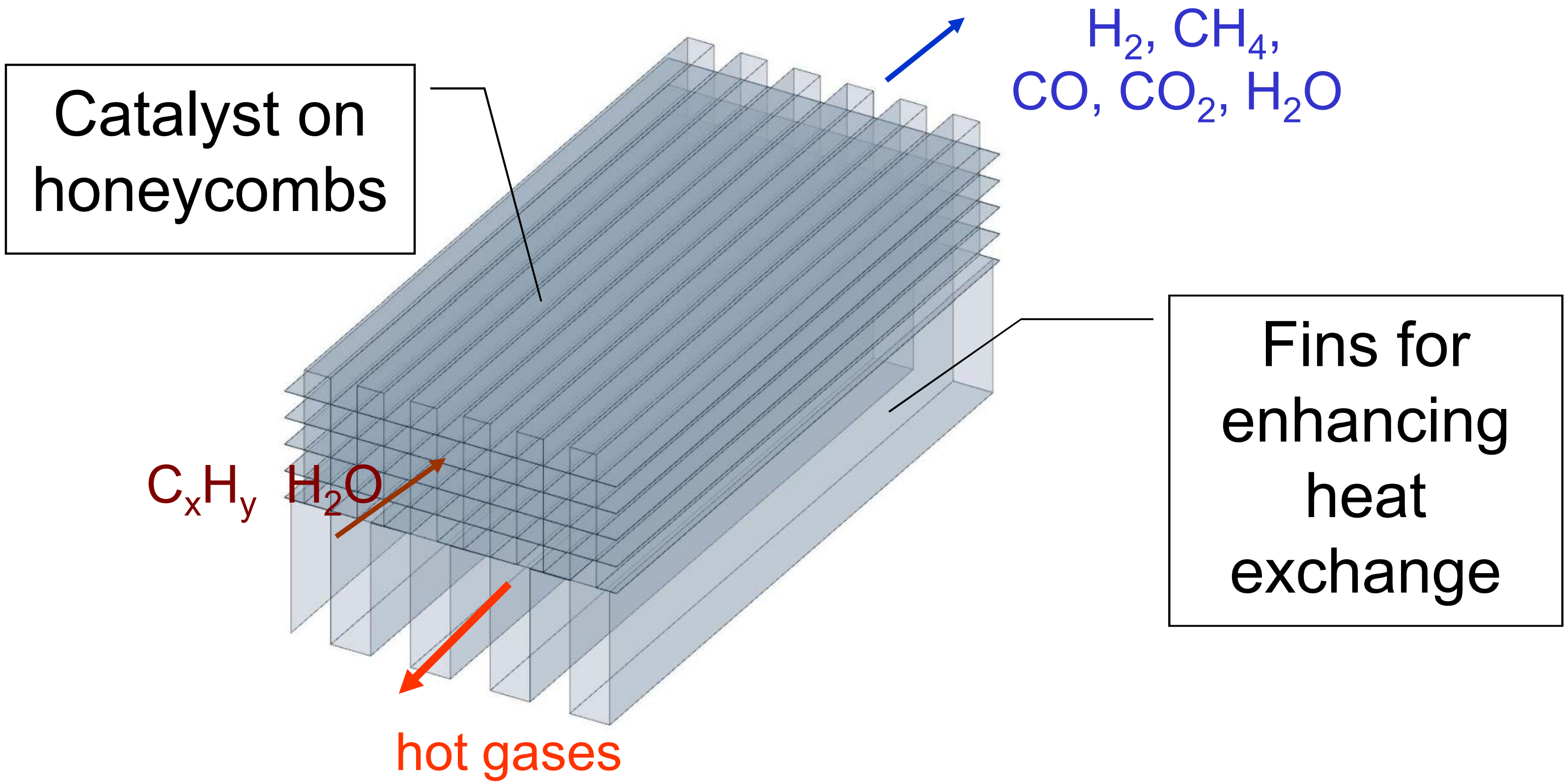


Metallic monolithic catalyst support

Metal : FeCrAl alloy or equivalent
Wall thickness : 0,05 mm or 0,1 mm
400 cpsi (cells of about 1.2 mm x 1.2 mm)

