

Thermo-fluidic heat transfer analysis in U-shaped mini-channel sinks

Giorgio Moscato^{1,*}, Mohammad Moradi¹, Giovanni P. Romano¹

1: Dept. of Mechanical and Aerospace Engineering, University of Rome La Sapienza, Italy

* Correspondent author: giorgio.moscato@uniroma1.it

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ABSTRACT

The thermal and fluid dynamic performance of a U-shaped mini water-cooled heat sink featuring a semicircular bend is investigated experimentally at three Reynolds numbers ($Re \approx 320, 735$ and 1540) and a fixed heat source temperature of 50°C . Thermal characterization is performed through K-type thermocouples placed in the aluminium basement and Infrared Thermography (IRT), while velocity and vorticity fields are obtained by means of high-speed micro Particle Image Velocimetry (μPIV). A thermal transfer function is derived to relate the external Plexiglass surface temperature, measured by the infrared camera, to the temperature at the internal channel roof. Infrared images are pre-processed through a combined temporal and spatial filtering approach based on Proper Orthogonal Decomposition and an edge-preserving elliptic Gaussian kernel. Lumped system analysis is applied to both thermocouple and infrared data to extract thermal transient rate fields, which serve as a proxy for the local convective heat transfer coefficient.

Results indicate that the thermal transient rate increases with Reynolds number, reflecting a faster cooling effect at higher flow rates. Its spatial distribution is, however, non-uniform along the channel. At the lowest Reynolds number, the region of enhanced heat transfer is located in the second pass, and progressively shifts upstream, toward the first pass as the Reynolds number increases. PIV results reveal a consistent flow behaviour: as the flow approaches the semicircular bend, it detaches from the convex inner wall and is displaced toward the concave outer wall. This displacement intensifies with increasing Reynolds number. Vorticity fields show a bifurcation of two positive vorticity regions, indicative of the onset and propagation of Dean vortices. The Dean numbers computed for the three flow conditions range from approximately 152 to 725, above the critical threshold for the formation of primary counter-rotating vortex pairs. The observed heat transfer enhancement in the bend region is attributed to the additional fluid mixing induced by these coherent structures, which transport colder fluid from the channel core toward the heated wall. The spatial correspondence between the high thermal transient rate region and the area of vorticity bifurcation supports this interpretation.

1. Motivation and Background

Over the past few decades, the investigation of microfluidic devices has attracted substantial scientific interest, primarily driven by industrial demand for technological advances in devices such as electronic microchips and energy conversion systems. A critical aspect of these components is in their ability to exchange thermal energy with surrounding fluids in order to regulate operating temperatures. An everyday example is represented by smartphones or laptops. The significance of this thermal management aspect is heightened by ongoing miniaturization trends that necessitate the integration of high-power-density components, which in turn demand intensive cooling solutions to sustain operational performance. Although considerable research efforts have been devoted to this challenge, most of experimental investigations are focused on the overall thermal performance characteristics of such systems, largely due to the inherent difficulties associated with obtaining reliable measurements at such scale. In recent years, the rise of new experimental methods for the measurement of local thermo-fluidic properties has allowed for a deeper understanding of this phenomenon, as comprehensively reviewed by Morini (2019).

A number of studies have explored this topic with different approaches. Sharma & Khan (2023) conducted a numerical investigation comparing the thermo-hydraulic performance of elliptic versus sinusoidal-wavy serpentine configurations under laminar water flow conditions, for Reynolds numbers in the range 160 to 900. By maintaining constant microchannel amplitude, while systematically varying wavelengths, they identified that elliptic-wavy geometries demonstrated superior performance relative to sinusoidal counterparts across most conditions, with the exception of the highest waviness magnitude at the lowest Reynolds number. To validate their numerical results, the authors employed a micro-PIV system for the elliptical-wavy configuration at $Re=160$, whereas sinusoidal-wavy validation relied upon previously published results from Sui et al. (2011). Those researchers examined Nusselt number and friction factor behaviour in copper wavy microchannels featuring rectangular cross-sections of approximately $205 \times 404 \mu\text{m}$, with amplitudes varying between 0 and $259 \mu\text{m}$, and Reynolds numbers within 300-800. Temperature measurements were acquired within the test section using three thermocouples, with additional ones positioned at inlet and outlet locations, complemented by a differential pressure transducer. The authors demonstrated thermal superiority of wavy geometries over straight configurations, albeit at the cost of greater pressure drop in all test conditions. This finding represents a key aspect for the design and optimization of microfluidic devices. Indeed, following the formulations proposed by Promvonge (2008), Kumar et al. (2019), and Spizzichino et al. (2020), thermal performance factor (TPF) can be quantified as the ratio of mean Nusselt number to friction factor (raised to the power of 1 or $1/3$). This metric highlights the necessity of integrating fluid dynamic

aspects into the design optimization of microfluidic devices, which should not focus solely on heat transfer enhancement. To this aim, a particularly promising investigative strategy involves implementation of micro-PIV systems (Morini G.L. 2019, Sharma et al. 2023, Wereley & Meinhart 2007, Jung-Ryul Lee et al. 2025). Dai et al. (2015) used micro-PIV to study five wavy channel configurations and identified secondary flow structures (Dean vortices) as key mechanisms enhancing heat transfer in curved channels. This finding is further supported by works of other authors (Sui et al. 2011, Liu et al. 2023, Marshall et al., 2022). The formation of Dean vortices is connected to the difference in pressure distribution that arises across the bend owing to the centrifugal force that pushes the flow towards the concave wall. The onset and configuration of those vortices are characterized by the Dean number De (Saffar et al., 2023):

$$De = Re \sqrt{\frac{D_h}{2R_c}}$$

where Re is Reynolds number, D_h the hydraulic diameter and R_c the radius of channel curvature. In a single-curve channel, for a Dean number within the range 0-20, a pair of counter-rotating vortices appears across the bend, and the maximum velocity location is displaced towards the concave wall. As the Dean number increases, the strength of the vortices and the transverse velocity increase, up to a critical value ($De_c \approx 100-160$) at which secondary Dean vortices detach from the primary ones. These vortices are responsible for heat transfer improvements in microchannel devices owing to the additional mixing they provide.

However, most experimental work relies on pointwise temperature measurements via thermocouples, limiting full thermal field characterization (Abed et al. 2015, Khoshvaght-Aliabadi et al. 2016, Karale et al. 2013, Rastan H. et al. 2020). One solution to overcome this limitation was introduced by Massing et al. (2018), who used a combined approach based on astigmatism particle-tracking velocimetry (Cierpka et al., 2010b) and temperature-sensitive emission lifetime of luminescent particles (Massing et al. 2016). The authors managed this way to derive simultaneous three-dimensional velocity and temperature field measurements in a straight microchannel of square cross section $2 \times 2 \text{ mm}^2$. Another approach to measure the entire temperature field involves the use of Infrared Thermography (IRT). Mehta and Khandekar (2012) used IRT to map local wall temperature distributions in a mini-channel at Reynolds numbers 100-850, revealing high Nusselt values in the thermal entrance region and conjugate heat transfer effects. Yamada & Nakamura (2016) used a combined measurement system based on 2D-3C PIV and high-speed IRT to describe the flow behaviour and thermal fluctuations near a backward-facing step. More recently, Wang et al. (2023) successfully combined PIV and IRT to investigate heat transfer enhancement in a U-shaped square channel of hydraulic diameter equal to 45.5mm with detached curved ribs.

