ENHANCED HEAT TRANSFER IN OSCILLATORY FLOWS WITHIN MULTIPLE-HOLE BAFFLED TUBES

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ABSTRACT

Compound enhancement techniques are considered to be the forefront of heat transfer enhancement. In this work, the combination of active and passive techniques in low-Reynolds number tube flows is explored by means of the superposition of a fully-reversing oscillatory flow into a baffled tube. This arrangement has been employed during the last twenty-years in the so called 'oscillatory baffled reactors', focusing on the achievement of plug flow. Little work has been done, however, on the experimental and numerical analysis of the enhanced convective heat transfer that follows the resulting chaotic flow. A standard single-hole baffle geometry has been previously characterized experimentally by some authors, whereas the potential of multiple-hole baffles has not been studied from the point of view of enhanced heat transfer.

A numerical investigation has been undertaken to examine the heat transfer augmentation in multiple-hole baffled tubes with fully-reversing oscillatory flow. Different circular baffles with 1, 3, 7, 19 and 43 holes are analyzed, all of them releasing the same total cross sectional area. The flow across these baffles generates a beam of jets which extend downstream and upstream –according to the reversing flow - showing different swirl structures that promote intensive radial mixing and early onset of turbulence. As a consequence, heat transfer between the fluid and the tube wall is significantly enhanced.

A circular tube of 25 mm inner diameter has been modeled with 10 baffles uniformly spaced. The simultaneously hydrodynamic and thermal developing flow has been simulated with uniform heat flux as boundary condition in the tube wall, using water as working fluid. The achievement of spatial and time periodicity is thoroughly analyzed prior to the data reduction for the computation of Nusselt number.

The time-resolved and time-averaged heat transfer characteristics are presented for an oscillating frequency ranging from f=0.1 Hz to f=1Hz and oscillating amplitudes of $x_0=d$, $2d/3$ and $d/3$ (where d is the inner hole diameter for each baffle). The strong dependency of Nusselt number on the operating parameters of the oscillations is reported. Besides, the positive influence of an increasing number of baffle holes is demonstrated, and a description of the flow structures that induce this heat transfer augmentation is discussed.

INTRODUCTION

Oscillatory baffled reactors (OBRs) are a form of plug flow reactors, ideal for performing long reactions in continuous mode, as the mixing is independent of the net flow rate [1]. In tubular reactors, a high superficial velocity is required to obtain good mixing, but OBRs can provide plug flow behaviour at extremely low net flow Reynolds numbers. The interaction between the oscillating fluid and the sharp edge of the baffles produces this plug flow nature and creates a series of well-mixed volumes. This effect decouples the achievement of plug flow from the net flow velocity. This also makes OBRs a feasible option for converting long residence time batch processes to continuous ones, where the conventional designs have an impractical length-to-diameter ratio [2].

For the last ten years, mesoscale (millimetres) OBRs are being developed for laboratory processes. These are designed to scale up to industrial scale directly, or to be used as small-scale production platforms [3, 4, 5]. This type of OBRs have also been demonstrated to reduce feedstock materials and waste when screening processes due to its small volume [6, 7]. The different mesoscale baffle designs reported in the literature employ several types of diaphragm geometries, for example integral baffles, helical baffles, axial circular baffles, multiorifice plate baffles that can be easily fabricated at mesoscales and can be operated at low flow rates, whereas conventional OBRs designs cannot. Each baffle design have different advantages, so each design have different applications. The integral and central baffles behave in a similar way to conventional oscillatory baffled reactors. These designs are suitable for rapid screening operating conditions at laboratory scale. Since the maximum shear is reduced by the lack of sharp edges, the integral baffle design is more suitable for shear-sensitive applications [8]. Helical baffles are better for solid-liquid reactions. The absence of pronounced constrictions reduces the dead zones where particles can become lodged.