Honeycombs in steam-reformers

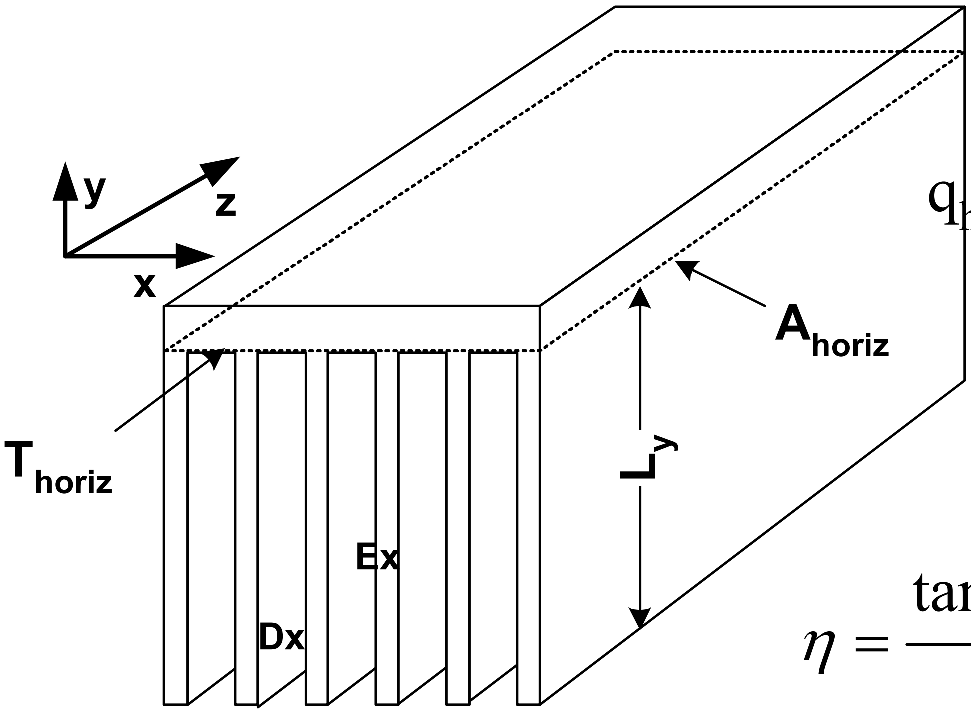
Steam-reformers reactor are heated by external exchange



Objective

Is-it possible to replace fins by honeycombs in the hot gases ?