The present work focuses on characterizing the thermal and fluid dynamic performance of a U-shaped mini water-cooled heat sink with a rounded semicircular bend. This geometry is proposed to improve the uniformity in heat transfer distribution compared to a previously investigated design featuring two consecutive 90° bends. The U-shaped channel has been selected as a simplified unit of the serpentine heat exchanger, which is considered one of the most promising for cooling processes (Spizzichino et al. 2020). Thermal and fluid dynamic characteristics are investigated using Infrared Thermography (IRT), thermocouples and micro high-speed Particle Image Velocimetry (μ PIV) at three Reynolds numbers (approximately 320, 750, and 1540, based on hydraulic diameter) and a heat source temperature of 50 °C.

2. Experimental set-up

The experimental system is composed of an open hydraulic circuit, PIV and thermal systems. Referring to the first one, it is composed of a syringe pump Landgraaff LA-800, a 140 ml plastic syringe (placed inside an aluminium case to avoid any expansion during compression), the heated test cells and a discharge bowl (Figure.1). The pump moves the syringe's piston allowing for control of the volumetric flow rate, with an absolute error of ± 0.05 ml/min. The working fluid is demineralized water at ambient temperature, whose value is measured before each test through a thermometer XS Temp 7 NTC with a resolution of 0.1 °C. A plastic outlet tube is connected at the exit of the cell and is coated with a thermal insulation layer in neoprene, thus preventing heat losses and assuring a reliable water temperature measurement during dischargement. The investigated cell design features a semicircular bend that aims at suppressing flow recirculating zones already observed in previous experiments for the case of two consecutive 90° bends geometry (Mohamed et al. 2024, Moscato & Romano 2024) that are responsible for a non-uniform heat transfer distribution. A Blueprint of the cell design is shown in Figure.2a-b, as well as an example of assembled cell (Figure.2c). It is milled in a Plexiglass plate of 110 mm in length, 15 mm in width and 4 mm in thickness. The top and side walls of the channel are made of Plexiglass, while the bottom side is closed by a copper thin sheet placed over an aluminium basement of the same length as the Plexiglass cell and cross-section of 10x15 mm. The cell is placed over a Stuart US-150 hot plate, which acts as the heat source at a fixed temperature of 50°C. The copper sheet is sufficiently thin and soft that, when compressed between the Plexiglass and the aluminium basement, it behaves as a sealing gasket to avoid leakages, while having a negligible impact on heat conduction from the basement thanks to its high thermal conductivity (approximately 400 W/mK). Its thickness is approximately 0.2mm and it is black painted for contrast-enhancement in PIV images. Six M3 and one M2 screws are used to fix the Plexiglass cell to the base, along with

washers. They are all coated with PTFE to minimize their thermal influence on the cell. The thermal measurement system consists of four ungrounded K-type thermocouples (three of $\phi=1$ mm diameter and one of $\phi=0.5$ mm, resolution of 0.1°C , accuracy of $\pm 1.5^\circ\text{C}$, acquisition rate 1 Hz) and an infrared camera Optris PI640i (640x480 pixels at 32 Hz, thermal sensitivity 80 mK, accuracy of $\pm 2^\circ\text{C}$) equipped with the Optirs MO44 optics ($12^\circ \times 9^\circ$, $F=1.1$, $f=44$ mm), which provides a minimum spot size (IFOV) of approximately $30\text{ }\mu\text{m/px}$ in current tests. The three $\phi=1$ thermocouples are placed in the aluminium basement, at approximately 0.5 mm below the channel floor, along the geometrical centreline of the channel (as shown in Figure.2b), and are used to monitor the temperature stability before running the test and its evolution during the test, while the smallest one is positioned at the outlet tube's exit to directly measure water temperature after cooling the cell. The infrared camera is placed on top of the cell and measures its external surface temperature. Therefore, aiming to derive the internal cell thermal field at the interface between Plexiglass and the coolant (i.e. at the channel roof), a transfer calibration function is computed as in previous studies (Mohamed et al., 2024, Moscato & Romano, 2024).

PIV images are acquired through a 12-bit CMOS Photron Nova S9 camera placed on top of the cell, equipped with three Kenko extension tubes (two 36 mm and one 12 mm, respectively) and a Nikkor 50mm f/1.2 lens. The resulting spatial resolution is $10\text{ }\mu\text{m/px}$. Depending on the flow rate, the frame rate is varied between 3000 to 7200 fps, with exposure times in the range 0.05-0.1 ms. Each full acquisition consists of 87354 images, recorded with a nine-partition scheme to improve the quality of statistics (Moscato & Romano, 2023). Hollow spherical glass particles with a mean diameter of $15\text{ }\mu\text{m}$, standard deviation $10\text{ }\mu\text{m}$ and Stokes number of approximately 10^{-3} are used as seeding. Appropriate illumination is provided by two 40 W LED white lamps.

The investigated flow rates are 30 ml/min, 70 ml/min and 150 ml/min, corresponding to Reynolds numbers of approximately 320, 735, and 1540, respectively. These Reynolds numbers are based on the hydraulic diameter $D = 0.002$ m and the kinematic viscosity evaluated at the bulk water temperature. The ratio between the radius of curvature and the hydraulic diameter is approximately $R_c/D=2.25$.

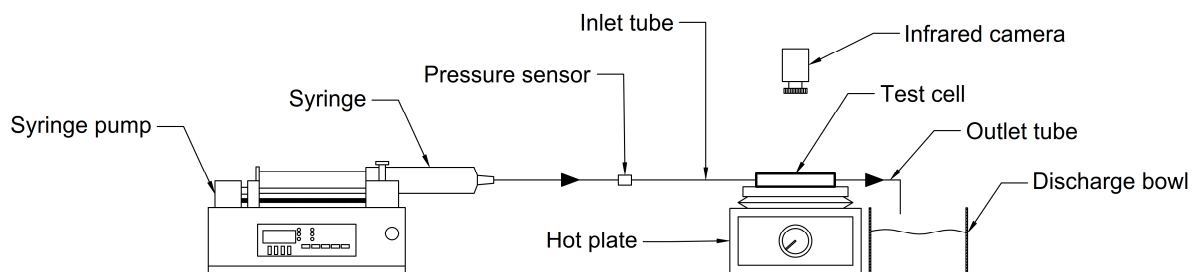


Figure.1 Sketch of the hydraulic system from syringe pump to discharge bowl. Infrared camera and high-speed camera (not reported here) are placed on top of the test cell. Infrared camera position is reported merely to illustrate the practical positioning of objects.

