

**Advanced Experimental Aero-Thermal Investigations
of Differently Enhanced Tubular Reactors**

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Notation

Roman symbols

A	Coefficients matrix	[-]
A_c	Surface area of heat exchange by convection	[m ²]
A_x	Surface area of heat exchange by conduction in the axial direction	[m ²]
A_γ	Surface area of heat exchange by conduction in γ direction	[m ²]
A_θ	Surface area of heat exchange by conduction in the angular direction	[m ²]
A_ρ	Surface area of heat exchange by conduction in the radial direction	[m ²]
A_ϕ	Surface area of heat exchange by conduction in the ϕ direction	[m ²]
$a_{i,j}$	Element of the matrix A located at row i and column j	[K]

B	Number of resampled datasets	[-]
$\bar{b}$	Vector of the well-known terms	[-]
b_m	Element at position m of the vector $\bar{b}$ with $m=1, \dots, M$	[K]
b_{X_i}	Systematic uncertainty of a variable X_i	[-]
C_f	Skin friction coefficient	[-]
$C_{f,smooth}$	Skin friction coefficient of a bare pipe	[-]
c_p	Specific heat capacity at constant pressure	[kJ/kg·K]
$c1$	Camera 1	[-]
$c2$	Camera 2	[-]
D	Diameter of the test section	[mm]
D_H	Hydraulic diameter	[mm]
D_p	Print diameter of the spherical cavities	[mm]
d	Maximum depth of the spherical cavities	[mm]
E	Height of the obstacles with a square-section	[mm]
e	Height of the continuous helicoidal turbulator	[mm]
e_{max}	Maximum height of the discontinuous obstacles	[mm]
$F_{camera,1}^X$	Mapping function $X=f(x,y,z)$ for camera 1	[mm]
$F_{camera,1}^Y$	Mapping function $Y=f(x,y,z)$ for camera 1	[mm]
$F_{camera,2}^X$	Mapping function $X=f(x,y,z)$ for camera 2	[mm]
$F_{camera,2}^Y$	Mapping function $Y=f(x,y,z)$ for camera 2	[mm]
f	Lens' focal length	[mm]
$f/$	f stop of camera objectives	[-]
H	Hue of light	[-]
$\bar{H}_{sp}$	Fit curve for the Hue distribution	[-]
h	Convective heat transfer coefficient	[W/m ² ·K]

hc	Channel's height	[mm]
I	Intensity of light	[-]
k	Dimensionless turbulent kinetic energy	[-]
k_f	Thermal conductivity of the film	[W/m·K]
k_{Inc}	Thermal conductivity of the Inconel foil	[W/m·K]
k_P	Acrylic glass thermal conductivity	[W/m·K]
k_{TS}	Thermal conductivity of the cast test section	[W/m·K]
k_{XPS}	XPS thermal conductivity	[W/m·K]
L	Length of the test section	[mm]
l_y	Width of the TLC calibration	[mm]
M	Number of nodes	[-]
M_y	Number of points of the Hue in the y direction	[-]
$\dot{m}$	Mass flow rate	[kg/s]
N	Temporal number of data points	[-]
N_t	Number of data points	[-]
N_x	Number of points of the Hue in the x direction	[-]
Nu	Nusselt number	[-]
Nu_s	Nusselt number for smooth pipes	[-]
$\mathbf{n}$	Vector normal to a surface	[-]
n_R	Refractive index of a material	[-]
n_{xz}	Projection of the normal vector on the xz plane	[-]
n_{yz}	Projection of the normal vector on the yz plane	[-]
Pr	Prandtl number	[-]
P	Number of points in the streamwise direction resolved by the IR camera during the in-situ calibration	[-]

p	Helix pitch	[mm]
p_{static}	Static pressure	[Pa]
Q	Number of points in the x direction resolved by the SLR camera during the in-situ calibration	[-]
$\dot{q}_{cond}$	Heat flux transferred by conduction	[W/m ²]
$\dot{q}_{cond,x}$	Heat flux transferred by conduction in the x-direction	[W/m ²]
$\dot{q}_{cond,\gamma}$	Heat flux transferred by conduction in the γ -direction	[W/m ²]
$\dot{q}_{cond,\theta}$	Heat flux transferred by conduction in the Θ -direction	[W/m ²]
$\dot{q}_{cond,\rho}$	Heat flux transferred by conduction in the radial direction	[W/m ²]
$\dot{q}_{cond,\varphi}$	Heat flux transferred by conduction in the φ -direction	[W/m ²]
$\dot{q}_{conv}$	Heat flux transferred by convection	[W/m ²]
$\dot{q}_{rad}$	Heat flux transferred by radiation	[W/m ²]
$\dot{q}_X$	Heat flux transferred by conduction in the axial direction	[W/m ²]
$\dot{q}_r$	Heat flux transferred by conduction in the radial direction	[W/m ²]
$\dot{q}_\theta$	Heat flux transferred by conduction in the angular direction	[W/m ²]
$\dot{q}_j$	Heat production by Joule effect	[W/m ²]
R	Pipe radius	[mm]
Re	Reynolds number	[-]
r	Radial position inside the test section	[mm]
r_{ext}	Outer radius of the tube	[mm]

r_{int}	Inner radius of the tube	[mm]
r_s	Radius of the spherical cavity	[mm]
(r_s, φ, γ)	Spherical coordinate system	[mm, °, °]
S_{i-1}	Surface of heat exchange between nodes (i,j,k) and (i-1,j,k)	[m ²]
S_{i+1}	Surface of heat exchange between nodes (i,j,k) and (i+1,j,k)	[m ²]
S_{j-1}	Surface of heat exchange between nodes (i,j,k) and (i,j-1,k)	[m ²]
S_{j+1}	Surface of heat exchange between nodes (i,j,k) and (i,j+1,k)	[m ²]
S_{k-1}	Surface of heat exchange between nodes (i,j,k) and (i,j,k-1)	[m ²]
S_{k+1}	Surface of heat exchange between nodes (i,j,k) and (i,j,k+1)	[m ²]
S_{xy}	Component x-y of the strain-rate tensor	[1/s]
s	Random uncertainty of a variable	[...]
$std(...)$	Standard deviation of a variable	[...]
T	Temperature	[K]
T_0	Total temperature of the air flow enetering the first stage of a the compressor of a gas turbine	[K]
T_b	Bulk temperature of the flow	[K]
T_c	Temperature at the axis of the pipe	[K]
T_i	TLC nominal activation temperature	[K]
T_{ref}	Reference temperature	[K]
T_s	Temperature of the surrounding elements	[K]
T_{solid}	Temperature of the nodes inside the thickness of the test section	[K]

T_{TLC}	TLC temperature	[K]
T_w	Wall temperature	[K]
$T_{w,ext}$	External temperature	[K]
t_{XPS}	Thickness of the XPS isolator	[]
U	Axial velocity measured inside the calibrated plane	[m/s]
$U_{95}(\dots)$	Extended uncertainty of a physical quantity with a probability of 95%	[...]
$u(\dots)$	standard uncertainty of a physical quantity	[...]
$\langle uu \rangle$	Variance of the axial velocity U	[m ² /s ²]
$\langle uv \rangle$	Covariance of the axial (U) and radial (V) velocities	[m ² /s ²]
U_b	Bulk velocity of the pipe flow	[m/s]
$\mathbf{V}$	Velocity vector	[m/s]
V	Radial velocity measured inside the calibrated plane	[m/s]
V_J	Finite volume of the Inconel foil	[m ³]
V_S	Voltage across the heating foil surface	[Volt]
$V_{X,c1}$	Velocity vector component in X detected by camera 1	[m/s]
$V_{X,c2}$	Velocity vector component in X detected by camera 2	[m/s]
$\langle vv \rangle$	Variance of the axial velocity V	[m ² /s ²]
$var(\dots)$	Variance operator	[...]
W	Azimuthal velocity measured inside the calibrated plane	[m/s]
$\mathbf{W}^*(x, y)_j$	Randomly resampled dataset of W velocities at the point (x,y)	[m/s]

$W_{boot,i}(x, y)$	Random sample chosen from the original dataset with the bootstrap method	[m/s]
$\overline{W^*(x, y)}_j$	Mean value of one resampled dataset	[m/s]
$\overline{\overline{W^*(x, y)}}_j$	Mean of all the mean bootstrap values	[m/s]
$\langle ww \rangle$	Variance of the axial velocity W	[m ² /s ²]
X/e	Dimensionless axial direction	[-]
(X, Y)	Coordinate system of the cameras sensors	[pix, pix]
x	Axial direction	[mm]
x_d	Streamwise distance between two consecutive dimples	[mm]
x^*	Dimensionless axial coordinate in the range [0,1]	[-]
(x, y, z)	Coordinate system of the real world	[mm, mm, mm]
(x, y', z')	Translated Cartesian coordinate system	[mm, mm, mm]
(x, ρ, θ)	Cylindrical coordinate system	[mm, mm, °]

Greek symbols

α	Angle of the normal vector with the horizon	[°]
β	Angle between n_{xz} and z	[°]
γ	Ratio of the constant pressure to constant volume specific heat capacities	[kJ/kg·K]
$\Delta(...)$	Difference of a variable or function	[...]
δ	Depth of the dimples	[mm]
$\partial(...)/\partial(...)$	Partial derivative of a function	[.../...]
ε	Dissipation rate	[m ² /s ³]

ε_{TLC}	TLC emissivity	[-]
η_{comp}	Isentropic efficiency of an axial compressor	[-]
η_{th}	Thermal efficiency of a gas turbine	[-]
η_{turb}	Isentropic efficiency of an axial turbine	[-]
Θ	Circumferential direction	[°]
θ_i	Sensitivity of a variable with respect to the quantity X_i	[.../...]
ϑ	Helix angle	[°]
ν	Degrees of freedom	[-]
ρ	Density	[kg/m ³]
ρ_s	Radial coordinate over the spherical cavity	[mm]
σ	Boltzmann's constant	[kg m ² /s ² K]
$\bar{\tau}_{wall}$	Wall shear stress averaged over the ribbed surfaces	[N/m ²]
$\tau_{yx} _{wall}$	Component y-x of the wall shear stress	[N/m ²]
ω	Vorticity component in z	[1/s]