The fluid mechanics and heat transfer enhancement in a small helically baffled reactor when a low net flow Reynolds number is superimposed was analysed at [9]. Results show an improvement by a factor of 4 compared to a smooth tube in the tested range of oscillating conditions. This kind of compound enhancement techniques are considered to be the forefront of heat transfer enhancement [10].

The multiorifice plate was initially conceived for the scaling up of OBRs [11]. The ease of manufacturing and the ability to reproduce the fluid mechanics and axial dispersion at laboratory scale make multi-orifice designs particularly attractive. A 150 mm diameter OBR with 37-orifice baffles proved to show similar axial dispersion than a 24 mm diameter 1-orifice OBR [12]. Nogueira et al. [13] characterized experimentally and numerically an oscillatory flow within a baffled tube containing tri-orifice baffles. Complex eddy motion at moderate oscillatory Reynolds numbers was reproduced with PIV experiments and simulation results, suggesting a uniform global mixing throughout the domain.

A previous experimental work [14] showed that the 1-central orifice design results in an important heat transfer enhancement. The greatest enhancement is obtained when both baffles and oscillations are present. The enhancement strongly depends on the product of frequency and amplitude (oscillatory Reynolds number).

In this work, heat transfer in different multiorifice circular baffles with 1, 3, 7, 19 and 43 holes are analyzed by means of numerical techniques using the commercial software Fluent. The time-resolved and time-averaged heat transfer characteristics are presented for a typical range of oscillation frequencies f=0.1 Hz – 1 Hzand amplitudes x_0=d, 2d/3, d/3, where d is the diameter of the orifices (which depends on the number of orifices). The scope of the paper is to provide a comprehensive analysis of the influence of these two parameters in the heat transfer enhancement.

NOMENCLATURE

A	$[m^2]$	Open cross sectional area
c_p	$[J/kgK]$	Fluid specific heat
D	$[m]$	Inner tube diameter
d	$[m]$	Inner hole diameter
e	$[m]$	Baffle thickness
f	$[Hz]$	Oscillation frequency
k	$[W/mK]$	Thermal conductivity
$q"$	$[W/m^2]$	Wall heat flux
T	$[K]$	Temperature
t	$[s]$	Time
v	$[m/s]$	Velocity magnitude
x, y, z	$[m]$	Cartesian coordinates
$x0$	$[m]$	Oscillation amplitude
Greek characters		
μ	$[kg/ms]$	Dynamic viscosity
ρ	$[kg/m^3]$	Fluid density
τ	$[s]$	Period
Γ	$[-]$	Boundary of the computational domain
Ω	$[-]$	Volume of the computational domain
Subscripts		
a		Area-averaged over a cross-section
f		Fluid
net		Net flow
x		Axial
Dimensionless groups		
Re_n		Net-flow Reynolds number, $\rho v_{net} D/\mu$
Pr		Prandtl number, $c_p \mu/k$
Br		Brinkman number, $\mu v^2/[k(T_{wall} - T_{bulk})]$
Gr		Graetz number, $Re_n Pr D/x$

NUMERICAL METHOD

Geometry and mesh

The reactor under study consists of a smooth duct of circular cross section with a diameter $D = 25\ mm$ inside which a series of circular baffles with thickness $e = 1.5\ mm$ are located. These baffles consist on multiple hole disks with diameter d that are separated equidistantly a distance of $l = 3d$.

The diameter of the holes, d, is such that the fluid total cross sectional area in each kind of baffle is the same independently from the number of holes. Since the distance between baffles is defined as $l = 3d$, the more the number of holes the lower the distance between baffles. This is to ensure the correct propagation of vortex between baffles.

The geometrical parameters for the different baffles are summarized in Table 1. The 3,7, 19 and 43 holes devices have been designed taking into account the cross sectional area of the annulus baffle with $d/D = 0.5$. The designed geometries are depicted in Figure 1.