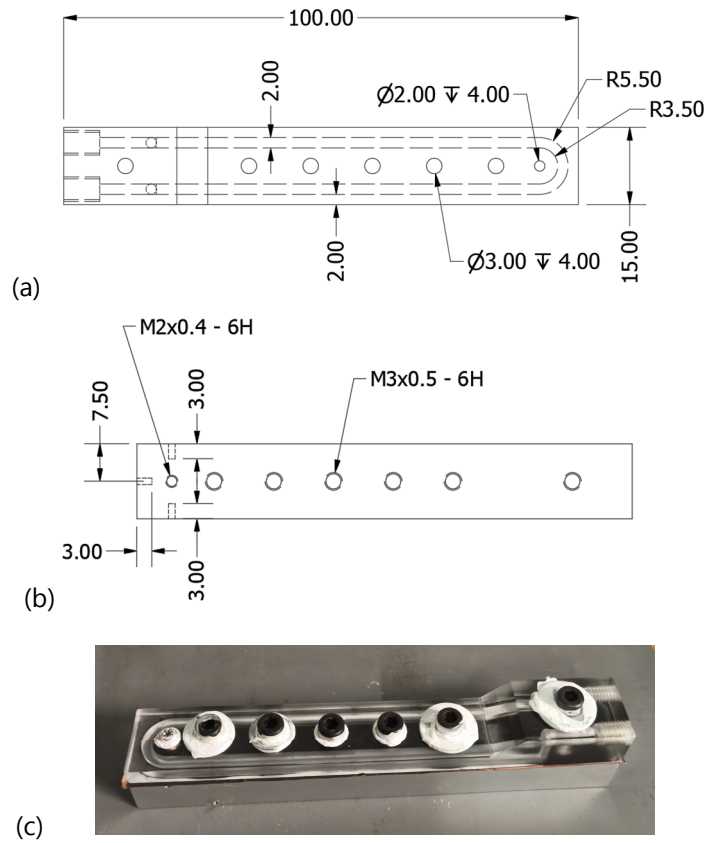


Figure.2 Blueprints of (a) cell with semicircular bend, (b) aluminium basement with holes for thermocouples and (c) example of assembled cell with semicircular bend.

3. Methods

3.1 Data preprocessing

PIV preprocessing, analysis and postprocessing are performed using PIVLAB (Thielicke & Sonntag, 2021) with multipass window deformation cross-correlation analysis. A final interrogation window size of 16x16 pixels with 50% overlap is employed, yielding approximately 21 vectors across the channel cross-section. Preprocessing consists of standard minimum background subtraction, while postprocessing is based on spatial standard deviation and local median filters for the velocity fields. Owing to the inherent noise characteristics of infrared thermography images, their preprocessing involves a combined approach employing temporal and spatial filtering. Temporal filtering is based on Proper Orthogonal Decomposition (POD), while spatial filtering utilizes a deformed Gaussian kernel. As demonstrated by Raiola et al. (2016), the POD-based approach provides robust spurious noise reduction in infrared images. Following their methodology, the number of POD modes k required for data reconstruction is determined using the scree test criterion: the number of modes is selected allowing the non-dimensional variation between two consecutive reconstruction errors based on consecutive mode numbers to remain approximately constant, i.e. 0.999. Spatial filtering

is based on the edge-preserving filter developed by Miozzi et al. (2019). Specifically, it applies an elliptic Gaussian kernel to an $n \times m$ pixel window centred on each infrared image pixel, with its major axis aligned to the local edge direction. For this study, an 11×11 pixel window is used, and the ellipticity degeneration threshold is set to 1.4. The effect of these temporal and spatial filters is shown in Figure.3a-d.

3.2 Lumped system analysis

A well-established approach to model heating and cooling processes is lumped system analysis (Cengel 2009, Incropera et al. 1996). This method assumes uniform temperature distribution within the considered body at each instant and employs three primary parameters to characterise the exponential temperature transient:

$$T(t) = T_{\infty} + \Delta T_{\infty} e^{-\frac{t}{\tau}}, \quad (1)$$

where T_{∞} is the stabilization temperature that is reached for an infinite time, ΔT_{∞} is the difference between the initial temperature T_0 and T_{∞} (representing the overall cooling magnitude) and τ is the thermal transient time constant. The validity of the uniform-temperature assumption is assessed through the Biot number, defined as $Bi = hL_c/k_s$ where h is the convection coefficient, L_c the characteristic length and k_s the solid thermal conductivity, and is fulfilled when $Bi < 0.1$.

In this study, lumped system analysis is successfully applied to both thermocouple and infrared thermography data. From a practical point of view, this analysis is applied by fitting temperature data in time following equation 1, so that an estimate of T_{∞} and τ is obtained. A typical temporal evolution of temperature in our measurements is shown in Figure.3a. Herein, it is possible to distinguish two main behaviours: an initial almost-constant region (corresponding to the condition of no water flowing through the channel) and an exponentially decreasing one. Therefore, understanding where the first region ends and the second begins is of crucial interest. To this end, an iterative fitting procedure based on equation 1 is applied to all temperature data, eroding the signal from the beginning up to 25% of its total length, for both thermocouple and infrared camera measurements. The detection of the time instant after which the temperature decay follows equation 1 is then obtained through a maximum-R-square criterion (for thermocouple data) and a T_{∞} and τ stability criterion (for infrared camera measurements).

The infrared camera records data on the external surface of the Plexiglass cell. Accordingly, the thermal transfer function is applied to those data to derive the thermal fields at the channel roof. The lumped system analysis described by equation 1 is applied to the temporal history of each pixel of the recorded infrared images. The validity of this model is assessed for a Biot number below 0.1 (Cengel, 2009), which corresponds to a condition on the maximum allowed size of a single pixel

unit $\delta < 0.1 \frac{\alpha_p}{b_{\max} D_h}$, where α_p is the Plexiglass thermal diffusivity and $b=1/\tau$ is the thermal transient rate (Mohamed et al., 2023).