Other symbols

$\overline{(\dots)}$	Area-average of a variable over the ROI
----------------------	---

Abbreviations

1D	One dimensional
2D	Two dimensional

3D	Three dimensional
ABS	Acrylonitrile butadiene styrene
AF	Auto focus
CCD	Charge-coupled device
CFD	Computational fluid dynamics
CL	Confidence level
CMOS	Complementary metal-oxide semiconductor
DC	Direct current
DNS	Direct numerical simulations
DPI	Differential Pressure
EDC	Ethylene dichloride
<i>EF</i>	Enhancement Factor
<i>HSI</i>	Hue-Saturation-Intensity
IHT	Intensified heat transfer
IR	Infrared
ISO	International organization for standardization
LCT	Laboratory for chemical technology
LCTG	Liquid crystals thermography
LE	Leading-edge
LED	Light emitting diode
LES	Large eddy simulation
LIP	Laser-induced phosphorescence
MERT	Mixing element radiant tube
MF	Magnification Factor
OPR	Overall pressure ratio
PCA	Principal Component Analysis

PET	Polyethylene terephthalate
PhD	Philosophy doctor
PIV	Particle image velocimetry
PLIF	Planar laser-induced fluorescence
PMMA	Poly methyl methacrylate (Acrylic glass)
POD	Proper orthogonal decomposition
s-PIV	Stereoscopic particle image velocimetry
RANS	Reynold averaged Navier-Stokes equations
RGB	Red green blue
ROI	Region of interest
RPRT	Reference platinum resistance thermometer
SLR	Single lens reflex
TE	Trailing-edge
TCF	Transparent conductive film
TIT	Total inlet temperature
TKE	Turbulent kinetic energy
TLCs	Thermochromic liquid crystals
TSP	Temperature sensitive paints
UGent	University of Ghent
WR-LES	Wall resolved-Large eddy simulation
PS	Extruded polystyrene

Summary

Generally, the enhancement techniques in heat exchangers make use of inserts, wall corrugations, and flow obstacles to augment the local convective heat transfer. Notably, the boundary layer gets disrupted, and a massive 3D separation takes place. Consequently, flow reattachment at the wall induces an augmentation of the local flow convection, which improves the overall thermal performance of the system. Also, the wall reattachment lowers the surface temperature distribution at the inner wall of the heating systems. This phenomenon extends the running life of the system since it keeps the metal temperatures at lower values than usual. Therefore, this enhancement methodology is considered energy-saving oriented, and it finds several industrial applications as in turbomachinery, in petrochemical processes, in electronic circuits, biomedical instrumentation, and microfluidics. For instance, inside the blades of the high-pressure turbine stages, the enhanced cooling channels allow increasing the Turbine Inlet Temperature (TIT). This leads to an improvement in the gas turbine thermodynamic efficiency. Likewise, in tubular reactors for the steam cracking of large molecules of hydrocarbons, it reduces the coke deposition at the wall. Notably, the increased turbulence level allows having a more significant mass and momentum exchange, which provides a more uniform radial temperature profile.. This augments the run length of the reactors, with the benefit of yielding a higher production of lighter

hydrocarbons. As a commercial example, the Mixing Element Radiant Tube (MERT) is a successful enhanced tubular reactor for the steam cracking process, designed by the Japanese company Kubota. This turbulator concept consists of a semi-circle rib element inserted helically on the internal surface of the tube. Its helicoidal shape provides an intense swirl motion to the flow, which contributes to avoid stagnation zones and corresponding coke deposition.

The work presented in this thesis addressed the experimental investigation of the effects of different types of turbulators on the flow field and convective heat transfer in tubular heat exchangers. As highlighted previously, the increase of the Nusselt number is mainly attributed to the flow separation and wall reattachment behind the obstacles. Because of the shape of the turbulators, swirling and secondary flows take place, also participating in the local convection at the wall. The experimental campaigns were based on the use of Liquid Crystals Thermography (LCTG) to measure the wall temperature and compute the heat transfer coefficient distribution at flow regimes in the range $Re=20000-80000$. Liquid Crystals Thermography indeed allows making 2D measurements of temperature fields over surfaces. Differently from Temperature Sensitive Paints (TSP), the light scattered by the Thermochromic Liquid Crystals (TLCs) is reversible with the temperature. This means that once calibrated in-situ, the measurement technique can be used several times until aging issues. Despite this, LCTG is affected by significant uncertainties on the temperature and, consequently, heat transfer measurements. This Ph.D. dissertation is also focused on providing a technical methodology to lower the measurement errors given by TLCs.

Stereoscopic-PIV (s-PIV) was applied as flow field velocimetry to quantify the wall-bounded turbulent structures and fluctuations at $Re=20000$. The optical technique requires the use of two CCD cameras placed at different angular positions with respect to the Region of Interest (ROI). Each camera acquires, in statistically independent events, a couple of particle images. As in normal planar PIV, the cross-correlation of the pairs of images provides, for each camera, 2D velocity distribution

over the plane depicted by the laser sheet. Through a Principal Component Analysis (PCA), the stereographic-PIV can reduce the four-velocity components measured by the two cameras to a new 3-components velocity system. This process is the basis for the third component reconstruction, not achieved by a standard planar PIV system. Despite the larger quantity of information, the uncertainty of s-PIV techniques reaches levels higher than the usual value of 0.1 pixels. This is mainly due to the distortions introduced by the angular positions of the cameras, the 3D reconstruction method, and the misalignment among laser and calibration plate. To reduce such a measurement uncertainty, the author used water prisms to reduce the image distortions, and applied self-calibration algorithms, as suggested by the literature.

It was also observed that, although the enhancement techniques offer an augmentation in heat transfer, the friction losses increase as well, when compared with smooth pipes. For this reason, a characterization of the pressure losses inside the enhanced tubes is also provided. Due to the low-pressure levels of the flow, a differential ultra-low-pressure instrument was used for the friction loss evaluation. Both the pressure losses and heat transfer measurements were carried out in a low-speed wind tunnel, while the s-PIV campaigns were performed in a water closed-circuit. The two different facilities were designed and built at the von Karman Institute for Fluid Dynamics, specifically for the measurement campaigns required by the here presented Ph.D. work-frame.

The three turbulator geometries presented in this thesis have geometrical parameters designed to maximize the *Nusselt number* while keeping the skin friction coefficient as low as possible. The aerothermal effects of continuous and discontinuous helicoidal ribs on the wall-bounded turbulent structures are compared. The continuous helix has a pitch-to-diameter-ratio (p/D) equal to 11 and an inclination angle of 80° . The obstacle height-to-diameter-ratio (e/D) is equal to 3.6%, this value is constant throughout the helix-wise direction. Alternatively, the discontinuous helicoidal insert is similar to the continuous one except that it has a varying obstacle

height throughout the helix. In addition to these ribs, the aerothermal effects of arrays of spherical cavities are also investigated. These dimples have a depth-to-print diameter (d/D_p) of 0.18, and a print diameter-to-pipe diameter (D_p/D) of 0.25. The streamwise distance between consecutive cavities of the array has a dimensionless value (x_d/D) of 0.56. All the geometrical parameters defining the three turbulators are in-line with those suggested by both parametric analysis and aero-thermal optimizations presented by several authors in the literature.

The results of the experimental campaigns put on evidence the heat transfer augmentation obtained with the three proposed technologies. Also, the flow field phenomena and turbulent structures have been captured and linked with the rise in local wall convection. Besides, the increase in the skin friction coefficient in the three enhanced tubular systems are measured and compared. In Chapters 2 and 3, the experimental results concerning, respectively, the flow field and the wall heat transfer in helicoidally ribbed pipes are shown. In Chapter 4, the investigation reveals a better overall performance in using cavities instead of helicoidal inserts, which provide the minimum rise in Cf with an increase in Nu equal to 1.8 times the convective heat transfer in smooth pipes. The collection of the results with their correspondent experimental laboratory conditions are considered as a dataset for the validation of different numerical codes, performed at the Laboratory of Chemical Technology (LCT) of UGent. Validated flow solvers are subsequently known to be considered in implementations for optimization routines. The future goal of the leading project consists of finding the most performant shape to be used in enhanced tubular reactors.

In addition to this, a detailed uncertainty analysis of the experimental methodology is provided by utilizing the statistical tools suggested in literature. The analytical method used for the uncertainty evaluation in the S-PIV and the LCTG is, respectively, shown in Chapters 2 and 5.

In the conclusions of this dissertation, future possible experimental investigations are proposed, particularly in reactive flows, for validation purposes. Moreover, brand-new methodologies, as a raytracing algorithm, are recommended as an innovative way for flow measurements with PIV in strongly corrugated channels.

Samenvatting

Over het algemeen maken verbeteringstechnieken in warmtewisselaars gebruik van inzetstukken, wandruwheden en stromingsobstakels om de lokale convectieve warmteoverdracht te verbeteren. Door deze technieken wordt de grenslaag verstoord en kan de grenslaag de wand loslaten. Nadien herbevestigt de grenslaag aan de wand wat aanleiding geeft tot een vergroting van de lokale convectieve warmteoverdracht, die de algehele thermische prestaties van het systeem verbetert. Ook verlaagt de wandherbevestiging de oppervlaktetemperatuurverdeling aan de binnenwand van de verwarmingssystemen. Dit fenomeen verlengt de levensduur van het systeem omdat het leidt tot lagere metaaltemperaturen. Daarom wordt deze verbeteringstechniek beschouwd als energiebesparend georiënteerd en wordt ze industrieel toegepast in turbomachines, petrochemische processen, elektronische circuits, biomedische instrumenten en microfluïdica. Zo maken de verbeterde koelkanalen binnen de bladen van hogedruk turbinetrappen het mogelijk de turbine-inlaattemperatuur (TIT) te verhogen. Dit leidt tot een verbetering van de thermodynamische efficiëntie van de gasturbine. Ook in het stoomkraken van koolwaterstoffen worden deze technieken toegepast, waar ze de vorming en afzetting van cokes aan de binnenwand van tubulaire reactoren vertragen. De geïnduceerde turbulentie maakt het mogelijk om een betere massa- en momentumuitwisseling te hebben, wat een uniformer radiaal

temperatuurprofiel oplevert. Dit vergroot de productietijd van de reactoren wat leidt tot een verhoogde productiecapaciteit. Commercieel is de Mixing Element Radiant Tube (MERT), ontworpen door het Japanse bedrijf Kubota, een succesvol turbulatorconcept voor het stoomkraakproces. Dit concept bestaat uit een halfcirkelvormig ribelement dat spiraalvormig op het inwendige oppervlak van de buis is aangebracht. De helicoïdale vorm zorgt voor een intense wervelbeweging van de stroom, wat bijdraagt aan het voorkomen van stagnatiezones en overeenkomstige cokesafzetting.