The classical theory to calculate fin performance

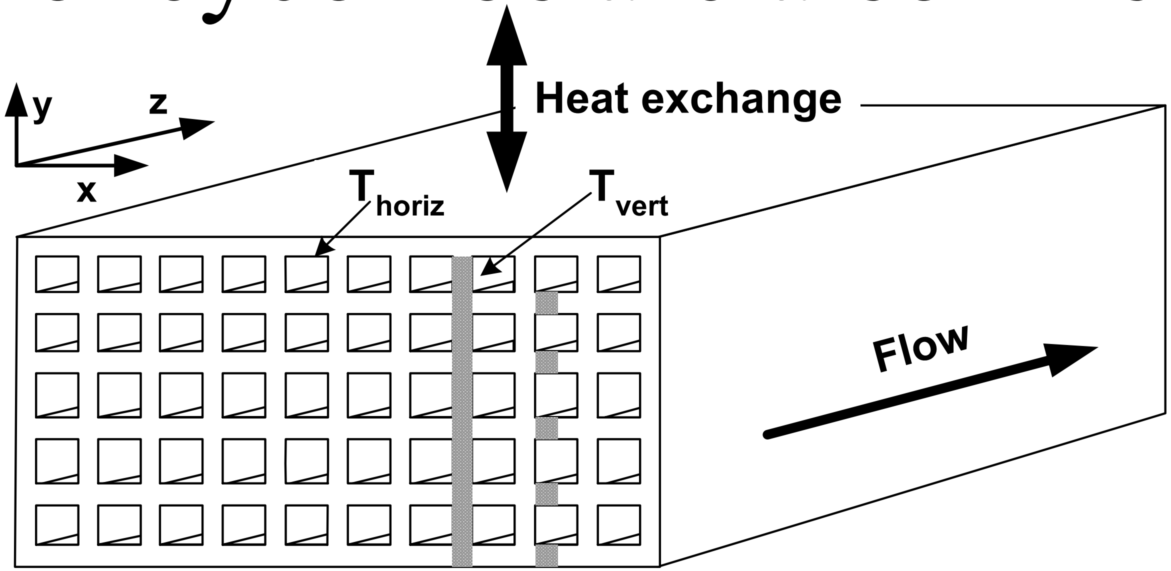


$$q_{horiz} = h\,F\,A_{horiz}\,(T_{fluid} - T_{horiz})$$

$$F = (D_x + 2\,\eta\,L_y)\,/(D_x + E_x)$$

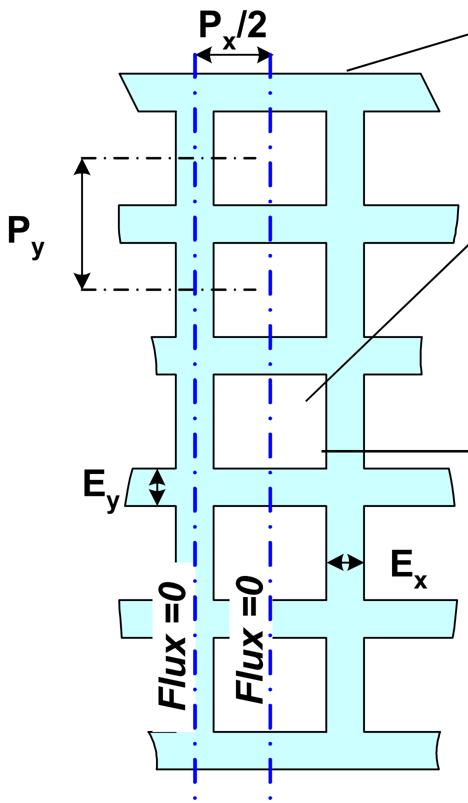
$$\eta = \frac{\tanh(mL_y)}{mL_y} \qquad m = \sqrt{\frac{2h}{k_s E_x}}$$

Honeycombs are also fins



The preceding fin efficiency calculation gives interest in using honeycombs for heat exchange
But is this theory valid ?

2D modelling in COMSOL 4.1



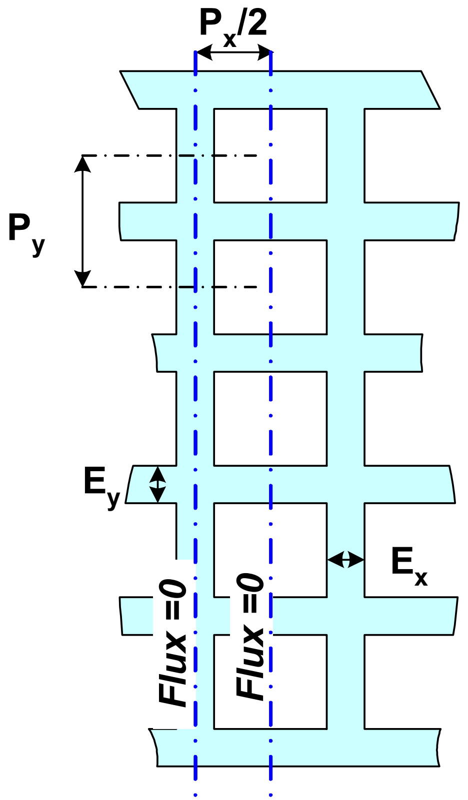
800°C

Air at 900°C

Heat transfer coefficient h

h is calculated by theoretical value of Nusselt in laminar flow

Heat flux from 10 channels with no radiative transfer



Px (mm)	1.30	1.25
Py (mm)	1.30	1.30
Ex=Ey (mm)	0.1	0.05
Coefficient h ($W/m^2/K$)	191	191
heat transfer rate (kW/m^2) COMSOL	95	74
Heat transfer rate (kW/m^2) Fin Model	108	96