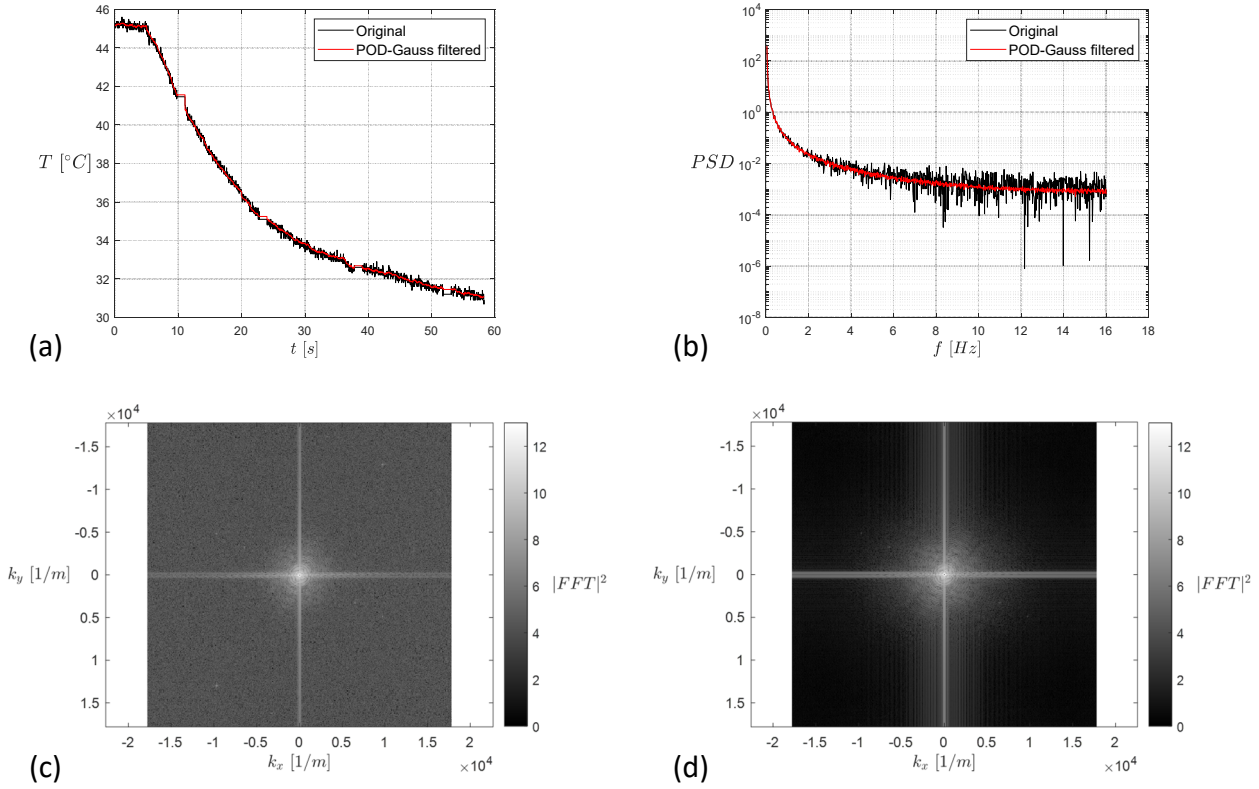


Figure.3 (a) Temporal temperature evolution for a point at middle of the semicircular bend; (b) power spectral density of the temperature signal (a). Black and red solid lines indicate original and POD-elliptic Gaussian filtered data, respectively. (c) Fourier Spectrum of an original IRT image and (d) Fourier Spectrum after both POD and elliptic Gaussian filters.

Within this dimension, the approximation uniform temperature is reliable. In the present case, the camera resolution is approximately $28\mu\text{m}/\text{px}$, about 70 times smaller than channel hydraulic diameter, which is sufficiently small to match the previous condition (i.e. $\delta < 92\mu\text{m}$) and capture the main spatial differences in temperature across the surface.

The lumped analysis highlights the strict connection between the thermal transient rate b and the convective heat transfer coefficient h (Cengel, 2009):

$$b = h \frac{A_s}{\rho V c_p}, \quad (2)$$

where A_s is the solid surface through which the heat is exchanged, V is the solid volume, ρ the solid density and c_p the specific heat at constant pressure. Therefore, the investigation of thermal transient rate fields is of primary importance as it provides indications of the most effective regions for heat transfer along the channel.

3.3 Thermal transfer function

The infrared camera measures the temperature distribution over the external surface of the Plexiglass cell. Since the no-temperature-jump condition holds at the solid–fluid interface (Cengel, 2009), the water layer in direct contact with the channel roof is assumed to be at the same temperature as the wall itself. Accordingly, a thermal transfer function is derived to relate the external surface temperature, measured by the infrared camera, to the temperature at the channel roof. To this end, a 0.5 mm K-type thermocouple was inserted through a drilled hole in the cell side wall and placed in direct contact with the channel roof at half the channel width. The corresponding location on the external Plexiglass surface was simultaneously monitored by the infrared camera. The cell was filled with demineralised water and heated to approximately 50°C. Once thermal stabilisation was reached with the fluid at rest, the hot plate was switched off and the cooling transient was recorded simultaneously by the thermocouple and the infrared camera. Following Mohamed et al. (2024), the relationship between the two temperature signals is assumed to be bilinear and independent of the working fluid and thermal process (heating or cooling):

$$T_{TC} = p_1 * T_{IRT} + p_2 \quad (3)$$

where T_{TC} is the temperature measured by the thermocouple in direct contact with the channel roof, and T_{IRT} is the corresponding temperature measured by the infrared camera on the Plexiglass external surface.

The infrared camera acquisition rate was set to 8 Hz to limit data volume while retaining sufficient temporal resolution of the transient. The recording duration exceeded one hour, ensuring that the cooling transient was fully extinguished and the final recorded temperature matched ambient conditions. In post-processing, a spatial window of 18×18 pixels was extracted from each thermal image at the thermocouple location. This window size was selected based on the camera spatial resolution of approximately 35.6 px/mm to match the thermocouple footprint. Indeed, the thermocouple diameter of 0.5 mm corresponds to roughly 16.3 pixels, so an 18×18 pixel window captures the thermocouple footprint entirely on the Plexiglass surface. Pixel values within this window were spatially averaged, yielding a single temperature value per thermal frame and reducing the effect of spatial noise. The resulting time series was then temporally averaged over consecutive 8-frame windows, further attenuating noise and matching the thermocouple acquisition rate of 1 Hz.

4. Results

4.1 Thermal transfer function

The resulting scatter plot and linear least-squares fit for the thermal transfer function are shown in Figure.4, together with a grey-shaded region representing the combined systematic uncertainty of

the thermocouple ($\pm 1.5^\circ\text{C}$) and infrared camera ($\pm 2^\circ\text{C}$). The linear fit provides an accurate representation of the data, with a coefficient of determination $R^2 \approx 0.9993$. The regression coefficients and their 95% confidence intervals are reported in Table.1.

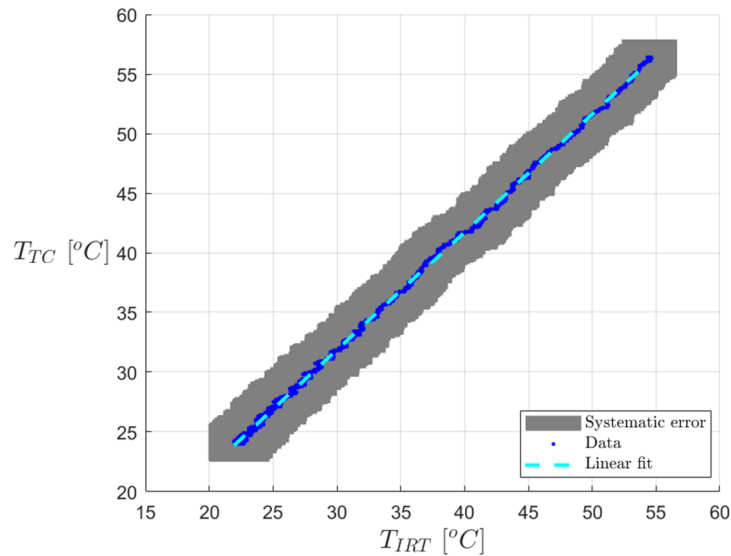


Figure.4 Scatter plot of temperature measurements acquired simultaneously by the thermocouple and the infrared camera (blue markers). The dashed cyan line represents the linear transfer function obtained by least-squares fitting. The grey-shaded area indicates the combined systematic uncertainty associated with the accuracy of the infrared camera ($\pm 2^\circ\text{C}$) and the thermocouple ($\pm 1.5^\circ\text{C}$).

Table.1 Linear fitting parameters of the thermal transfer function. The coefficients p_1 and p_2 represent the slope and intercept, respectively, of the linear equation in the form $y = p_1 \cdot x + p_2$. The 95% confidence bounds for these values are provided in parentheses.