Het in dit proefschrift gepresenteerde werk richt zich op het experimentele onderzoek naar verschillende types turbulators en hun effecten op het stromingsveld en convectieve warmteoverdracht in buisvormige warmtewisselaars. Zoals eerder aangegeven, wordt de verbeterde warmteoverdracht voornamelijk toegeschreven aan de stroomafscheiding en muurherbevestiging achter de obstakels. Afhankelijk van de vorm van de turbulatoren ontstaan wervelende en secundaire stromen, die de lokale convectie aan de muur verder beïnvloeden. De experimentele campagnes, gebaseerd op het gebruik van Liquid Crystals Thermography (LCTG), meten de wandtemperatuur en laten toe de 2D-verdeling van de warmteoverdrachtscoëfficiënt te berekenen bij stroomregimes met een Reynolds getal tussen 20000 en 80000. LCTG maakt 2D-metingen van temperatuurvelden over oppervlakken mogelijk door middel van Thermochrome Vloeibare Kristallen (TLC). Anders dan temperatuurgevoelige verf (TSP), is het licht dat wordt weerspiegeld door de TLCs gecorreleerd met de temperatuur van de kristallen. Dit betekent dat de meettechniek, eenmaal ter plaatse gekalibreerd, meerdere malen kan worden gebruikt vooraleer veroudering optreedt. Desondanks is de techniek onderhevig aan aanzienlijke onzekerheden op temperatuurmetingen en dus ook op het bepalen van de warmteoverdracht. Dit proefschrift behandelt technische methode om de meetfouten geassocieerd met TLCs te verminderen.

Stereoscopische-PIV (s-PIV) wordt toegepast om lokale snelheden, wandgebonden turbulente structuren en fluctuaties bij $Re = 20000$ te kwantificeren. Deze optische techniek vereist het gebruik van twee CCD-camera's die geplaatst zijn op verschillende hoekposities ten opzichte van de Region of Interest (ROI). Net als bij de conventionele PIV zorgt de kruiscorrelatie van de beelden voor elke camera voor een 2D-snelheidsverdeling over het vlak dat wordt weergegeven door het laserveld. Via een hoofdcomponentenanalyse (PCA) kunnen de twee paar snelheidscomponenten gereduceerd worden tot een nieuw 3-componenten snelheidssysteem. Dit proces is de basis voor de reconstructie van de derde component, wat niet mogelijk is met een conventioneel planair PIV-systeem. Ondanks de grotere hoeveelheid informatie bereikt de onzekerheid van s-PIV-technieken hogere niveaus dan de gebruikelijke waarde van 0,1 pixels. Dit is voornamelijk te wijten aan vervormingen veroorzaakt door de hoekposities van de camera's, de 3D-reconstructiemethode en de verkeerde uitlijning tussen laser en kalibratieplaat. Om de meetonzekerheid te verminderen, werden waterprisma's gebruikt om de beeldvervalsingen te verminderen en zelfkalibratie-algoritmen toegepast, zoals gesuggereerd in de literatuur.

Hoewel de verbeteringstechnieken leiden tot een betere warmteoverdracht, wordt vastgesteld dat de wrijvingsverliezen, in vergelijking met gladde buizen, ook toenemen. Om deze reden worden ook de drukverliezen in de verbeterde buizen gekarakteriseerd. Vanwege de kleine drukverschillen doorheen de stroming wordt een differentieel ultra-lage druk instrument gebruikt voor de evaluatie van het wrijvingsverlies. Zowel de drukverliezen als de warmteoverdrachtsmetingen worden uitgevoerd in een windtunnel met lage snelheid, terwijl de s-PIV campagnes worden uitgevoerd in een watercircuit. De twee verschillende faciliteiten werden specifiek voor deze meetcampagnes ontworpen en gebouwd bij het von Karman Institute for Fluid Dynamics.

De drie turbulatorgeometrieën in dit proefschrift hebben geometrische parameters die zijn ontworpen om het Nusselt getal (Nu) te maximaliseren terwijl de wrijvingsweerstandscoefficiënt zo laag mogelijk wordt gehouden. De aerothermische effecten van continue en discontinue helicoïdale ribben op de wandgebonden turbulente structuren worden vergeleken. De continue helix heeft een pitch-tot-diameter-verhouding (p / D) gelijk aan 11 en een hellingshoek van 80° . De obstakelhoogte-tot-diameter-verhouding (e / D) is gelijk aan 3,6% en constant in de helixgewijze richting. Hier staat de discontinue helicoïdale rib tegenover, met eenzelfde p/D en hellingshoek, maar met een variërende obstakelhoogte. Naast deze ribben worden ook de aerothermische effecten van rijen van bolvormige kuiltjes onderzocht. Deze kuiltjes hebben een diepte-tot-afdrukdiameter-verhouding (d / D_p) van 0,18 en een afdrukdiameter-tot-diameter-verhouding (D_p / D) van 0,25. De afstand tussen opeenvolgende holtes in de gemiddelde stromingsrichting heeft een dimensieloze waarde (x_d / D) van 0,56. Alle geometrische parameters die de drie turbulatoren definiëren, liggen in lijn met die voorgesteld door verschillende auteurs in de literatuur door middel van parametrische analyse enerzijds als aerodynamische optimalisaties anderzijds.

De resultaten van de experimentele campagnes bewijzen de toename van de warmteoverdracht voor alle voorgestelde technologieën. Daarnaast zijn de stromingsprofielen en turbulente structuren in kaart gebracht en gelinkt met de toename van de lokale wandconvectie. Bovendien wordt de toename van de wrijvingsweerstandscoefficiënt in de drie verbeterde buisvormige systemen gemeten en vergeleken. In de hoofdstukken 2 en 3 worden de experimentele resultaten met betrekking tot respectievelijk het stromingsveld en de convectieve warmteoverdracht in helicoïdaal geribde buizen weergegeven. In hoofdstuk 4 toont het onderzoek een betere algehele prestatie bij het gebruik van kuiltjes in plaats van helicoïdale inzetstukken, deze combineren een minimale stijging van C_f met een toename in Nu gelijk aan 1,8 keer de convectieve warmteoverdracht in gladde buizen. De resultaten die bij dit onderzoek worden verzameld gekoppeld met de experimentele condities,

kunnen beschouwd worden als een gedetailleerde dataset voor de validatie van verschillende numerieke codes, uitgevoerd in het Laboratorium voor Chemische Technologie (LCT) van UGent. Gevalideerde oplossingsmethoden voor stromingsproblemen kunnen worden geïmplementeerd in optimalisatieroutines. Dit staat toe om in de toekomst de meest performante vorm voor buisreactoren te gaan bepalen.

Daarnaast wordt een gedetailleerde onzekerheidsanalyse van de experimentele methodologie gegeven met behulp van de in de literatuur voorgestelde statistische hulpmiddelen. De analysemethode die wordt gebruikt voor de onzekerheidsevaluatie in de S-PIV en de LCTG wordt respectievelijk weergegeven in de hoofdstukken 2 en 5.

In de conclusies van dit proefschrift worden toekomstige mogelijke experimenten voorgesteld, met name onder reactieve condities, om doorgedreven validatie mogelijk te maken. Bovendien worden gloednieuwe methoden, als een raytracing-algoritme, aanbevolen als een innovatieve manier voor stromingsmetingen met PIV in sterk gegolfde kanalen.

Glossary

<i>Blockage ratio</i>	Percentage of the channel’s cross section area blocked by the presence of obstacles
<i>Backflow</i>	Axially directed recirculation region created in a fluid due to an adverse pressure gradient.
<i>Blue-stop</i>	Highest value of temperature measured by thermochromic liquid crystals
<i>Boolean operation</i>	Set of logical operations including the conjunction, disjunction and negation.
<i>Bootstrap method</i>	Statistical test relying on random sampling with replacement
<i>Boundary layer</i>	Part of the flow affected by the shears generated by solid surfaces
<i>Bulk velocity</i>	Mass-average of the velocity profile inside channel flows

<i>Cavity zone</i>	Zone of flow recirculation due to flow separation
<i>Computational fluid dynamics</i>	Branch of fluid mechanics that uses numerical methods and algorithms to solve and analyze problems that involve fluid flow.
<i>Coherent structures</i>	Multiscale elementary components of turbulent flows which persist in time for long periods
<i>Color-play</i>	Range of temperatures measured by the thermochromic liquid crystals
<i>Convection</i>	Heat transfer given by the movement of fluid particles
<i>Covariance</i>	Measure of the joint variability of two random variables
<i>Counter-rotating vortices</i>	Pair of vortices generated in cross-flows by the interaction flow-wall, rotating in the clock and counter-clock wise directions
<i>Cross-correlation</i>	Measure of similarity of two functions
<i>Development length</i>	Channel distance from the entrance require by the flow to get fully developed
<i>Dimple</i>	Spherical cavity used for array of indentations in enhanced heat exchangers
<i>Direct Numerical Simulation</i>	Numerical method which solves the Navier-Stokes equations for the fluid dynamics without modelling the turbulence