Radiative transfer

For modeling radiative heat transfer in COMSOL the channels cannot be splitted by the middle

Discrepancy between COMSOL and the classical theory of radiative transfers in successive sheets

Discrepancy depends on channel geometry

3D modelling in COMSOL 4.1

Solid equation (no radiative heat transfer)

$$\nabla.(k_s \nabla T) = 0$$

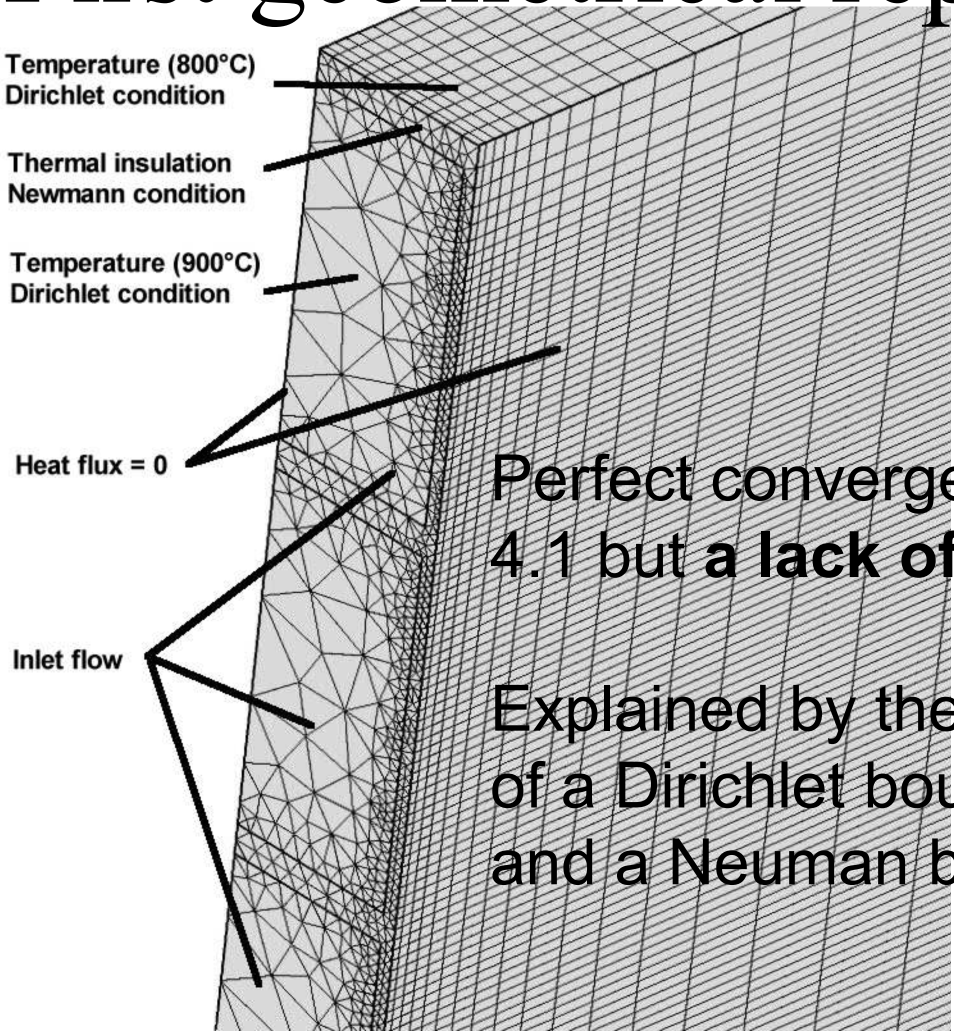
Fluid equations in laminar flow

$$\rho C \mathbf{u}.\nabla T = \nabla.(k_f \nabla T)$$

$$\rho(\mathbf{u}.\nabla)\mathbf{u} = \nabla.\left[-p\mathbf{I} + \mu(\nabla\mathbf{u} + (\nabla\mathbf{u})^T) - \frac{2}{3}\mu(\nabla.\mathbf{u})\mathbf{I} \right]$$