Regression coeff.	Values	95% confidence interval
p_1	0.9896	[0.9888, 0.9904]
p_2	2.1557	[2.132, 2.179]

4.2 Thermocouples results

Tests were conducted at three Reynolds numbers, namely ≈ 320 , ≈ 735 and ≈ 1540 . These values were defined based on the hydraulic channel diameter and on fluid properties evaluated at the bulk temperature (i.e. the average between the inlet and outlet stabilisation temperature). In this Reynolds number range, the resulting flow is laminar, at least until the bend inlet. In order to determine the hydrodynamic and thermal state of the flow when approaching the bend, hydrodynamic and thermal entry lengths were computed. For a developing flow in a rectangular channel, Shah and London (1978) proposed the following empirical expression for the developing hydrodynamic length:

$$L_{h,lam} = (0.06 + 0.07\lambda - 0.04\lambda^2) ReD_h \quad (4)$$

where λ is the channel cross-section aspect ratio, equal to 1 for square ducts, D_h is the hydraulic diameter, and Re is Reynolds number. The thermal entry length can be defined as follows (Cengel, 2009):

$$L_{th,lam} \approx 0.05 Re Pr D_h \quad (5)$$

where Pr is the Prantl number.

In our device, the inlet channel has a length of approximately $46D_h$, so that a hydrodynamically fully developed flow can be reached only at the lowest Reynolds number condition, while a thermally fully developed state is never attained (Table.2). Therefore, most of tests are conducted in the simultaneously developing regime, which is of practical relevance for applications in compact mini-channel systems.

Table.2 Required hydrodynamic and thermal entry lengths for the current measurements.

	Re=320, Pr=6.3	Re=745, Pr=6.5	Re=1540, Pr=6.6
$L_{h,lam} / D_h$	≈ 29.0	≈ 66	≈ 138
$L_{th,lam} / D_h$	≈ 101	≈ 238	≈ 507

An example of non-dimensional temperature signals collected through the thermocouple placed at the middle of the bend, in the aluminium basement, below the channel floor, is shown in Figure.5 for different Reynolds numbers. Herein, it is possible to observe that the temperature is initially approximately stable at the steady initial temperature T_0 , while no water is flowing through the channel. After the syringe pump is turned on, the temperature starts to decrease and follows the exponential decay trend described in equation 1 (Cengel 2009, Incropera et al. 1996). However, owing to the finite volume of the syringe, the stabilization temperature T_∞ is clearly reached only for the lowest flow rate, at the end of the test. Nonetheless, applying the lumped system model through an iterative fitting process and selecting as the final fitting starting time the one that maximises the R-square coefficient (always greater than 0.99), it was possible to estimate T_∞ and τ for all cases. As a general outcome, these parameters are inversely proportional to the Reynolds number, thus showing lower values for higher flow rates. The microfluidic device is equipped with three thermocouples placed at 0.5 mm below the channel floor, at three "measurement stations" positioned along the channel geometric centreline: Station 1, Station 2 and Station 3. The first is located at the beginning of the bend/end of the inlet channel, the second is at the middle of the bend, and the third is positioned at the end of the bend/beginning of the outlet channel (Figure.2b). An additional thermocouple is inserted at the end of the outlet tube (Station 4), just before the discharge bowl, to directly measure the water temperature (Figure.1). A convenient way to compare

the temperature decay trends for all these thermocouples at each Reynolds number is obtained by introducing the non-dimensional parameter Θ :

$$\theta = \frac{T(t) - T_{\infty}}{T_0 - T_{\infty}}, \quad (6)$$

which represents a non-dimensional expression of the exponential part of equation 1. Trends of this parameter for the three considered flow rates are shown in Figure.6a (Station 1), Figure.6b (Station 2), Figure.6c (Station 3) and Figure.6d (Station 4).

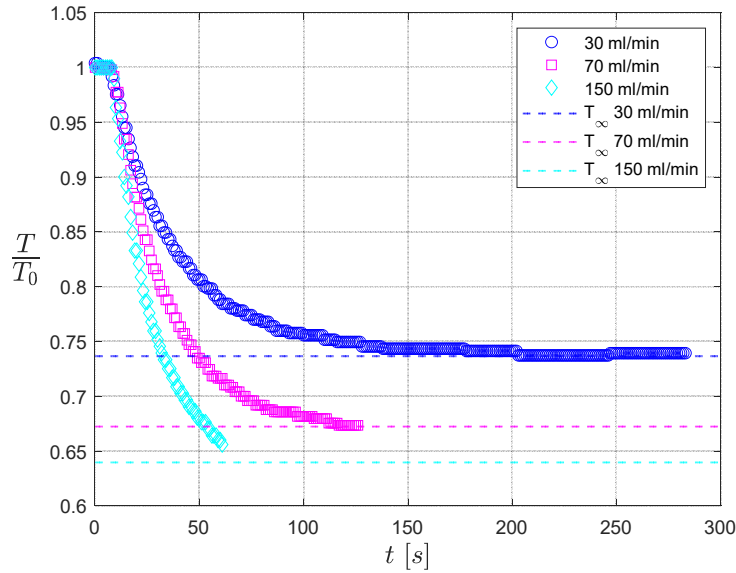


Figure.5 Non-dimensional temperature profile in time obtained from a thermocouple placed in the aluminium base, at 0.5 mm below the middle point of the bend. T_0 is the stable temperature reached before running the test. T_{∞} is the estimated stabilization temperature.

Logarithmic plots of Θ show a clear linear trend whose slope represents the rate of decay, thus the thermal transient rate $b=1/\tau$. As a general remark, higher Reynolds numbers are associated with higher transient rates, thus exhibiting faster cooling at all thermocouples stations (Table.3). Notably, this increasing b -trend is evident for all basement thermocouple signals, with a sharp distinction among the b -values of the three Reynolds numbers. However, for Station 4 thermocouple, b -values of Reynolds numbers 735 and 1540 get closer to each other, and a clear difference is shown only for the lowest Reynolds number case.

