Comparison between Honeycomb and Fin Heat Exchangers

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Abstract: This work is a part of a project to reduce the weight of heat exchangers in steam reforming reactors. Metal honeycombs are used as catalyst supports for a wide range of appliances. In contact with a planar heat exchange surface, the honeycombs can be considered as complex fins. A heat transfer model developed for fins was used to optimize the geometry. This model was compared with 2D simulations using COMSOL Multiphysics 4.1 for sheet metal from 0.05 mm to 0.2 mm thick. The cell dimensions were about 1mm. There was a good correlation when the fluid temperature was the same in all cells. There was, however, significant discrepancy when compared with a 3D simulation with laminar flow. Honeycomb cells produced a temperature gradient which reduced the heat transfer. There was less discrepancy for the thicker fins but there was also a gradient in the fluid owing to laminar flow. The radiant transfer was also investigated using 2D simulation. Modeling using COMSOL revealed the drawbacks of using honeycombs in steam reforming reactors. 3D modeling showed that a careful representation of the inlet was needed for realistic results.

Keywords: heat exchanger, fin, honeycomb.

1. Introduction

SYNGAS produces prototype reforming reactors [1,2] which convert hydrocarbons into hydrogen by steam-reforming.

The process requires a heat exchange between a hot fluid in contact with the wall of the catalytic reformer and the mixture of hydrocarbons and steam passing through the reformer.

Figure 1 shows a monolithic metal honeycomb used as a support for the catalyst. It can be combined with a fin heat exchanger (Figure 2).

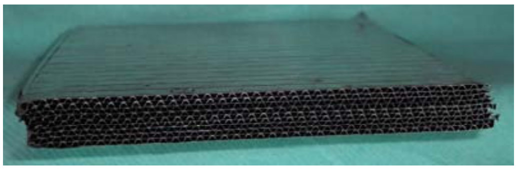

Figure 1. Monolithic catalyst support

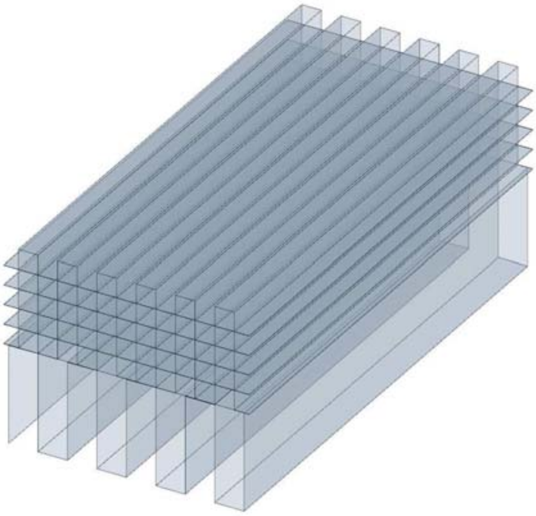

Figure 2. Honeycomb and fin heat exchangers

The honeycombs can have, for example, 400 cells per square inch (400 cpsi) with a wall thickness of 0.05 mm or 0.1 mm.

They are made by an automated system. They are rigid and light with a large surface area. The aim was to determine whether they could replace a fin heat transfer system.

The initial study was carried out using fin heat exchange theory. However, the geometric complexity of the honeycomb meant that the results had to be verified using finite element models. 2D and 3D modeling was carried out using COMSOL Multiphysics 4.1.

2. Fin theory modeling

For straight fin modeling (figure 3) see reference [3] pages 122 and 418.

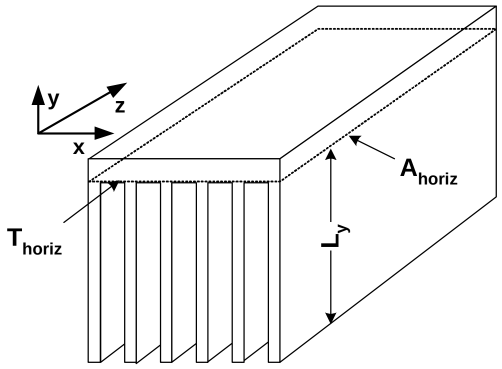

Figure 3. Straight fin schematic

Heat is exchanged between the fluid and the fins with a heat transfer coefficient of h. The fins have an efficiency η, defined as the ratio between the real heat transfer rate into the fin and the heat transfer rate that would occur if the surface temperature of the fin were equal to the temperature at the horizontal exchange surface, T_{horiz}.