<i>Dissipation rate</i>	Rate at which the turbulent kinetic energy is absorbed in the process of the turbulent cascade to brake bigger scales into smaller ones till converted in heat by viscous forces.
<i>Ejection of turbulent structures</i>	Movement of turbulent structures from the wall into the main flow
<i>Extended uncertainty</i>	Standard uncertainty times the coverage factors, which are given by t-student probability analysis
<i>Film cooling</i>	Technique used to cool turbine's blades at high-pressure stages
<i>Flow reattachment</i>	Point on the wall where the dividing streamline attaches to the wall again, after flow separation
<i>Focal length</i>	Measure of how strongly the optical system converges or diverges light
<i>f stop (f/)</i>	Size of the aperture of a camera lens
<i>Gas azimuthal velocity</i>	Gas velocity in the azimuthal direction θ .
<i>Gas radial velocity</i>	Gas velocity in the radial direction r .
<i>Horseshoe vortex</i>	Turbulent structures having a bound vortex of constant circulation, particularly representatives of the wing vorticity of airplanes
<i>Infrared thermography</i>	Measurement technique to quantify the temperature distribution over surfaces by means of the radiative emission in the infrared spectrum

<i>In-situ calibration</i>	Expression used to mean that the position of the sensors is the same during calibration and measurement campaigns
<i>Isentropic efficiency</i>	Measure of the degree of degradation of energy in steady state conditions
<i>K-type thermocouple</i>	Combination of Nickel-Chromium/Nickel-Alumel cables to measure pointwise the flow temperature by means of the Seebeck effect
<i>Large eddy simulations</i>	Numerical model to solve the Navier-Stokes equation for turbulent flows, by ignoring the small eddies contribution to the turbulence
<i>Leading edge</i>	Obstacle point where the flow impinges
<i>Liquid crystals thermography</i>	Measurement technique to quantify the temperature distribution over surfaces by means of the emission in the visible light spectrum
<i>Mach number</i>	Ratio of the magnitude of a jet velocity to the speed of sound in the surrounding medium.
<i>Magnification factor</i>	Ratio among the size of an image projected over a camera's sensor and the real size of the object
<i>Notch filter</i>	Very narrow bandwidth band-stop filter
<i>Nusselt number</i>	Ratio of the convective to conductive heat transfer rate
<i>Olefin selectivity</i>	Production rate of olefin during chemical reactions
<i>Particle image</i>	Measurement technique to quantify 2 components

<i>velocimetry</i>	velocity vectors in one flow plane
<i>Peak locking</i>	A bias error in the flow velocimetry given by PIV images of particle whose diameter is lower than 2 pixels
<i>Pitch</i>	Distance corresponding to one revolution of the rib in an auger screw.
<i>Prandtl number</i>	Ratio of momentum diffusivity to thermal diffusivity
<i>Prandtl's second kind secondary flows</i>	Type of turbulent structures generated in noncircular cross sections where the wall shear stress distribution is not uniform
<i>Pressure drop</i>	Difference in total pressure between two points inside the flow, given by friction losses
<i>Proper Orthogonal Decomposition</i>	Optimized mathematical method which decomposes signals in a finite series of orthonormal functions
<i>Pumping power</i>	Energy in time required by a system to have a flow with a specific Reynolds number
<i>Raytracing</i>	Computer graphics rendering technique used to generate real images by simulating the scattering in a given scenario of all the light rays
<i>Reattachment length</i>	Distance between the separation and the reattachment points
<i>Recirculation bubble</i>	Term used in fluid dynamics to point at the recirculating zone behind obstacles and/or after flow separation

<i>Red-start</i>	Temperature at which thermochromic liquid crystals start to be optically active
<i>Refractive index</i>	Property of a material to refract light rays
<i>Reynolds-averaged Navier-Stokes</i>	Set of equations resulting from Reynolds decomposition and time averaging of the Navier-Stokes equations.
<i>Reynolds number</i>	Ratio of inertial forces to viscous forces
<i>Secondary flows</i>	Velocity structures perpendicular to the crossflow
<i>Self-calibration technique</i>	Post-processing technique applied to stereoscopic images with the purpose of improving the calibration function of the PIV system
<i>Scheimpflug mount</i>	Mechanical component used to enhance the image focus when the lens plane is not parallel to the object plane
<i>Shear layer</i>	Part of the flow where strong velocity gradients take place
<i>Shear stress</i>	Pressure forces, inside the flow, parallel to the velocity direction with an opposite orientation
<i>Skin friction coefficient</i>	Dimensionless value for the shear stress at the wall
<i>Separation</i>	Phenomenon where the boundary layer detaches from the wall due to the encounter of either obstacles or adverse pressure gradients
<i>Standard uncertainty</i>	Estimate attached to a test result which characterizes the range of values within which the true value is asserted to

lie.

<i>Stanton number</i>	Ratio of the heat transferred into a fluid and the flow thermal capacity
<i>Steam cracking</i>	Petrochemical process where saturated hydrocarbons are broken into smaller ones
<i>Stereolithography</i>	Manufacturing technique used for 3D printing of objects in a layer-by-layer fashion using photochemical processes
<i>Stereo particle image velocimetry</i>	Measurement technique to quantify 3 components velocity vectors in one flow plane
<i>Strain rate tensor</i>	Matrix containing the 6 components shear components describing the rate of deformation of fluid particles
<i>Streamlines</i>	Lines, inside the flow, tangential to the velocity vectors
<i>Subgrid-scale model</i>	Mathematical model used in LES to simulate the effects of small eddies in turbulent flows
<i>Swirling flow</i>	Flow with a predominant azimuthal velocity component.
<i>Test section</i>	Section of the facility where the flow characteristics are under investigation
<i>Tornado-like vortex</i>	Type of recirculating flow ejected intermittently from the inside of wall cavities
<i>Trailing edge</i>	Obstacle point from where the flow detaches

<i>Turbomachinery</i>	Part of mechanical engineering which studies and describes the exchange of energy inside turbines and compressors
<i>Turbulators</i>	Devices and/or inserts used to augment the level of turbulence and mixing inside channel/pipe flows
<i>Turbulent kinetic energy</i>	Energy associated to the eddies' fluctuations inside turbulent flows
<i>Variance</i>	Measure of how far a set of random values are spread out from their expected value
<i>Viscous sublayer</i>	Laminar region of the turbulent boundary layer near the wall
<i>Vorticity</i>	Vector describing the spinning motion of flow elements
<i>Wake</i>	Separated region behind flow obstacles
<i>Zero-velocity line</i>	Iso-line connecting the points of the boundary layer profiles where the velocity is null, in separated regions

1

Introduction and motivation

1.1. Introduction to enhanced pipe technology

Energy savings considerations and economic incentives have led to a growth of activity in the last decades to produce more efficient heat exchange systems [1]. The objective is to reduce the size of the heat exchanger and/or to lower the pumping power needed for a specified heat duty. The research on improved heat transfer performance is referred to as heat transfer enhancement. Generally speaking, this means to augment the heat transfer coefficient [2]. Commonly, the types of methods applied for the heat enhancement are classified as active and passive [3]. Active methods involve an external power input for the enhancement of the heat transfer. Some examples of active methods are the use of a magnetic field to disturb the seeding in a flowing stream [4], mechanical aids, surface and fluid vibration [5], electrostatic fields [6], suction or injection and jet impingement [7].

On the other hand, passive methods use surface modifications of the flow channel by incorporating obstacles or additional devices. For example, twisted-tape inserts [8], swirl flow devices [9], treated surfaces, rough surfaces, extended surfaces, and coiled tubes [10]. Wall obstacles aim at disturbing the boundary layer. They separate and reattach the latter on the wall (Figure 1.1) at a certain distance upstream and downstream from them [11]. In this way, the impingement of the shear layer and the increased Turbulent Kinetic Energy (TKE) augment and over-compensate the effects of the flow detachment leading to wall Nusselt number values (Nu) in excess of those measured for smooth channel flows [12].

On the other hand, these configurations also increase the wall shear stresses, as well as the skin friction factor [13]. Recently, considerable attention has been devoted to optimizing the geometry of turbulators, i.e. channels inserts, for cooling applications [14]. As stated before, the implementation of such obstacles provides higher local wall heat transfer but also generates a significant increase in pressure drop, which induces higher pumping costs [15]. Hence, it is essential to find the best performing geometry, which also minimizes the pressure losses [16]. Any enhancement device or technique employed for heat exchangers should be optimized between the benefits of better heat transfer coefficient and the burden of higher pumping cost due to the increased pressure losses.

Optimization algorithms have only recently been considered for heat exchangers applications. In the past years, different passive techniques were experimentally studied to have an overall image of the most performing enhancement approach [2, 3, 13, 18, 19]. As a result of those studies, and also because of their simple and cheap application, the ribbing [13] and corrugation [20] techniques are the most commonly used for heat exchangers, in turbomachinery and chemical reactors. A particular interest is given to swirl flow devices, which produce and superimpose secondary recirculation(s) to the main flow.

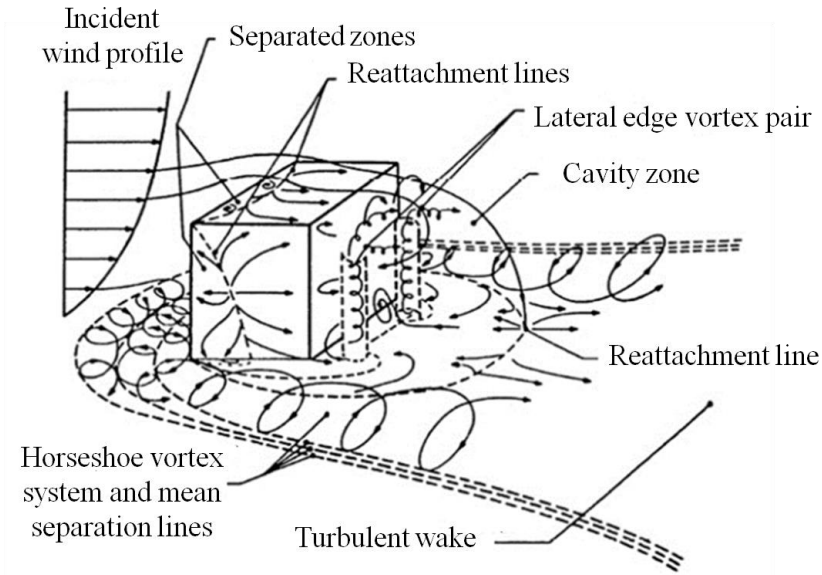


Figure 1.1 - Example of the separation/reattachment phenomena induced within the flow by the presence of a generic obstacle [17]

Several studies in the literature compare their effect on both heat transfer and pressure losses. T. S. Ravigururajan & A. E. Bergles [21] provide correlations for the evaluation of the friction factor and the heat transfer enhancement, particularly for the case of ribbed tubes. In their work, data from frequently referred investigations were gathered for a wide range of tube parameters with an obstacle height-to-tube diameter ratio (e/D) ranging between 0.01 and 0.2; a rib pitch-to-tube diameter ratio (p/D) ranging between 0.1 and 7.0; a variation in the obstacle angle ($\theta/90^\circ = 0.3$ to 1.0). The main flow characteristics such as the Reynolds number ($Re = 5000$ to 250000) and the Prandtl number ($Pr = 0.66$ to 37.6) were widely varied as well. The correlations were verified by testing different types of commercial ribbed tubes and five wire coil inserts, under both heating and cooling conditions. Instead, for the case of cooling channels with rectangular cross-sections, friction factor and heat transfer correlations are given by P. R. Chandra et al. [22].