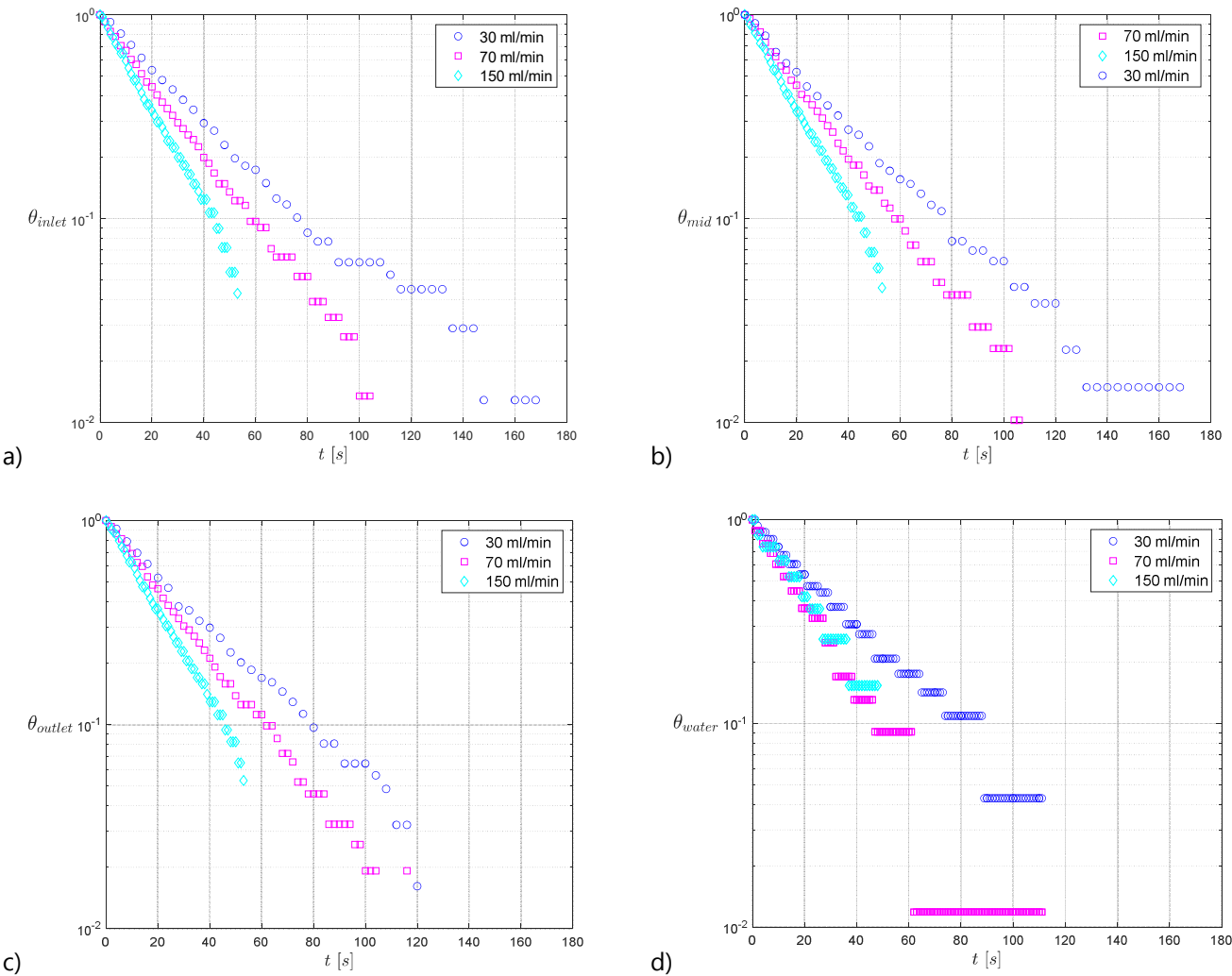


Figure.6 Non-dimensional Θ trends for thermocouple signals obtained at four measurement stations: (a) begging of the bend on the inlet side, (b) in the middle of the bend, (c) end of the bend on the outlet side, and (d) end of the outlet tube.

Table.3 Thermal transient time constant estimated at Station 1 (beginning of the bend), Station 2 (middle of the bend), Station 3 (end of the bend/beginning of the outlet channel) and Station 4 (end of the outlet tube), for all considered Reynolds numbers.

Meas. Station	τ [s]		
	$Re \approx 320$	$Re \approx 735$	$Re \approx 1540$
1	33.60	25.38	18.69
2	32.59	25.10	18.87
3	32.67	25.96	19.68
4	32.75	20.64	23.30

4.3 Velocimetry results

Figure.7a-c-e shows the non-dimensional average velocity magnitude fields obtained through PIV for all considered Reynolds numbers. Flow behaviour is consistent in all cases: at the beginning of

the first pass of the bend (i.e. first half) it is nearly undisturbed. It then starts to be deflected as it approaches the bend and separates from the inner convex wall. Subsequently, it is partially pushed towards the outer concave wall of the bend due to the centrifugal force while a region of lower velocity develops on the inner side. In the second half of the bend (second pass), at the lowest Reynolds number, it partially migrates back toward the bend geometric centreline, until it approaches the outlet, where it is again pushed toward the outer wall. Meanwhile, for the intermediate Reynolds number, it remains more attached to the concave wall, and for $Re \approx 1540$ it shows only a small area of lower velocity next to the outer wall slightly after the middle of the bend. Further downstream, it is again pushed toward the concave wall and at the end of the bend (i.e. the beginning of the outlet channel) the highest local velocity region is clearly leaning against the concave wall, while showing lower velocity in the inner part of the channel. These findings align well with what observed by Zhang&Liu (2020) for a serpentine microchannel and Liou et al. (2018) for a U channel equipped with two consecutive 90-degree bends. To summarise, as the Reynolds number increases, the effects of separation and velocity peak displacement toward the concave wall become more pronounced. Indeed, while at the lowest Reynolds number the main flow manages to recover toward the channel centreline in the second pass, as evidenced by the local velocity maximum in that region, this recovery becomes less effective at higher Reynolds numbers. As a result, moving from the outer concave wall toward the inner-convex wall along the channel in the first pass, velocity profiles in the radial direction, r , exhibit a progressively stronger radial reduction, depending on the Reynolds number. As an example, Figure.8 shows the non-dimensional average velocity magnitude profile at the middle cross-section of the bend (i.e. at $X/D_h=0$), where the velocity decreases from approximately 20% of the local maximum at the lowest Reynolds number, to approximately 30% at the intermediate Reynolds number, and to approximately one-half at the highest Reynolds number. In any case, beyond the middle of the bend, the local velocity peak decreases in magnitude, while the inner low-velocity region shows a progressive increase in velocity magnitude. Approaching the outlet channel, at the bend exit, a second displacement occurs as evidenced by a clear low-velocity region close to the inner wall.