# HOLES	$d(mm)$	$l(mm)$
1	12.5	37.5
3	7.2	21.6
7	4.7	14.2
19	2.9	8.6
43	1.9	5.7

Table 1. Geometrical parameters of selected baffles

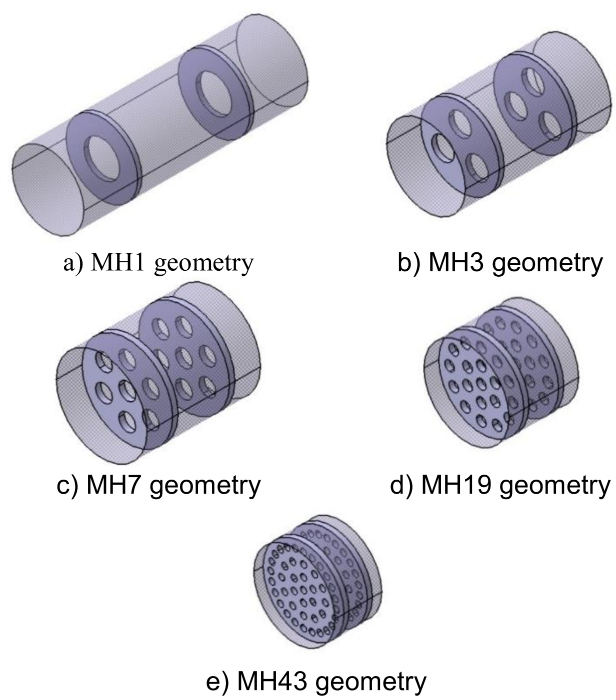

Figure 1. Geometries designed for this study

Numerical simulations

Numerical simulations were performed using the CFD package ANSYS-Fluent v14, which applies the finite volume discretization to the computational domain. To carry out the three-dimensional and unsteady numerical simulations of the flow, the set of standard incompressible mass-conservation, Navier-Stokes and energy equations were considered:

$$\nabla \cdot (\rho \vec{v}) = 0 \tag{1}$$

$$\rho \left(\frac{\partial \vec{v}}{\partial t} + \vec{v} \cdot \nabla \vec{v} \right) = -\nabla \mathrm{p} + \mu \nabla^2 \vec{v} \tag{2}$$

$$\frac{\partial T}{\partial t} + \vec{v} \nabla T = \frac{k}{\rho c_p} \Delta T \tag{3}$$

Since the Brinkman number had a maximum value of $Br \sim 10^{-2}$ for the test cases selected, the viscous dissipation term is negligible in the energy equation.

The pressure-based solver and second-order implicit formulation have been chosen to study the flow behavior. SIMPLE and second-order upwind schemes have been used in order to discretize pressure-velocity coupling, momentum and energy equations. The inlet velocity in a baffled tube with a net flow rate and a superimposed oscillatory motion is governed by:

$$v = v_{net} + 2\pi f x_0 \cdot \sin(2\pi f t) \tag{4}$$

Uniform heat flux simulations were carried out in order to solve the heat transfer problem between the fluid and the tube wall. This problem involved simultaneous solution of equations (1) and (2) together with equation (3). Instead of solving the problem with periodic flow boundary conditions, the computation of the asymptotic Nusselt number requires the analysis of the thermal boundary development. In order to achieve the fully development conditions, a computational domain of 10 tube-cells $(10 \times l)$ was solved. Velocity-inlet boundary condition given by equation (4) was imposed with the aid of a user-defined function and a uniform temperature profile at T=300 K was considered. Zero-diffusion flux and constant gauge static pressure of 0 Pa was prescribed at the tube outlet. A geometry-dependent uniform heat flux was imposed in the tube wall to drive an overall temperature difference of the fluid between the tube inlet and the outlet of about 15 K.

Pure forced convection was considered. The working fluid chosen was water with temperature-dependent properties. Third and fourth order polynomial regressions of the thermal properties of water reported in [15] were implemented. The values of thermal properties for and inlet at 300 K were: $\rho = 996.6\ kg/m^3$, $\mu = 8.5 \cdot 10^{-4} kg/m \cdot s$, $k = 0.613\ W/m \cdot K$, $c_p = 4179\ J/kg \cdot K$. The resulting Prandtl number at $T = 300\ K$ is $Pr = 5.8$.