$$\nabla.(\rho\mathbf{u}) = 0$$

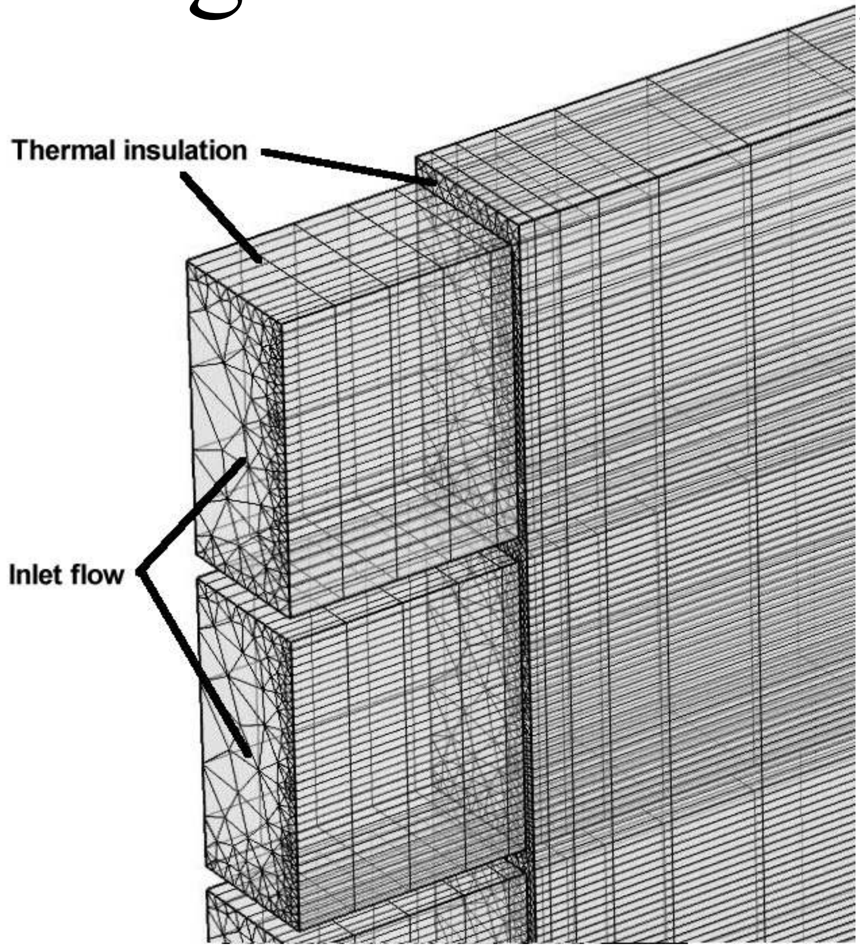
First geometrical representation



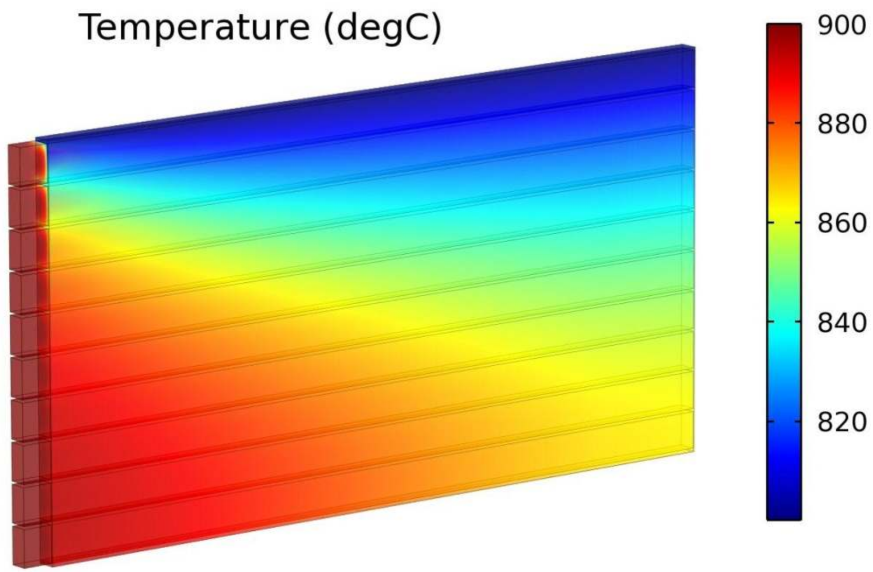
Perfect convergence with COMSOL 4.1 but **a lack of energy balance**

Explained by the proximity of a Dirichlet boundary condition and a Neuman boundary condition

New geometrical representation



3D results for honeycombs

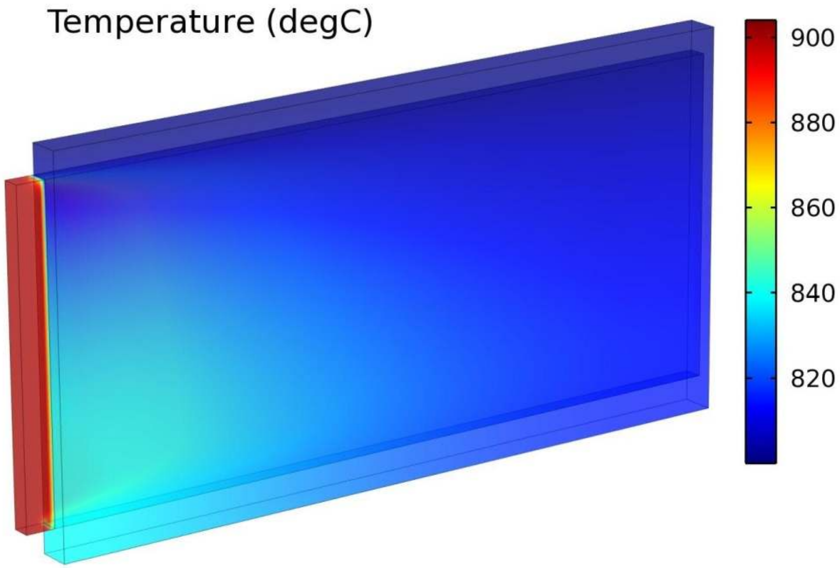


10 channels
1.3 mm 1.3 mm
Thickness : 0.1 mm

A strong discrepancy between COMSOL and fin theory

The longer is the channel the stronger is the discrepancy

Results for true fins



Fin = 1 channel
Height 12 mm (Py)
Width : 2.3mm (Px)
Metal thickness : 1 mm

Discrepancy is also confirmed for fins

Explanation

In the calculation of fin efficiency a global temperature is applied to the whole fin surface.

This is also the case in the COMSOL 2D modeling.

But in 3D modeling this is only true at entrance.

The honeycomb and the laminar flow between fins prevents mixing. This results in a temperature gradient which is disastrous to fin efficiency.

Conclusions

3D modeling using COMSOL throw light on heat transfer along the flow within a metal honeycomb and between fins.

Internal mixing of gases is required in honeycomb heat exchangers:
- By inserting mixing zones between small honeycomb lengths
- By boring holes with corrugations between channel layers.

Between fins, turbulence promoters should be used to break laminar flow an avoid the temperature gradient in the direction of heat flux.

Acknowledgment

The COMSOL model was built by Patrick Namy of SIMTEC and used by Paul Gateau of SYNGAS

The mathematical problem related to the Dirichlet and Neuman boundary conditions was resolved by Nicolas Huc of Comsol France.