Of fundamental importance in this field, inserts and wall corrugations, providing swirling flows with high intensity, substantially contribute to the heat transfer enhancement [2]. For the case of helically corrugated tubes (circular-cross sections), the experimental results of S. Pethkool et al. [18] show that the thermal performance is significantly increased compared to bare tubes. The authors studied the effects of the pitch-to-diameter ratio ($p/D = 0.18, 0.22$ and 0.27) and the rib-height-to-diameter ratio ($e/D = 0.02, 0.04$ and 0.06) of the enhanced pipes on both the heat transfer and friction coefficient in the flow regime $Re=5500-60000$. A similar investigation on corrugated pipes was carried out by P. G. Vicente et al. It was extended to a higher range of Prandtl numbers. The authors found an increase from 20% to 300% in friction factor and a maximum of 250% in Nusselt number (Nu). Equally important, they reported that the heat transfer augmentation rises with the Prandtl number, i.e., for fluids whose thermal boundary layer is thicker than the viscous boundary layer ($Pr \gg 1$).

In case of helicoidal inserts, the intensity of swirling flows velocity, which contribute positively to the wall heat transfer rate by convection, depends on both the shape of the turbulator and the helix angle. Those geometry effects on both the pressure losses and heat transfer were investigated by D. L. Gee & R. L. Webb [13]. Three different helicoidal ribs ($\theta=30^\circ, 45^\circ, 70^\circ$) were investigated, with the same pitch-to-diameter ratio of 15. This investigation clearly shows that helical rib-roughness generates better heat transfer per unit friction than transverse repetitive obstacles. A recent, detailed experimental investigation is provided by I. Mayo et al. [23] in the case of a pipe ribbed with one helicoidal turbulator at $Re=20000$. The authors show experimentally the connection between peaks in the Nusselt number and the turbulent kinetic energy (TKE) of the wall-bounded swirling flow generated by the helicoidal obstacles. Higher values of TKE inside the turbulent flow mean stronger levels of mixing, leading in particular to more uniform radial temperature profiles. This is a benefit for steam cracking applications since it lowers the unwanted production of carbon coke in the core of tubular reactors [24].

1.2. Industrial applications

Representative and successful applications of enhancing techniques are respectively found in turbomachinery, for the high-pressure stages of the turbines, and in tubular reactors for steam cracking processes.

1.2.1. Applications in turbomachinery

In turbomachinery, the use of internal cooling channels of blades provides a higher thermodynamic efficiency [25]. Pressurized air is extracted from the compressor to cool down the turbine blades of the high-pressure stages (Figure 1.2). In this way, the expansion of the hot gas through the turbine can start from a higher total inlet temperature (TIT). The latter is the temperature at the exit of the combustion chamber, i.e. at the inlet of the first stage of the turbine. It corresponds to the highest value seen in the thermodynamic cycle of the machine. Since the thermal efficiency of the engine is a direct function of the TIT , the higher the temperature, the better the gas turbine performance. This is expressed in the following equation [26]:

$$\eta_{th} = \frac{\left[1 - \frac{1}{OPR^{\frac{\gamma-1}{\gamma}}}\right] \left[\eta_{comp}\eta_{turb} \frac{TIT}{T_0} - OPR^{\frac{\gamma-1}{\gamma}}\right]}{1 + \eta_{comp} \left[\frac{TIT}{T_0} - 1\right] - OPR^{\frac{\gamma-1}{\gamma}}} \quad (1.1)$$

where OPR is the overall pressure ratio of the thermodynamic cycle, γ is the ratio of the specific heats at constant pressure and volume, η_{comp} is the isentropic efficiency of the compressor, η_{turb} is the isentropic efficiency of the turbine and T_0 is the total temperature of the air flow at the entrance of the compressor. A typical annual TIT improvement of 10 K corresponds to a thermal efficiency increase of 0.1%. In a time-lapse of 5 years, the estimated improvement reaches 0.6% for the typical TIT increase of 50 K [27]. To bear in mind that the relation between TIT and efficiency (Eqn. 1.1)

is also a function of the gas properties, by means of γ . At higher temperatures, the gas has a lower specific heat ratio, which influences the thermal efficiency of the engine negatively. Therefore, the choice of TIT has to be taken in order to optimize the real thermodynamic efficiency (η_{th}), taking into account the variation of γ with temperature.

The most compelling evidence is that this type of achievement allows reducing the size of the engine for a specified output power. As a consequence, the rise in TIT affects not only the thermal but also the propulsive efficiencies and the specific fuel consumption. Cumpsty [28] evaluated in 2015 the TIT impact on the engine performance, showing that the 5-years increment of 50 K leads to a specific fuel consumption reduction of 2.8%. In other words, this would provide, as an order of magnitude, a saving of €4.5 billion per year, only in the case of civil aircraft engine sector. For this reason, it is clear that the design of cooling channels for gas turbines, allowing to increase the TIT, has a significant economic as well as ecological impact.

An overview of the state-of-the-art in cooling techniques is presented in the work of Ligrani [29]. Cooling techniques such as film cooling [30], impingement cooling [31], and rib/pin insertions [25] are employed for gas turbine engine blades and vanes. The ‘film cooling’ technique (Figure 1.2), consists in generating a protective layer of pressurized ‘fresh’ air in between the airfoil and the hot surrounding gas. This air is ejected from the internal channels through small inclined holes. In such a way, the turbine blades of the high-pressure stages are protected from the high-temperature gas. A complementary solution is the use of ribs on the inner walls of internal cooling channels. Several rib configurations were investigated.

The particular effect of angled-ribs on the local turbulent convective heat transfer was experimentally investigated by J. C. Han & P. Zhang [33]. The ribbed test section consisted of three different square channels connected by two U-bends. The angled-

rib geometry was characterized by a height-to-hydraulic diameter of 0.063 and a pitch-to-diameter of 10.

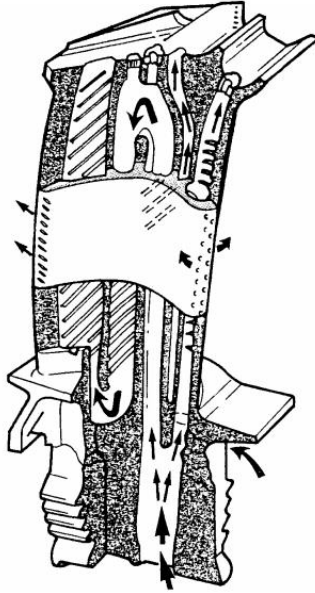


Figure 1.2 – Cooling systems for the Rolls Royce RB211-524G/H HP gas turbine blade [32]

The work, done in 1989, concluded that the rib angle, and the 180 bends strongly influence, in a positive way, the local heat transfer distribution. G. Rau et al. [34] studied the effects of the rib pitch-to-channel height on the local flow field, by using a 2D LDV technique. The distribution of the Nusselt number over the wall between the obstacles in a straight channel was quantified by the Liquid Crystals Thermography (LCTG) technique. This work shows the presence of secondary flows, generated by the obstacles inside the square channel, which generate flow impingements on both smooth side walls increasing the heat transfer. To examine the turbulent structure generated by the flow separation-reattachment induced by the insertion of ribs, high spatial and temporal resolution-Particle Image Velocimetry

(PIV) was applied by F. Coletti et al. [35]. The statistical description of the unsteady turbulence was performed at the flow regime of $Re = 15000$.

In a real scenario, the turbine blade is rotating around the engine axis; centrifugal forces are applied to the turbulent structures generated by the separation. Those effects of rotation on the flow characteristics were shown in the work of J. Ahn et al. [36] by means of Large Eddy Simulations (LES) with a dynamic subgrid-scale model. The presence of the ribs and the rotation generates strong wall-normal motions which are augmented on the trailing-wall side, resulting in a significant enhancement of the heat transfer. Besides, the rotation leads the separation bubble developing behind each rib to be smaller on the trailing-wall side and bigger on the leading-wall side.

Engineering companies, which manufacture and distribute power systems for both aviation and other industries, considerably invest in new, more advanced internal cooling techniques.

1.2.2. Applications in steam cracking reactors

In steam cracking processes, the purpose of tubular reactors is to break saturated hydrocarbons into lighter chemical structures [37]. Light olefins are considered as building blocks for many commercial essential chemicals and intermediates. For this reason, steam cracker installations are considered as the heart of many petrochemical sites. Unfortunately, those chemical reactors are one of the most pollutant processes in the chemical industry, with approximately 200 million tons of CO_2 emissions in 2000 [38]. A major challenge for this technology is related to the formation of coke deposits on the inner wall of the tubular reactors. This solid carbonaceous layer blocks part of the tube cross-section, causing an increased pressure drop, and a reduced wall heat transfer.

The resulting higher hydrocarbon partial pressures in the tubular reactors will generate a loss in olefin selectivity. In order to maintain the same cracking severity, the energy from the radiative burners is progressively increased, placing extra demands on the tube metallurgy such as tube metal temperatures, which can reach up to 1200 K. Therefore, either the metallurgic impediments in the reactor or the immoderate pressure drop will lead the operators to shut down production in order to de-coke the tubes. Therefore, this process will reduce the annual production of ethylene [39], inducing a loss of capital for the industrial company.