The total heat transfer rate through the horizontal exchange area, q_{horiz}, reads

$$q_{horiz} = h\ F\ A_{horiz}\ (T_{fluid} - T_{horiz}) \quad (1)$$

$$\text{where } F = (D_x + 2\ \eta\ \ L_y)\ /(D_x + E_x) \quad (2)$$

$$\eta = \frac{\tanh(mL_y)}{mL_y} \quad (3)$$

$$m = \sqrt{\frac{2h}{k_s E_x}} \quad (4)$$

A honeycomb behaves like a set of vertical fins, one of which is shown in gray in figure 4. The vertical fins have horizontal half-fins, also shown in gray.

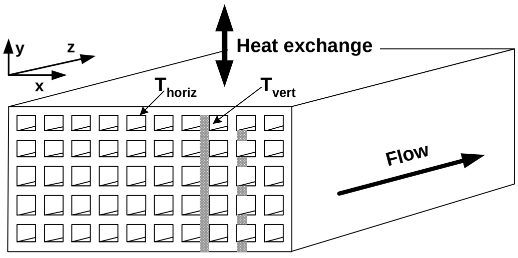

Figure 4. Heat exchange in a honeycomb

Equations 1 to 4 may be applied to the vertical surfaces assuming that these surfaces are at a uniform temperature, T_{vert}.

The heat transfer to the vertical surfaces no longer occurs at the coefficient h previously used but at a coefficient h'F' where F'can be calculated as above. h' is the heat exchange coefficient of the fluid in the channels in the honeycomb.

The heat transfer rate through each side of the vertical wall, q_{vert}, reads

$$q_{vert} = h'\ F'\ A_{vert}\ (T_{fluid} - T_{vert}) \quad (5)$$

$$F' = (D_y + \eta'\ D_x)\ /(D_y + E_y) \quad (6)$$

$$m' = \sqrt{\frac{2h'}{k_s E_y}} \quad (7)$$

$$\eta' = \frac{\tanh(0.5m' D_x)}{0.5m' D_x} \quad (8)$$

For the heat transfer rate through the horizontal wall, equation 4 is replaced by equation 9 and the heat transfer rate equation 1 is replaced by equation 10.

$$m \ \ = \sqrt{\frac{2h' F'}{k_s E_x}} \quad (9)$$

$$q_{horiz} = h'\ F''\ A_{horiz}\ (T_{fluid} - T_{horiz}) \quad (10)$$

$$\mathbf{F'' =} (D_x + 2\ \eta\ F'\ L_y)\ /(D_x + E_x) \quad (11)$$

Radiation occurs at high temperatures. The contribution of radiation to be added to the heat transfer by thermal conductivity can be determined by assuming that the honeycomb is a porous material [4].

$$k_{ray} = 4\varepsilon\sigma D_y T^3 \quad (12)$$

3. 2D Modeling using COMSOL

The coefficient of exchange between the fluid and the wall inside the channels was h' as defined above. The model was defined as conduction in a solid with channels.

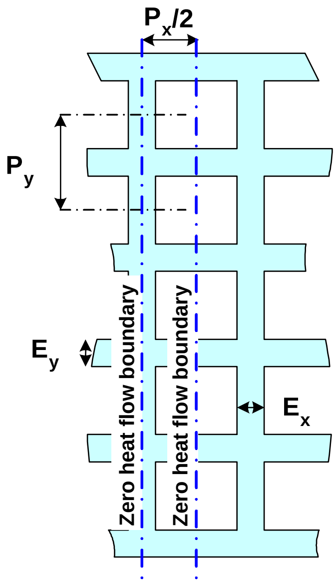

Figure 5. 2D geometry

Edge effects are ignored assuming that the dimensions are large. Ignoring radiation, the model then has 2 zero heat flow boundaries as shown in figure 5. Taking account of radiation, the geometry is based on whole channels. COMSOL 4.1 was used to parameterize the geometry.

The fluid was taken to be similar to air at 900°C and the temperature on the upper surface was defined as 800°C. The parameters for the calculations were Px (= Py), Ex (= Ey) and the number of channels. The value used for h' in COMSOL was calculated externally using laminar flow depending on the cross section of the channels. The walls were assumed to be black bodies.

The geometry f (Table 1) represents a straight fin heat exchanger.

Without radiation, the fin derived model and COMSOL both gave similar heat transfer rates (Table 1). The fin derived model may overestimate by 20%, which could be considered acceptable.

Table 1: Comparison between COMSOL 2D and fin derived models without radiative heat transfer.