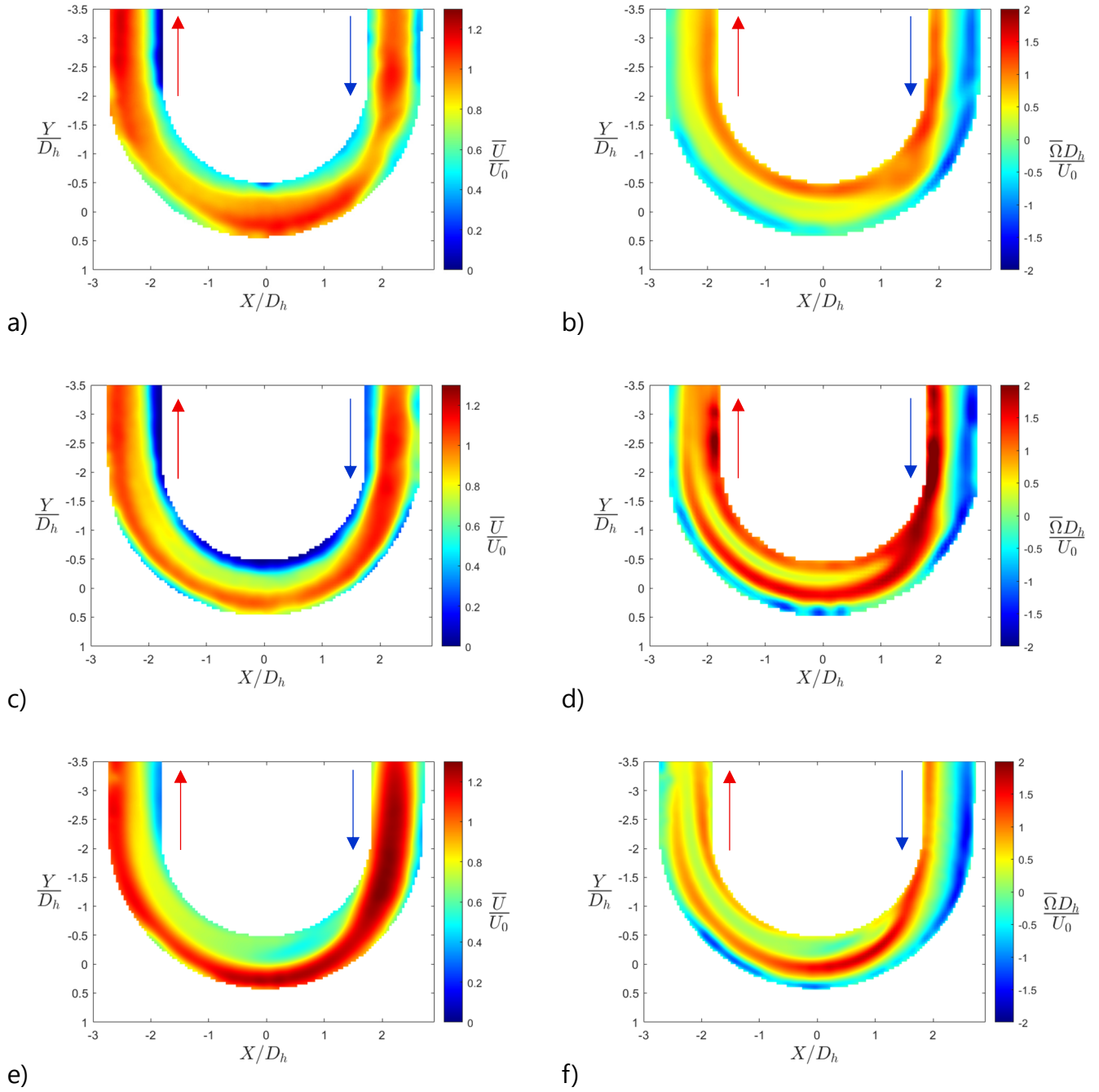


Figure.7 Non-dimensional average velocity magnitude fields (a,c,e) and vorticity fields (b,d,f) for the three considered Reynolds numbers: ≈ 320 (a,b), ≈ 735 (c,d) and ≈ 1540 (e,f). Blue and red arrow indicate the main flow direction.

Consistent to this description, vorticity fields show a coherent behaviour across the three Reynolds number cases, with differences due to the different velocity conditions (Figure.7b-d-f). In all cases, the presence of a positive inner (i.e. close to the convex wall) and negative outer (i.e. close to the concave wall) shear layer is observed in the inlet channel. As the flow approaches the bend in the first pass, the inner shear layer increases in size and splits into two positive vorticity regions: one detached and one attached to the wall. This gives rise to a vorticity distribution in the middle of

the bend characterised by an elongated region of approximately zero vorticity located in between the two. Simultaneously, the outer shear layer becomes thinner as the flow approaches the first pass and is clearly displaced toward the concave wall. The evolution of the two positive vorticity regions in the inner part of the bend, along with the almost zero one in between, is dependent on Reynolds number. Indeed, for the lowest flow rate, the positive-attached region is thicker and closely follows the convex wall, while the detached vorticity shows lower intensity values (equal to or less than half of the former) along its entire extent, so that the area approximately zero vorticity in between is not clearly distinguishable. At the intermediate Reynolds number (i.e. $Re=735$), the two positive regions show similar intensity and thickness, so that the zero-vorticity area in between is clearly delimited. For the highest Reynolds number, the detached vorticity region exhibits high values and thickness, while the attached one almost vanishes as it approaches the middle of the bend, only to reappear after this position and increase again in thickness to a size comparable to the detached one. Interestingly, the bifurcation point of the two positive vorticity regions progressively shifts upstream as the Reynolds number increases.

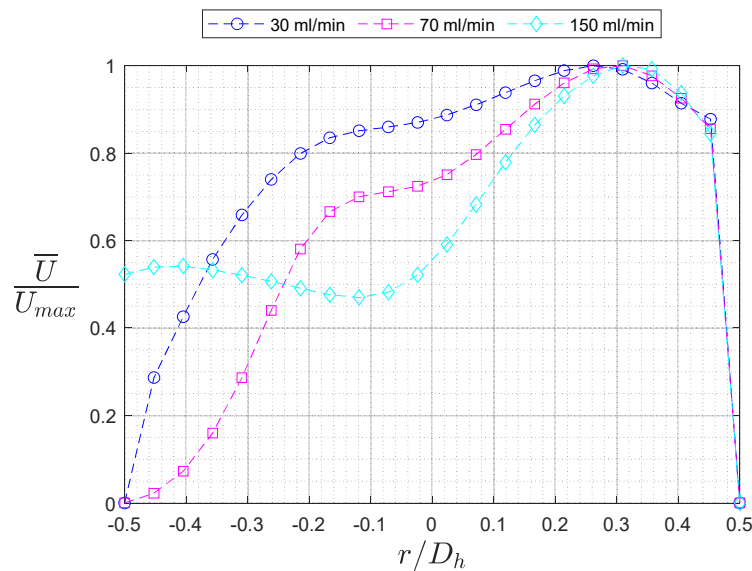


Figure.8 Non-dimensional velocity magnitude profiles along the radial direction, r , at the cross-section through the middle of the bend, i.e. at $X/D_h=0$. The local r -axis is centred at the middle point of the bend, i.e. $(X/D_h, Y/D_h)=(0,0)$, and is positive toward the concave wall and negative toward the convex wall.

4.4 IRT results

Equation 2 highlighted the strict connection between thermal transient rate b and the convective heat transfer coefficient. Therefore, the study of the spatial distribution of this parameter is of primary importance. Figure.9 shows non-dimensional thermal transient rate fields for all considered Reynolds numbers, together with contours at 0.05 intervals. As a result, the thermal transient rate increases with Reynolds number, ranging from approximately 1 to 1.5, suggesting a

faster cooling when the flow rate is increased. However, its distribution is not uniform along the channel. Indeed, in the lowest and intermediate Reynolds numbers, it is initially low, with minimum values located in the inlet channel. It then starts to increase, reaching a maximum at approximately the middle of the bend, and remains high across the entire second pass, up to the outlet channel.