To solve the issues mentioned above, the tubular chemical reactors used in steam cracking or ethylene dichloride (EDC) pyrolysis are frequently equipped with wall obstacles [24, 37, 40, 41, 42]. Notably, these turbulators disrupt the inner wall boundary layer. As in gas turbine cooling channels, the subsequent reattachment of the shear layer increases the local heat transfer when compared with systems without wall corrugation. The wall-bounded swirling flow and the impingement of the reattaching boundary layer decrease the wall temperature, leading to a decrease in coke deposition on the wall and a longer operating time. Moreover, due to the ribbed structure, the flow is ejected from the hot wall to the cooler center of the channel, reducing radial temperature and concentration gradients, positively influencing selectivity on specific products during the chemical reactions [37, 43]. In short, the implementation of these structures allows a higher production of ethylene and propylene [44]. This encourages the industry to invest in such technologies.

The Japanese company Kubota patented the Mixing Element Radiant Tube (MERT), it is, so far, the most successful commercial design of enhanced tubular reactors for steam cracking process [44]. This turbulator concept consists of a semi-circle rib element inserted helically on the internal surface of the tube. The periodic disruption of the boundary layer and the rise in turbulence intensity of the flow structures lead to vast improvements of the heat transfer. The helicoidal shape provides an intense swirl motion to the flow, which contributes to avoid stagnation

zones and corresponding coke deposition. Due to the significant pressure penalty induced, the MERT (Figure 1.3) geometry needs to be improved according to optimization routines.



Figure 1.3 – Mixing Radial Element Tube of KUBOTA

To minimize this friction issue while harnessing the intensity in the flow swirling motion, alternative techniques have been developed. Notably, the Intensified Heat Transfer (IHT) [45] technology by Lummus/Sinopec (Figure 1.4) is a successful alternative applied in more than 50 Chinese furnaces. This device uses a modular application of twisted tape inserts to promote a high strong local swirl component, but with limited additional pressure losses. Despite the before mentioned behavior, they are especially sensitive to plugging during the decoking caused by detached cokes fragments. Another application of swirling flow is the Swirl Flow Tube by Technip [46]; the tube itself is corrugated to follow a small amplitude helical shape. This induces large secondary flows without providing any flow blockage or enhanced wet surface. 1

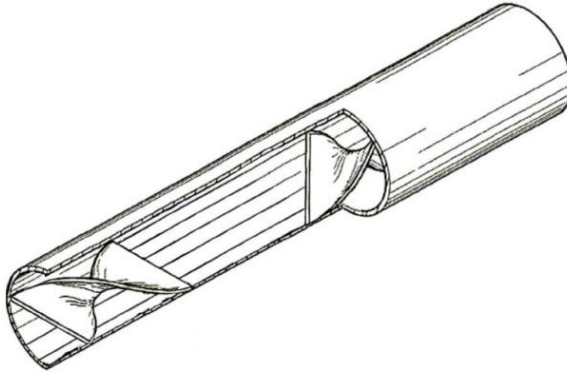


Figure 1.4 – Intensified Heat Transfer technology by Lummus [45]

In conclusion, the need in finding the most performant device that might help the industry in reducing the production costs has a relevant interest. In the open literature, several numerical tools are presented. They provide the community with flow solvers to be implemented in optimization routines. Van Cauwenberge et al. [37] and Campet et al. [47] studied the turbulent flow in a helicoidally ribbed pipe, for petrochemical applications, using Wall-Resolved Large Eddy Simulation (WR-LES) with a kinetic energy sub-grid scale model. These authors demonstrated the effects of a turbulator on both the heat transfer and the friction losses.

1.3. Objective of the present thesis

The big question is thus how to optimize the rib shape minimizing the friction losses while keeping the wall convection at high values. Numerical methods are employed today, as they provide more reliable predictions of local wall heat transfer than semi-empirical correlations. As stated before, these methods require a thorough comparison with experimental data to prove their validity and fidelity. Comparison between experimental data and the results from different numerical flow solvers for ribbed channels are provided by H. Iacovides and M. Raisee [48]. The closures used

by the authors ranged from eddy-viscosity $k-\epsilon$ to algebraic and differential stress closures. Since the large-scale motion in ribbed channels is not simulated sufficiently well in $k-\epsilon$ models, and since the use of wall functions might cause discrepancies in the heat transfer computation [49], the focus on numerical investigations shifted towards Large Eddy Simulations [37]. Notably, since the design of advanced steam cracking reactors needs to account, in detail, for the mechanism of turbulence, eddies resolving technique must be taken into consideration. D.J. Van Cauwenberge et al. [50] investigated with LES, based on a dynamic subgrid-scale model [51], both the flow field and the heat transfer in finned steam cracking reactors. The authors show the advantages of using a LES code, highlighting the inefficiency of RANS in capturing secondary flows phenomena arising from the anisotropy of the Reynolds stresses. Validated Large-Eddy Simulations in single-started helically ribbed tubes are provided by the authors D.J. Van Cauwenberge et al. [37] and [47]. However, to verify how the filtered energy equation and the corresponding sub-grid scale model represent the local heat transfer, a validation is mandatory to compare the numerical solutions with the corresponding measurements. Therefore, the main objective of the present thesis is to provide a detailed experimental characterization of 3 different enhanced geometries: 2 helically corrugated cooling channels (Figure 1.5) and 1 spherical dimpled tube (Figure 1.6). The choice of the helicoidal rib as a continuous insert (2D turbulator) in tubular channels is in-line with commercial applications, such as the MERT geometry [44].

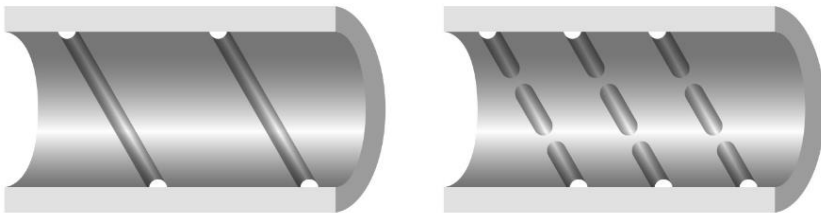


Figure 1.5 – Examples of a continuous and a discontinuous helicoidal inserts

A broken configuration of the helicoidal turbulator (3D turbulator), where the obstacle height changes along the helix pattern, is investigated in a second step. The effects of the discontinuity on the surrounding turbulent flow field and the difference in the wall convection heat transfer are compared with the continuous configuration. A third configuration is considered. In a different way from the existence of wall obstacles, where material is added to a bare tube, the third geometry applies the use of dimples. In this case, material is removed from the pipe thickness in order to generate spherical cavities where the phenomenon of separation and reattachment takes place (Figure 1.6). Static pressure losses, turbulent structures, and heat transfer performances are then compared with the other two devices. The geometrical and flow parameters of these 3 geometries were chosen in compliance with the literature, notably, in accordance with a number of parametric analysis results, provided by several authors [1, 3, 15, 52]. The effects on the turbulent flow field and the local heat transfer are analyzed to contribute with a well-documented test case for high-fidelity computations. Numerical codes, whose results are matching the experimental investigations in pipes equipped with the different obstacles, are considered as flow solvers for optimization routines in the field of enhanced tubes.

Considering the needs expressed earlier, this Ph.D. project therefore aims to measure the pressure losses, to quantify and analyze the time-averaged flow field features and the turbulence statistics by means of Particle Image Velocimetry for a flow regime close to $Re=20000$. The turbulent structures generated near the wall obstacles [34], were already investigated with planar PIV by Wang et al. [53] and Coletti et al. [54] for flows in rectangular cross-section channels. These authors highlighted the waving shear layer separated from the obstacle's edge, which generates a significant intermittence inside the whole flow field. The turbulence statistics were provided for flow regimes in the Reynolds number (Re) range 2500-20000 with a blockage ratio of 12% and a pitch-to-rib height ratio of 8. The authors showed that the rib wake region is dominated by a periodic process of bursting vortices which generate an intermittent ejection of the structures from the wall to the

core of the flow. Considering that, for the case of helically ribbed pipes, the flow has a significant velocity component value in the azimuthal direction, stereo-PIV (s-PIV) must be considered to provide useful 3D data. This technique expects the use of two cameras, which allows to lower the perspective error that occurs in the case of strong particles displacement in the direction perpendicular to the laser plane, as explained by Prasad [55].

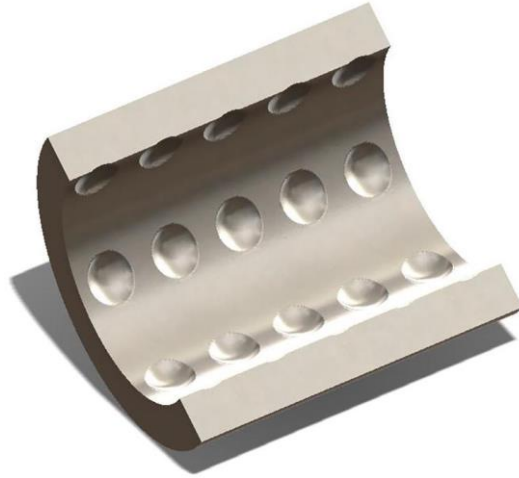


Figure 1.6 – Example of a dimpled tube [52]

In addition to the flow field investigation, a quantification of the wall heat transfer distribution between the wall obstacles is provided by using steady-state Liquid Crystals Thermography (LCTG) for the flow regime in the range $Re=20000-80000$. This technique exploits the selectivity of liquid crystals in reflecting different wavelengths of the light source as temperature changes. In Chapter 2, the uncertainty of the liquid crystals thermography and the high repeatability of the TLCs calibration are shown in detail.

In the open literature, experimental heat transfer data for enhanced tubes are often provided in terms of either a surface-averaged wall heat transfer or axial Nusselt number profile. Pointwise measurements were eventually chosen for thermal

investigations, although significant 2D variations of convective wall heat flux exist between the edges of two adjacent obstacles. In fact, the heat transfer results should be carried out using a 2D measurement techniques, to monitor the full heat transfer surface. This complies with the interest of 3D finite element models in having fully surface data generated [56]. P. T. Ireland et al. show in their work [57] how the liquid crystals thermography can be considered as a non-intrusive technique for heat transfer studies. In addition, the authors provide the experimental methodology for TLCs applications, which is based on a true-colour image processing. In the same way, F. Coletti et al. [58] investigated the flow structures' effects on the local heat transfer in rib-roughened trapezoidal cross-section channels experimentally.