Peripherally averaged instantaneous Nusselt number is computed in each section of the computational domain as:

$$Nu(t) = \frac{q"}{T_{wall}(t) - T_{bulk}(t)} \cdot \frac{D}{k} \tag{5}$$

where $T_{wall}(t)$ is the instantaneous peripherally averaged wall temperature in a given tube section

$$T_{wall}(t) = \frac{1}{2\pi} \int_0^{2\pi} T(\theta, t) d\theta; \quad T \in \Gamma_{wall} \tag{6}$$

and $T_{bulk}(t)$ is the mass-weighted average temperature of the fluid in the tube cross-section

$$T_{bulk}(t) = \frac{\int_A T(z, y, t) \rho \vec{v} d\vec{\sigma}}{\int_A \rho \vec{v} d\vec{\sigma}}; \quad T \in \Omega_{fluid} \tag{7}$$

Test cases

A net flow Reynolds number $Re_{net} = 5$ has been set for all the cases. This value of net Reynolds number along with the reactor diameter and fluid properties means a net flow rate of $v_{net} = 0.00017\ m/s$.

In order to study the influence of oscillation frequency and amplitude in heat transfer, several cases have been undertaken. Frequency values of $f = 0.1, 0.25, 0.5, 0.75\ and\ 1\ Hz$ and oscillation amplitude of $x_0 = d, 2d/3\ and\ d/3$ have been set.

Validation of the method

A convergence study of the numerical method has been accomplished by considering a global refinement approach. This has been undertaken with MH1 geometry, $f = 1\ Hz$ and $x_0 = d$. Table 2 summarizes the dimensions of the five meshes employed for the analysis. Temperature, static pressure and axial velocity in the measurement section were tested for meshes #1, #2, #3 and #4 and the relative error with respect to the solution in mesh #5 was computed according to:

$$error(\%) = \left| \frac{\phi_i - \phi_5}{\phi_5} \right| \times 100, \ \ i = 1,2,3,4 \tag{8}$$

where ϕ is the physical quantity evaluated. Average values of the error of each quantity in the three points tested are also reported in Table 2.

Mesh	**#elements**	**Temperature error (%)**	**Static pressure error (%)**	**Axial velocity error (%)**
1	836	22	9.6	11
2	985	12	6.4	7.1
3	1054	4.8	2.3	2.7
4	**1141**	**0.75**	**0.56**	**0.82**
5	1253	-	-	-

Table 2. Errors obtained in the grid refinement analysis

In this case, mesh #4 was considered. Average errors of 0.75% for temperature, 0.56% for static pressure and 0.82% for axial velocity have been computed. This meshing configuration was employed for subsequent numerical simulations of the baffled tube.

RESULTS AND DISCUSSION

Time-resolved and time-averaged heat transfer

The steady-state solution for $Re_n = 5$ and $Pr = 5.8$ is considered as initial condition for the oscillatory flow problem. The Table 3 summarizes the heat transfer augmentation when the multi-hole baffles are inserted in a smooth tube under fully-developed pure forced convection conditions.

Geom	Smooth	MH1	MH3	MH7	MH19	MH43
Nu	4.36	6.16	8.23	10.35	15.61	25.19
Nu/Nu_s	1	1.41	1.89	2.37	3.58	5.77

Table 3. Steady-state Nusselt number for all the geometries.

To ensure a thermally fully developed conditions, the test section selected for each case accomplished a Graetz number below 1000.

For each operating condition, the computation of the unsteady heat transfer is continuously monitored in order to identify the observance of time-periodic conditions: at each time-step, the asymptotic Nusselt number is compared with the solution obtained in the equivalent time-step of the previous oscillating cycle. The solution is considered time-periodic when the difference between all the solutions in a given cycle and the solutions in the previous cycle are below 0.5%. As shown in Figure 2, the transient phenomena preceding the fulfillment of time-periodic heat transfer depends on the operating conditions, and it is in general higher for increasing values of frequency.

