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Laminar flow and heat transfer in U-bends: The effect of secondary flows in ducts with partial and full curvature

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ABSTRACT

The present study numerically explores the flow and heat transfer in "U-type" curved pipes of either partial or full curvature, which are often used as integral parts of heat exchangers and heat sinks in industrial applications such as solar thermal systems, among many others. Specifically, the case of a high-Prandtl-number Newtonian fluid (thermal oil) in laminar forced convection inside partially and fully curved "U-bends" is investigated, in order to illustrate the effect of the generated secondary flows on the various local and global scales. The 3D steady-state simulations are performed using the free and open source computational fluid dynamics software "OpenFOAM". The analysis of the computational results is initially based on data at selected characteristic locations of interest; they are properly established and further explored by extracting internal field variables, in the form of 2D contours of quantities of interest, such as temperature and vorticity. A parametric study on the flow rate and curvature impact, namely in terms of increasing *Re* and *De* numbers (Re = 100, 1000, 2000), on the heat transfer process inside the U-bends is conducted comparatively for both geometries. The presence of secondary flows is confirmed and the differences between the geometries under investigation are visually illustrated. For moderate and high *Re* numbers, the presence of multiple counter-rotating secondary cells is generally observed, while the partially curved U-bend ("composite") displays a more complex flow topology. This behavior is projected on heat transfer through the examination of temperature distribution on inner and outer arcs of both geometries. It is shown that the composite bend causes an abrupt decrease and oscillations on temperature distributions, which are found to be related with phenomena like flow impingement, separation and re-attachment.

The *Pr* number effect is illustrated through a comparison with a lower *Pr* fluid and its impact is demonstrated both inside and downstream of the U-bends. The averaged Nusselt distribution along the length of both curved ducts further highlights the impact of these phenomena on local heat transfer and provides initial evidence in favor of the composite bend. The overall performance of the investigated curved geometries is compared based on a "performance factor" criterion and a superior performance of the composite U-bend is further established, attaining a value of almost 45% for higher flow rates. Finally, the effect of each U-bend on the straight pipe downstream of their respective exit is quantified. High divergence from the regular "combined hydrodynamic and thermal entrance flow" is found for the forced convection inside the downstream pipe. A characteristic overshoot of the downstream-to-upstream ratio Nusselt number is found for moderate and higher flow rates at small lengths from entrance (z*). The partially curved U-bend downstream effect seems to be greater for the investigated cases of Re = 100, 2000, while the fully curved outperforms the composite for the moderate *Re* case (Re = 1000). The source of this discrepancy cannot be clearly identified through this work and further investigation is needed on this matter. All in all, the simulations performed here show that partially curved U-bends can potentially be advantageous, in terms of performance, compared to the standard fully-curved pipes for laminar forced convection applications. These results can provide a good starting point for the optimal design of such ducts for several heat transfer operations.

Abbreviations: 2D, 3D, Two Dimensional, Three Dimensional; CFD, Computational Fluid Dynamics; Duct C, Composite, partially curved U-bend; Duct T, Typical, fully curved U-bend; PCM, Phase Change Material

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Nomenclature

A	Cross-section surface area [m^2]
c_p	Specific heat capacity [J/(kg*K)]
D	Diameter [m]
k	Thermal conductivity [W/(m*K)]
L	Straight part length [m]
q”	Heat flux [W/m^2]
p	Pressure [Pa]
R	Radius [m]
S_{bend}	U-bend outer side arc length [m]
T	Temperature [K]
u	Velocity (Cartesian notation) [m/s]
w	Flow axial velocity [m/s]
z	Streamwise coordinate (from entrance) [m]