Two-dimensional temperature fields can be acquired with an excellent spatial and temporal resolution by using non-intrusive pyrometry [59]. Nevertheless, their use implies the exact knowledge of the surface emissivity, which in turn depends on the emission wavelength, viewing angle, and the surface properties. The choice of using thermal paints [60] was discarded a priori due to their peculiarity of providing an irreversible colour change with time. A valid alternative to the thermochromic liquid crystals is the thermographic phosphors. The latter are luminescent materials which emit visible light after excitation by short-wavelength (e.g., UV light). This process is basically based upon two processes: a radiative transfer and a non-radiative one (which relies only on internal energy transfer). The rates of those two processes are a function of the phosphors' temperature. J. Brübach et al. [61] provides an extensive review of which phosphors to be used and which experimental methodology can be used. The authors suggest a time-resolved method where the acquired time decay of the luminescence is calibrated with temperature. To the best of the author's knowledge, an exhaustive quantization of the measurement uncertainty in using the thermographic phosphors in cooling channels hasn't provided yet in the literature. For all these reasons, the author decided to measure the Nusselt number fields by using the steady-state Liquid Crystals Thermography and continue with the experience already developed by his laboratory work-team.

The heat transfer results are provided in terms of enhancement factor (EF) distributions on the inner pipe wall. The EF is defined as the ratio between the local Nusselt numbers of the corrugated and bare pipes. Values higher than unity show where the effects of the turbulators provide an increase in the local convective heat transfer.

1.4. Outline

This thesis is a compilation of contributions that have been accepted or are to be submitted for archival journal publication. Every chapter starts with a short introduction describing the relevant context.

Chapter 2 addresses the detailed uncertainty evaluation methodology in using a Liquid Crystals Thermography. Particularly, Chapter 2 presents two examples of temperature and heat transfer measurements made in different setups, both being representative cases of internal cooling cavities. The goal is to analyze the impact of each different error source on the Nusselt number uncertainties, represented according to the ASME PTC 19.1 code [62].

In Chapter 3, stereo-PIV measurements are performed on both continuous and discontinuous helicoidal turbulators, to investigate their effects on the surrounding flow field. Specifically, at the flow regime of $Re=20000$, the effects of the discontinuity of the tube corrugation on the flow separation/reattachment, turbulence statistics, vorticity, strain-rate, swirling velocity component and Turbulent Kinetic Energy are compared between the continuous and discontinuous inserts. Besides, ultra-low differential pressure measurements are done to show the impact of the 2 helixes on the skin friction coefficient.

In Chapter 4, the experimental characterization and comparison of the local convective heat transfer distribution between the obstacles of the continuous and discontinuous configurations are carried out. The investigation is performed at 4 different flow regimes in the range $Re=20000-80000$ to evaluate the influence of the Reynolds number on the local Nusselt number. The experimental campaigns were done by using steady-state narrow-band Thermochromic Liquid Crystals (TLC) thermography, to measure the temperature distribution over a ribbed cylindrical surface. Due to their sensitivity to the angular observation direction, a novel calibration methodology is provided to reduce the measurement uncertainty. A rigorous quantification of the measurements errors on the Nusselt number values within the Region of Interest (ROI) is provided.

Chapter 5 illustrates the effects of repetitive dimples inside tubular channels. The experimental campaign was conducted by means of both PIV and stereo-PIV to investigate how the separation and following recirculating flow take place inside the spherical cavity. In addition, LCTG was used to measure the wall EF distribution due to the flow reattachment inside the dimples. For this particular case, the TLCs calibration process followed the methodology suggested by Kodzwa [63], which is slightly different from the procedure used in Chapter 4. The uncertainty related to the in-situ calibration of the wall temperature measurements is provided as well.

Finally, in Chapter 6, the general conclusions are presented and perspectives for future research are proposed. In particular, a ray-tracing-based PIV approach is suggested for the flow investigation inside highly corrugated pipes, where strong image distortions are encountered in the flow field optical measurements.

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2

Measurement uncertainty in Liquid Crystals Thermography applied to internal cooling research: Two practical examples

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Abstract

In this chapter two examples of temperature and heat transfer measurements performed in different setups, both being representative cases of internal cooling cavities are discussed. In both cases, Thermochromic Liquid Crystals (TLCs) were applied on the walls to perform steady-state Liquid Crystal Thermography (LCTG) in the active range of 308 - 328 K, allowing the determination of the Nusselt number distribution. The main goal of this work is to analyze the impact of the different error sources on the wall temperature and Nusselt number uncertainties, and to represent them according to the ASME PTC 19.1 code [1]. The first example is a stationary cooling channel model with a single heated wall, on which ribs are placed perpendicularly to the stream-wise direction. The working fluid is air, and the flow conditions are set to match a Reynolds number value equal to 53000. The corresponding surface temperature measurement uncertainty at a 95% Confidence Level (CL) is below 0.8 K, which is about 4% of the active temperature range of the employed TLCs. On the other hand, the relative uncertainty at a 95% CL on the Nusselt number measurement is estimated to be equal to 17%. The second instance is a cylindrical smooth pipe where the heating is applied over the complete inner surface. Air is also employed as operational fluid, whereas the Reynolds number value is varied from 20000 to 78000 in order to compare the experimental results with well-established correlations. In this setup, the surface temperature measurement uncertainty is below 1 K at a 95% CL, which corresponds to 5% of the TLC active temperature range. The relative uncertainty on the measured Nusselt number is estimated to be between 7 and 12% at a 95% CL, depending on the Reynolds number.

2.1. Introduction

Liquid Crystal Thermography (LCTG) is an experimental technique widely used to measure temperature and infer convective heat transfer distributions on a surface, becoming a primary tool in the internal cooling research field. LCTG has allowed the understanding of the underlying physics of both academic and simplified cooling concepts, as well as the accumulation of experimental data to be used for numerical validation. The review by Ireland and Jones [2] addresses several examples of wall heat transfer and shear stress measurements by LCTG, as well as the basic principles of liquid crystals, their main characteristics, the factors modifying the TLCs behavior, the range of application and limitations. Additional examples of wall and fluid temperature measurements by means of TLCs are included in the review by Kowalewski et al. [3]. On the other hand, Wiberg and Lior [4] present an analysis of the typical error sources in the temperature measurement by means of liquid crystals, such as hysteresis, aging, surrounding illumination disturbance, illumination and viewing angles, variation in film thickness, response time, varying amount of light into the camera, digital resolution of the image conversion system, electric fields, pressure, acceleration, and measurement noise. In the present work, these error sources are controlled in order to minimize the resulting uncertainty, although other errors sources need to be considered.

In order to obtain the heat transfer distribution from the wall temperature measurements, two main approaches can be adopted: transient or steady-state LCTG, both described in Ref. [2]. A significant number of operations are needed to obtain the final heat transfer distribution, which propagate the measurement errors corresponding to the primarily evaluated quantities (such as the wall temperature) into the final uncertainty. Despite that the measurement uncertainty of the dimensionless heat transfer coefficient, i.e. the Nusselt number, is often reported to be of order 5 to 15% in internal cooling applications, the way it is calculated is hardly ever described.

The objective of this chapter is to detail the estimation of the temperature and heat transfer measurement uncertainties in two different internal cooling experimental facilities. The first setup corresponds to a stationary cooling channel with a heated ribbed wall, whereas the second one consists of a heated smooth pipe. In both cases, LCTG is employed to measure the temperature distribution on the heated wall(s), and a number of equations are used to calculate the corresponding dimensionless heat transfer distribution. Since both facilities present elements in common with those found in the open literature, the procedure given in this chapter aims at providing a practical approach to estimate and present the temperature and dimensionless heat transfer measurement uncertainties in LCTG.

Different publications address the estimation and presentation of the measurement uncertainty, such as the ISO JCGM 100:2008 [5], NASA HDBK-8739.19-2 [6] and the ASME PTC 19.1 [1]. In the present work, the ASME PTC 19.-2005 code was employed due to its simplicity, conciseness, and its consistency with respect to the ISO code. If not explicitly stated, the uncertainties expressed in the present chapter are at the level of the standard deviation, called “standard uncertainties” in the ASME PTC 19.1. The standard uncertainty of a measured magnitude, u_{X_i} , results from the consideration of the systematic uncertainty, b_{X_i} , and the random uncertainty, s_{X_i} , corresponding to the measurement of a quantity X_i :

$$u_{X_i} = \sqrt{b_{X_i}^2 + s_{X_i}^2} \quad (2.1)$$

The challenge in any uncertainty analysis is precisely to identify all measurement uncertainty sources and to quantify them accurately, especially in LCTG. For this reason, the present chapter aims at providing a summary of the main measurement uncertainty sources affecting the temperature and heat transfer distributions in steady-state LCTG. When performing LCTG, a considerable amount of primary quantities are measured, and a relatively large system of equations is employed to obtain the

results (such as the Nusselt number). Therefore, the resulting uncertainty needs to consider the primary error sources. If a primary measured quantity is designated by X_i , the systematic and random uncertainties associated to the Nusselt number due to the measurement of X_i (b_{Nu,X_i} and s_{Nu,X_i}) are calculated as:

$$b_{Nu,X_i} = \theta_i \cdot b_{X_i} \quad , \quad s_{Nu,X_i} = \theta_i \cdot s_{X_i} \quad (2.2)$$

where $\theta_i = \partial Nu / \partial X_i$ represents the sensitivity of the Nusselt number with respect the primary quantity X_i . The resulting systematic and random uncertainties associated to the Nusselt number would result from the consideration of the uncertainty due to each primary quantity:

$$b_{Nu} = \sqrt{\sum_i (b_{Nu,X_i})^2} \quad , \quad s_{Nu} = \sqrt{\sum_i (s_{Nu,X_i})^2} \quad (2.3)$$

whereas the standard uncertainty on the Nusselt number would be calculated by applying eq. (2.1). In order to obtain the uncertainty at a confidence level equal to 95% ($U_{Nu,95\%}$, also called “expanded uncertainty”), the ASME PTC 19.1-2005 recommends multiplying the standard uncertainty by a factor of 2: $U_{Nu,95\%} = 2 \times u_{Nu}$. A more accurate proportionality factor can be obtained by calculating the Student’s-t value at a 95% CL with the resulting degrees of freedom. To calculate the resulting degrees of freedom, it is necessary to apply the Welch-Satterthwaite formula [1], for which the number of degrees of freedom of each primary quantity needs to be known. In the particular case of temperature measurements, the approach to obtain the corresponding degrees of freedom is presented in detail in this chapter to show a practical example of calculation of the Student’s-t value.