A closer view of the instantaneous Nusselt number over two oscillation cycles for the test case MH43 at f=0.1 Hz is depicted in Figure 3. Since heat transfer coefficients in oscillatory motion might be influenced by the changes in the fluid bulk temperature, an insight to this effect must be carried out. Figure 4Figure 3 shows that wall temperature fluctuations are very small compared to fluid temperature fluctuations so, for practical purposes, wall temperature can be considered constant during the time-periodic heat transfer problem. The mass averaged fluid temperature decreases during the forward semicycle as the colder fluid downstream is moved due to the higher fluid velocity. Conversely, during the backwards semicycle, hotter fluid upstream comes back. Comparing this behaviour with the instantaneous Nusselt allows us to state that the Nusselt number (Eq. 5) is clearly driven by the fluid temperature fluctuations.

In order to summarize all the results in this study, Figure 5 shows the value of time-mean Nusselt number, as defined in Eq. 9, for all the frequencies and amplitudes established previously.

$$\overline{Nu} = \frac{\overline{q"}}{\overline{T_{wall}} - \overline{T_{bulk}}}\frac{d}{k} \quad (9)$$

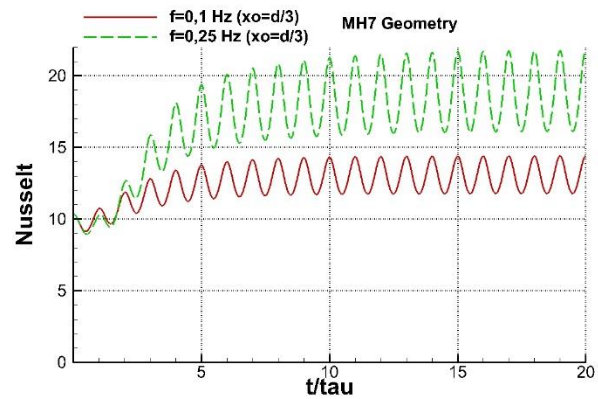

Figure 2. Temporal evolution of asymptotic Nusselt number

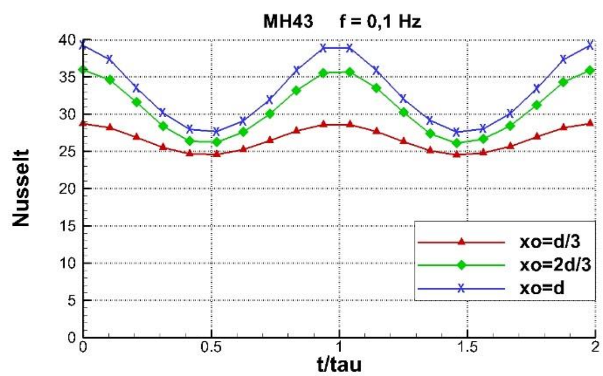

Figure 3. Time periodic Nusselt number evolution for different oscillating amplitudes. MH43, f=0.1 Hz