Geometry	a	b	c	d	e	f
Number of channels	10	10	5	10	5	1
Px (mm)	1.30	1.25	1.30	1.30	1.30	2.3
Py (mm)	1.30	1.30	1.30	1.30	2.60	12.0
Ex=Ey (mm)	0.1	0.05	0.1	0.2	0.1	1
Coefficient h (W/m2/K)	191	191	191	208	164	189
Specific heat transfer rate (kW/m^2) COMSOL	95	74	95	132	79	112
Specific heat transfer rate (kW/m^2) fin model	108	86	108	144	89	116

The simplified model which treats the honeycomb as a porous material tends to underestimate the radiative heat transfer by comparison with the COMSOL model.

The geometry f is a straight fin heat exchanger. In this case, the simplified model treats the radiative heat transfer as occurring between two planes, one of them being at the same temperature as the fluid. COMSOL, however, showed that the radiative heat transfer was less because the solid angle for propagation was small.

Table 2: Increase in heat transfer rate due to radiation.

Geometry	a	c	e	f
COMSOL	26%	7%	10%	4%
simplified model	3%	6%	7%	26%

4. 3D modeling

The problem was studied for geometries a and f. In both cases the fluid entered at 900°C and the upper heat exchanger surface was at 800°C. For 10 channels (geometry a) the input speed was 2.7 m/s. To have the same fluid flow for each unit cross section the speed was increased to 4.8 m/s at the input for geometry f.

Various heat exchanger lengths were tested.
The flow was calculated using the Navier-Stokes equations.

The heat transfer in the system was conduction only. Radiative heat transfer was not taken into account.

$$\nabla.(k_s \nabla T) = 0 \quad (13)$$

For the fluid, the viscous stress was ignored.

$$\rho C \mathbf{u}.\nabla T = \nabla.(k_f \nabla T) \quad (14)$$

The laminar flow was calculated using the Navier-Stokes equations. There were no body forces.

$$\rho(\mathbf{u}.\nabla)\mathbf{u} = \nabla.\left[-p\mathbf{I} + \mu(\nabla\mathbf{u} + (\nabla\mathbf{u})^T) - \frac{2}{3}\mu(\nabla.\mathbf{u})\mathbf{I}\right] \quad (15)$$

$$\nabla.(\rho\mathbf{u}) = 0 \quad (16)$$

All the external boundaries of the mass were insulated except the upper surface where the temperature was set to 800°C. The side walls were planes where the flow was also assumed to be zero.

The fluid inlet was defined with a temperature set to 900°C and a mean speed with laminar flow conditions. The outlet pressure was defined (pressure, no viscous stress). There was a no slip condition on the walls in contact with the fluid and a symmetry condition on the left-hand plane on the following figure.

5. 3D geometry

The geometry was defined initially as shown in figure 6.

COMSOL gave a solution but this had an incoherent energy balance: the heat flow through the upper surface was not the same as the heat loss between the inlet and the outlet.

This geometrical representation gave problems by the proximity of a Dirichlet boundary condition and a Neuman boundary condition. At the inlet, the surface of the mass was subject to a zero heat transfer whereas the fluid had a defined temperature

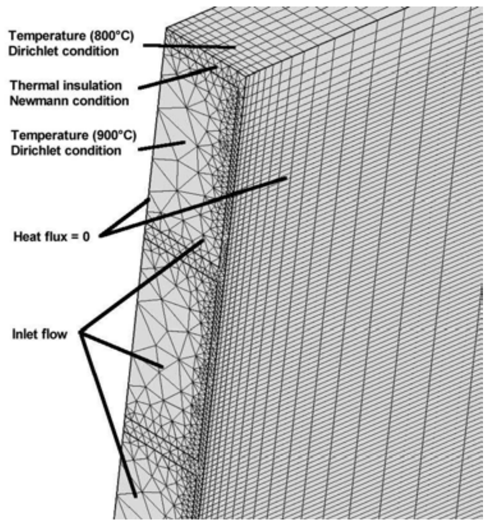

Figure 6. First 3D geometrical representation

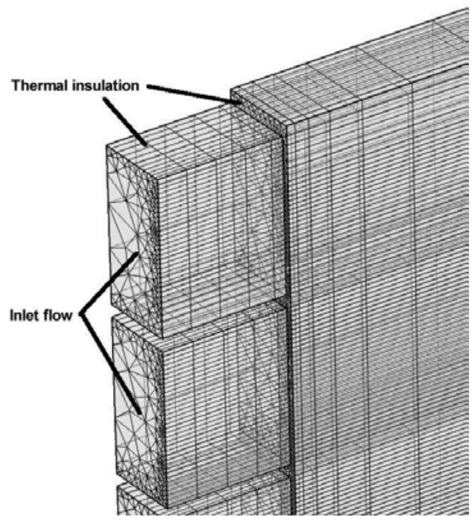

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Figure 7. Corrected geometrical representation

An overhang was therefore added to the fluid inlet.