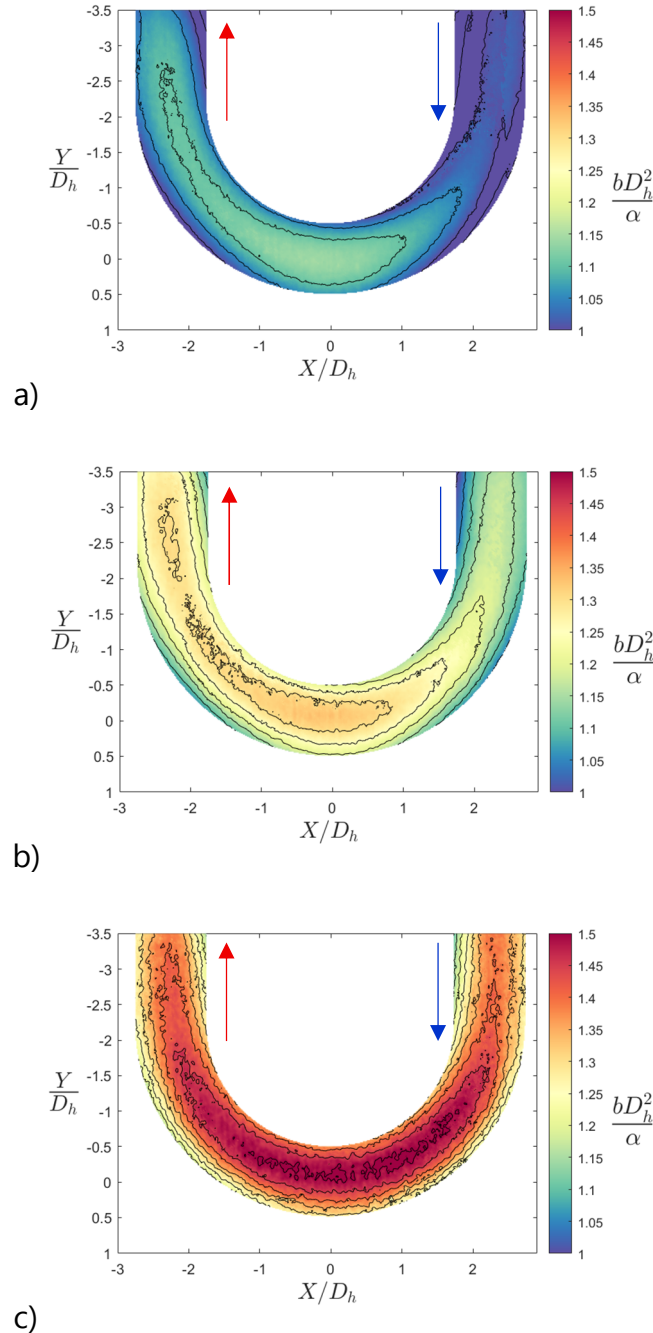


Figure.9 Non-dimensional thermal transient rate fields for Reynolds number ≈ 320 (a), ≈ 735 (b) and ≈ 1540 (c).

Taking the middle point of the bend as reference (i.e. $X/D_h \approx 0$ and $Y/D_h \approx 0$), contours of thermal transient rate (Figure.9) show that this increasing behaviour appears to be Reynolds number dependent, so that the onset of the rising region progressively shifts upstream by approximately

one hydraulic diameter in the X-direction and $\approx 0.2D_h$ in the Y-direction for $Re \approx 320$, to more than 1 hydraulic diameter in the X-direction and $\approx 1D_h$ in the Y-direction for $Re \approx 1540$.

In the second pass, this area of intense non-dimensional transient rate, greater than 1, 1.2 and 1.4 for the lowest, intermediate and highest Reynolds numbers, respectively, propagates downstream while remaining closer to the convex wall, until it reaches the centre of the channel at the beginning of the outlet channel. In fact, while for the lowest Reynolds number the high thermal transient region extends into the outlet channel ($Y/D_h \approx 3$), this occurs less markedly at the intermediate Reynolds number and appears to stop slightly earlier at the highest Reynolds number.

5. Discussion and conclusions

Thermocouple results at Station 1, 2 and 3 show a consistent increase of thermal transient rate with Reynolds number, as well as infrared results (Figure.6, Table.3 and Figure.9). This can be attributed to the higher average flow velocity inside the channel at higher flow rates. As the flow becomes faster, the fluid has less time to remove heat from the wall, so its temperature remains lower at a given downstream location than in the low-Reynolds-number cases. This leads to a larger temperature gradient at the wall and, consequently, to a higher local convective heat transfer coefficient (Cengel, 2009). However, the comparison of all thermocouple transient rates for the same Reynolds number, as well as the thermal transient rate fields derived from IRT, highlights a non-uniform distribution of this parameter along the channel. Its spatial distribution appears to depend consistently on Reynolds number: higher values are localised in the middle of the bend and the beginning of outlet channel for $Re \approx 320$, while shift upstream for the two higher flow rates. These findings are further confirmed by the spatial distribution of thermal transient rate on the top surface of the channel, whose highest values are located in the second pass of the bend for the lowest Reynolds number, at the middle of the bend for the intermediate Reynolds number and in a broader region slightly displaced toward the inlet channel for the highest Reynolds number (Figure.9). These findings can be related to the fluid dynamics quantities described in Figure.7. Indeed, the analysis of average velocity and vorticity fields reveals a qualitatively good correspondence between the onset position of the higher thermal transient rate region observed in Figure.9 and the area where the flow partially detaches from the convex wall in the first pass, at the vorticity bifurcation. Here, the vorticity behaviour shed lights on the formation of two distinct regions in terms of flow characteristics. On one side, part of the flow is pushed toward the outer convex wall and proceeds downstream in a more compact and regular way. Meanwhile, the other part of the flow describes a low-vorticity region that cannot be fully explained by our 2D measurements. Indeed, observations of raw PIV images highlight the rise of a helicoidal vortex in

this region, which propagates toward the middle and through the second pass of the bend. These vortices are described in literature as Dean vortices (Saffar et al., 2023) and have already been observed in microchannel flows (Marshall et al., 2022; Nivedita et al., 2017; Liou et al., 2018, Di Carlo, 2009). The onset and configuration of these vortices are characterized by the Dean number, De (Saffar et al., 2023). In our experiments, the relatively high Dean number (approximately 152, 346 and 725, evaluated at the bulk temperature for 30 ml/min, 70 ml/min and 150 ml/min, respectively) is sufficient to induce the triggering of primary counter-rotating vortices, whose strength increases with Reynolds number, and possibly the onset of secondary vortices (Marshall et al., 2018). These coherent structures are responsible for the heat transfer enhancement occurring at the curve, owing to the additional fluid mixing they induce (Marshall et al., 2018). It can be speculated that they promote the transport of colder fluid from the inner core of the flow toward the heated bottom wall, thereby replacing the near-wall fluid and thinning the thermal boundary layer. As a consequence, the local temperature gradient, and thus the convective heat transfer coefficient, increases. This effect persists until the vortices vanish, possibly associated with the closure of the two high-positive vorticity regions observed in Figure.7.

To summarize, this work investigated a mini-channel device equipped with a semicircular bend. This geometry was selected with the aim to provide a more uniform heat transfer distribution than that observed for a 90-degree bend in previous studies. Overall, point-wise measurements obtained through thermocouples in the aluminium basement, as well as the infrared thermography results, suggest the presence of an elongated enhanced heat transfer region that shifts from the second pass of the bend at low Reynolds number, towards the middle of the bend and the beginning of the first pass at higher Reynolds number. The onset of this region is associated to the formation and propagation of Dean vortices, whose strength depends on Reynolds number. Their presence is highlighted by vorticity fields, which show a bifurcation into two positive vorticity regions that propagates along the curve, with an extent that increases with Reynolds number.

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