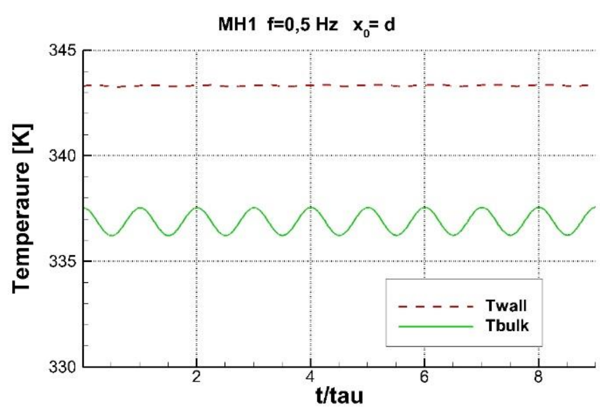

Figure 4. Wall and fluid bulk temperature time-periodic evolution

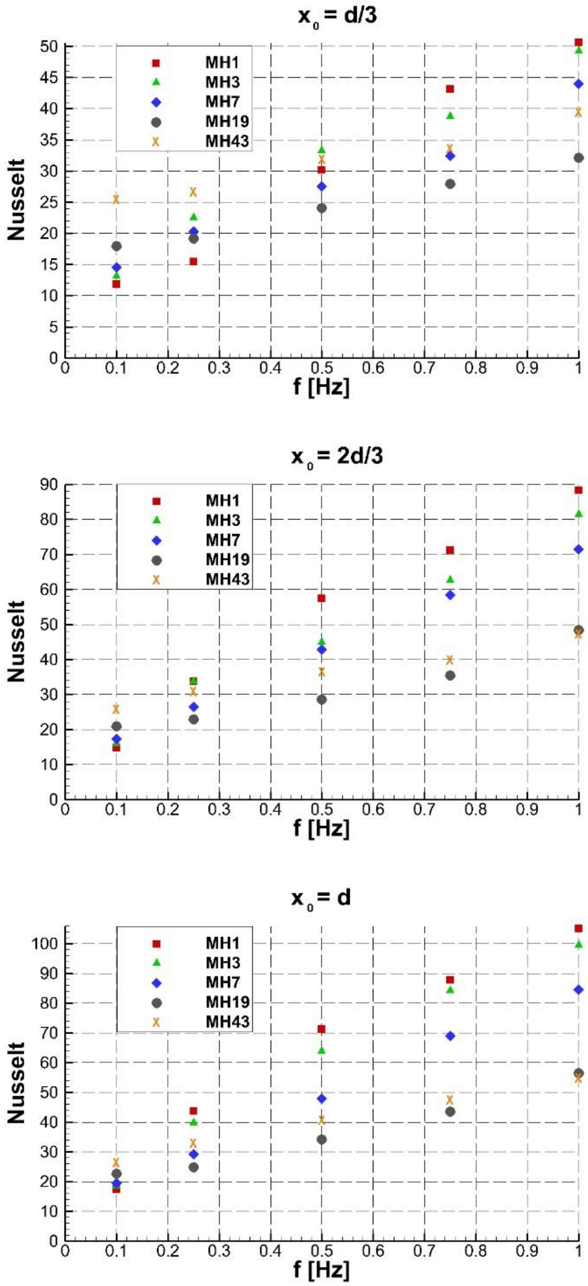

Figure 5. Time-mean Nusselt number for all the test-cases

Results show that for $f = 0.1\ Hz$ a higher number of holes provides better heat enhancement. When increasing the value of f, this trend changes for all x_0 values, i.e., a lower number of holes provides a better heat transfer enhancement. Since a lower number of holes implies a higher value of x_0 (see Table 1), this effect confirms an important influence of the oscillation amplitude magnitude in the heat transfer behaviour.

CONCLUSION

This work has studied the thermal behaviour of five different multi-orifice oscillatory baffled reactor. The main conclusions that can be extracted from the results are:

1. A higher value of frequency requires more cycles to reach the time-periodic solution.
2. Fully developed, time-periodic heat transfer is driven by the cold and hot flows that travels with the oscillatory motion, and by flow structures that promote radial mixing.
3. A huge influence of the x_0 magnitude in the heat transfer behaviour have been detected.
4. Maximum heat transfer augmentation of 20 fold have been obtained for the MH1 geometry under $x_0 = d\ and\ f = 1\ Hz$ conditions, compared with the smooth tube.

Future works will undertake a complete range of feasible flow conditions. The aim is to obtain a general correlation between the number of orifices, flow conditions and heat transfer enhancement. Also, several working fluids will be analysed to check the influence of the thermophysical properties.

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