The results from this geometry all meet the requirements for the conservation of energy.

6. 3D COMSOL results

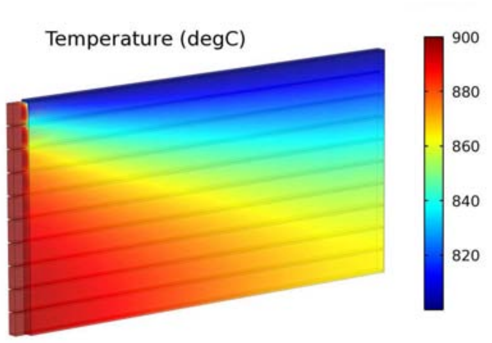

Figure 8. Surface temperature for geometry a

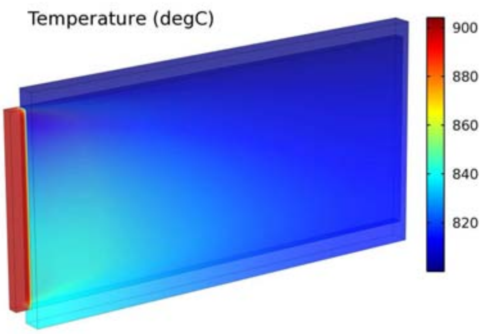

Figure 9. Surface temperature for geometry f

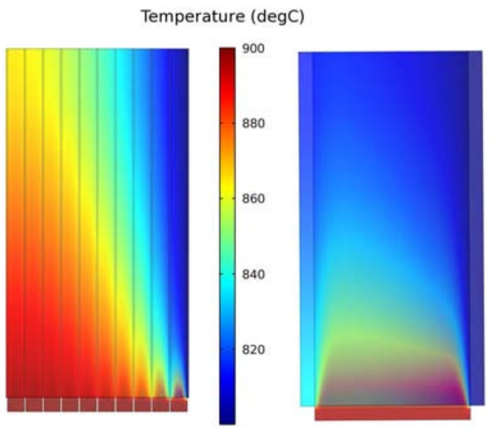

Figure 10. Surface temperature comparison between geometries a and f

There were significant differences between the use of the formulae based on the fin derived model and the 3D COMSOL model.

The fin derived model equations are based on a global temperature as for the 2D model. Figures 8, 9 and 10 show that there is a temperature gradient along the flow.

The honeycomb prevents mixing but this also occurs with geometry f where the laminar flow also restricts mixing.

This temperature gradient also occurs in the fins and half-fins and hampers conductive heat transfer in thin walls, which is the case for the honeycombs. The temperature gradient is reduced with thick fins which provide better thermal conduction.

Table 3: Comparison between COMSOL 3D and fin derived models without radiative heat transfer.

Number of channels	10	10	10		1
Length of flow	50	25	10		25
Px (mm)	1.30	1.3	1.30		2.3
Py (mm)	1.30	1.30	1.30		12.0
Ex=Ey (mm)	0.1	0.1	0.1		1.0
Specific heat transfer rate (kW/m^2) COMSOL	16	24	38		36
Specific heat transfer rate (kW/m^2) fin model	54	71	86		50

7. Conclusions

3D modeling using COMSOL threw light on heat transfer within a metal fin honeycomb along the flow.

For radiative heat transfer, it appeared that using fins did not produce greater heat transfer than using a honeycomb, given the small solid radiation angle.

Laminar flow gave a temperature gradient that was worse in honeycomb than in fins and this difference was greater for the thinner honeycomb walls.

Honeycomb heat exchangers will not give a uniform fluid temperature in the direction of the heat transfer.

If fins are used, turbulence promoters should be used between the fins to provide mixing within the fluid and thus increase the efficiency of the fins.

9. References

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10. Acknowledgements

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11. Symbols used

D_x D_y	channel dimensions or distance between fins
E_x E_y	wall thickness
P_x P_y	pitch
L_y	fin length
k	thermal conductivity
k_{ray}	radiative part of conductivity
T_{horiz}	uniform temperature at the horizontal plane
T_{vert}	uniform temperature at the vertical plane of secondary fins
T_{fluid}	global temperature of fluid
h h’	convection heat transfer coefficient
u	velocity
ε	emissivity
σ	Stefan-Boltzmann constant
ρ	density

Subscripts

$_f$	fluid flow
$_{horiz}$	horizontal plane at base of fins
$_{vert}$	vertical plane at base of half-fins
$_s$	solid
$_{x\ y}$	axes