Dimensionless numbers

D*	Non-dimensional diameter, $D^* = \frac{D_x}{D_p}$
De	Dean number, $De = Re*\sqrt{\delta}$
f_{Darcy}	Darcy friction factor, $\overline{f_{Darcy}} = \frac{2*D_p*\Delta P}{\rho_{fl}*S_{bend}*w_{in}}$, $f_{Darcy,\,x} = \frac{8*\tau_{wall,\,x}}{\rho_{fl}*w_{in}^2}$
L*	Straight part-to-total length ratio for composite bend, $L^* = \frac{L_1}{S_{bend}}$
Nu	Nusselt number, $Nu = \frac{q''*D_p}{k_{fl}*(\overline{T_{wall}} - \overline{T_{bulk}})}$
PF	Performance factor, $PF = \frac{\frac{\overline{Nu}_{bend}}{\overline{Nu}_{str@P1}}}{\left(\frac{\overline{f_{Darcy}}_{bend}}{\overline{f_{Darcy}}_{str}}\right)^{1/3}}$
Pr	Prandtl number, $Pr = \frac{c_{P}*\mu_{fl}}{k_{fl}}$
Re	Reynolds number, $Re = \frac{w_{in}*D_p*\rho_{fl}}{\mu_{fl}}$
S^*_{total}	non-dimensional total domain length, $S^*_{total} = \frac{S_{bend}+L_{p,up}+L_{p,down}}{S_{total}}$
T*	non-dimensional temperature, $T^* = \frac{T-T_{in}}{\Delta T_{ref}}$
W*	Non-dimensional axial velocity, $W^* = \frac{w}{u_{ref}}$
z*	Non-dimensional stream-wise coordinate (thermal entrance length),
ω^*_{normal}	Non-dimensional normal (to cross-section plane) vorticity, $\omega^*_{normal} = \frac{\omega_{normal}*D_p}{w_{in}}$

Greek symbols

δ	Pipe-to-curvature radii ratio, $\delta = \frac{R_p}{R_{curv}}$
ΔP	Pressure drop [Pa], $\Delta P = p_{in} - p_{out}$
ΔT_{ref}	Reference temperature difference [K], $\Delta T_{ref} = \frac{q''*D_p}{k_{fl}}$
μ	Dynamic viscosity [Pa*s]
ρ	Density [kg/m^3]
τ	Wall shear stress [Pa]
φ	Azimuthal angle [degrees]
ω	Vorticity [1/s], $\vec{\varpi} = \vec{\nabla} \times \vec{U}$

Subscripts

1,2,3	Identifier of straight pipe segment of the composite U-bend
bend	Curved segment referring to U-bend part only
bulk	Bulk/mixing cup fluid temperature [K], $T_{bulk} = \frac{1}{U*A}\int u_i*T\, dA$
comp	Composite U-bend
critical	Critical value of magnitude
curv	Curvature
down	Downstream pipe
fl	Fluid
H	Constant heat flux boundary condition
i,j	Indices of Cartesian tensor components
in	Inlet
out	Outlet
p	Pipe
ref	Reference value
str	Straight part
total	Total length including upstream, U-bend and downstream pipes
typ	Typical U-bend
up	Upstream pipe
x	Local value
wall	Wall

Superscripts

$\overline{\square}$	Average value
*	Dimensionless value
$\vec{\square}$	Vector

1. Introduction and literature review

The use of curved ducts in heat exchangers and heat sinks is well established in various industrial applications, including but not limited to heat absorbing units used in every solar thermal system. The reversal of the main flow direction guarantees a more compact layout while also a higher contact surface area for the heat transfer processes is achieved [1]. However, this simple geometrical modification introduces significant complexities in the heat and mass transfer process, compared to the straight pipe. Due to curvature, as fluid flows through curved pipes, centrifugal force is generated; this force, combined with the emergence of a radial pressure gradient acting on the fluid, jointly induce a secondary motion, superimposed on the main axial flow and the velocity peak is shifted towards the outer side of the duct. This motion occurs as a pair of counter-rotating longitudinal vortices, named in the open literature as “Dean cells” after the British scientist W.R. Dean (1896–1973).

Dean followed the theoretical work of Boussinesq [2] on flows in curved channels, the experimental findings of Thompson [3,4] on the analogy between river windings and bend pipes and those of Williams et al. [5] and Eustice [6,7] on water flow through curved pipes. He was the first to develop an analytic solution for the laminar flow in curved circular ducts [8] and demonstrated the significance of the dimensionless parameter $Re^2*\delta$, today called the “Dean number” in its square root form, on the development of the secondary flow. In a subsequent paper [9], Dean showed the emergence of this secondary flow and the appearance of two counter-rotating cells, when the flow is projected on a cross-stream plane. Nowadays, the flow in curved ducts is widely referred to as “Dean flow” and is extensively used in practice, especially regarding heat transfer processes, as the induced secondary motion is capable of locally enhancing the rate of heat transfer at the expense of increased hydrodynamic losses. For all these reasons, curved pipe flow is considered a very popular “passive enhancement” technique for heat transfer [10].

More specifically, various heat absorbing units used in conventional and concentrated solar thermal systems, like solar absorbers [11] and heat sinks [12], very frequently make use of curved ducts and especially their configuration as U-bends, where the flow is fully reversed. These ducts are normally composed of uniform curved parts with constant curvature radius, forming this characteristic “U-shape”. However, in