2.2. Case A: Rectangular ribbed channel

2.2.1. Geometry and experimental setup

The ribbed channel considered in the present work was previously investigated by Mayo et al. [7]. It is shown schematically in Figure 2.1 and was tested in stationary conditions. It consists of a 1,390 mm long rectangular channel, with a width equal to 138 mm and a height equal to 153 mm. Eight ribs with a squared cross section of $15 \times 15 \text{ mm}^2$ are placed on one of the walls of the channel, perpendicularly to the main flow direction, with a pitch equal to 150 mm. The ribs are manufactured in PMMA (Poly-methyl methacrylate, or acrylic glass) and are fixed to the channel with two screws on the lateral walls. The resulting hydraulic diameter is equal to 145 mm, the channel aspect ratio is equal to 0.90, the rib pitch to height ratio is equal to 10, and the blockage ratio is equal to 9.8%. The working fluid is air provided by a blower operating at ambient conditions. The mass flow is controlled by tuning the blower speed, and monitored by measuring the pressure drop across a Venturi nozzle. Before entering the ribbed channel, the air crosses a settling chamber of $240 \times 240 \text{ mm}^2$ equipped with honeycombs and screens in order to suppress secondary flows and reduce the turbulent fluctuations. A contraction of 240 mm length and with an area ratio equal to 2.73:1 is installed between the settling chamber and the ribbed channel to provide a smooth transition to the test section. The Reynolds number (based on the bulk velocity and the hydraulic diameter) is equal to 53300; the molecular Prandtl number is equal to 0.71. More details about the experimental facility can be found in Refs. [7, 8]. The wall temperature and Nusselt number distributions are studied in the space between the 6th and 7th ribs, presented in Figure 2.1 together with the selected frame of reference. Its origin is located at the intersection of the downstream wall of the 6th rib with the bottom wall, in the symmetry plane. The x axis corresponds to the stream-wise direction, the y axis to the wall-normal direction, and the z axis to the span-wise direction. The lateral and top walls of the channel are built in acrylic glass

in order to provide optical access to the investigated surface area. The bottom wall is a block of 1390×138×30 mm³ made of extruded polystyrene (XPS) on which an Inconel heating foil is glued.

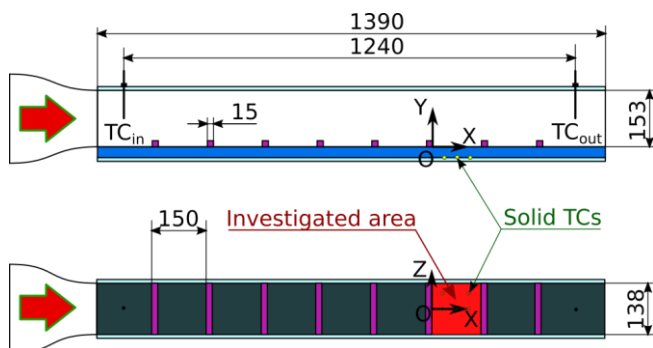


Figure 2.1 - Schematic of the rectangular ribbed channel (dimensions given in mm)

A matt black paint is applied on top of the Inconel foil in order to minimize light reflections. It is then covered by a layer of TLCs with a nominal active temperature between 35 and 55°C (308.15 and 328.15K). Finally, the ribs are placed on top of the TLC layer. A sketch of the cross section and the layers applied on the ribbed wall is shown in Figure 2.2. The foil is heated up by Joule effect, applying enough power to activate the TLCs on the wall. A SLR camera is placed at 45° from the normal to the plate to capture the color distribution reflected by the TLC layer. The illumination is provided by a set of two fluorescent lamps placed in parallel with the ribbed wall. The optical setup, also represented schematically in Figure 2.2, has been selected following the recommendations given by Cukurel et al. [9]. The calibration and experiments are carried out in a dark room in order to avoid the effect of additional sources of light. The first step of the data analysis consists in obtaining the wall temperature distribution by characterizing the color map of the TLC layer, acquired by the SLR camera.

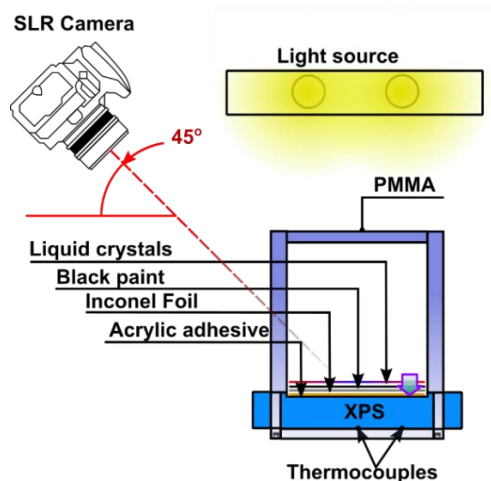


Figure 2.2 - Optical setup and cross section sketch of the channel

The relationship between the TLC color distribution and the temperature is obtained through a prior calibration. Secondly, the heat transfer distribution is calculated by applying an energy balance to finite surface elements, considering that the heat generated by Joule effect is transferred by convection to the flow, by radiation to the surrounding elements, and by conduction to the wall. In order to monitor the fluid temperature, two thermocouples are placed at the inlet and outlet of the test section, separated by 1,240 mm, as represented in Fig. 2.1. Five additional thermocouples are installed on the external surface of the XPS block to allow the computation of the heat loss by conduction.

2.2.2. TLC calibration and wall temperature uncertainty

The measurement of the wall temperature is performed by mapping the color distribution over the surface. The color is mapped by obtaining the Hue (H) from the RGB values captured by the camera at each pixel of the picture as:

$$H = \frac{1}{2\pi} \tan^{-1} \left[\frac{\sqrt{3}(G - B)}{2R - G - B} \right] \quad (2.4)$$

In order to relate the Hue distribution with the wall temperature, a calibration is performed in a separate rig, following the linear temperature gradient method, fully described in [9]. The optical setup during the calibration is exactly the same as in the channel facility, including the relative distances and angles with respect to the analyzed surfaces. The calibration rig consists of an aluminum plate on which the matt black paint and the TLCs are applied in the same way as in the investigated ribbed channel, in order to obtain identical layer characteristics. One of the sides of the plate is heated up by means of an electrical resistance, whereas the opposite one is cooled down by water flowing in a serpentine channel. This generates a linear temperature profile, characterized with 9 thermocouples embedded in the plate, just below its surface. The present analysis considers the x direction as the one corresponding to the temperature gradient, and the y direction perpendicular to it, along the plate width. During the calibration, the SLR camera acquires the Hue distribution along the calibration plate $H(x, y)$. In order to obtain a direct relationship between temperature and Hue, $H(x, y)$ is firstly averaged in the y direction to obtain the mean Hue distribution along the plate,

$$\bar{H}(x) = \frac{1}{l_y} \int_0^{l_y} H(x, y) dy \quad (2.5)$$

where l_y is the dimension of the plate in the y direction. Then $\bar{H}(x)$ is interpolated by means of a cubic spline curve with 5 knots, denominated $\bar{H}_{sp}(x)$ in this analysis. Once $T(x)$ and $\bar{H}_{sp}(x)$ are found, the TLC calibration curve $T = T(\bar{H}_{sp})$ is directly obtained (shown in Figure 2.3). The secondary ordinate axis in Figure 2.3 represents the reduced temperature, $(T - T_i)/\Delta T$, where T is the measured temperature, T_i the nominal activation temperature of the TLCs, and ΔT the TLCs nominal range. In the present work, T_i is equal to 305.15K and ΔT is equal to 20 K. The relationship between

Hue and reduced temperature is used to allow the comparison with TLCs at different nominal temperatures. The error bars shown in Figure 2.3 represent the temperature measurement uncertainty at a 95% Confidence Level (CL) for a given value of the measured Hue and therefore, for a given temperature. The sources of error considered in the present analysis are described in the following paragraphs.

2.2.2.1. Systematic error due to the digital resolution of the SLR camera ($b_{T,RGB}$)

The camera provides RGB values between 0 and 255 in each of the pixels, with a resolution of 8 bits. The quantization errors [6] for the RGB components are:

$$b_R = b_G = b_B = \frac{255/2^{8+1}}{\sqrt{3}} \quad (2.6)$$

Consequently, the systematic error due to the resolution of the camera is

$$b_{T,RGB} = \sqrt{\left(\frac{\partial H}{\partial R} \cdot b_R\right)^2 + \left(\frac{\partial H}{\partial G} \cdot b_G\right)^2 + \left(\frac{\partial H}{\partial B} \cdot b_B\right)^2} \cdot \frac{\partial T}{\partial H} \quad (2.7)$$

where $\partial H/\partial R$, $\partial H/\partial G$ and $\partial H/\partial B$ are obtained from equation (2.4), and evaluated with the RGB values at each point of the calibration plate. $\partial T/\partial H$ is evaluated from the calibration curve (Figure 2.3) with the local value of H and T . The degrees of freedom of the RGB values are considered equal to infinity, and therefore, the degrees of freedom of the systematic error due to the digital resolution, is $\nu_{b_{T,RGB}} = \infty$.

2.2.2.2. Systematic error in the determination of the temperature distribution along the calibration plate due to the limited accuracy of the thermocouples embedded in the calibration plate (b_{Tprof})

The thermocouples employed in this case were connected to an in-house thermocouple module and calibrated in a liquid bath against an Isotech 935-14-61 Reference Platinum Resistance Thermometer (RPRT) with an accuracy of 0.01 K (95% CL). During the calibration, 13 points are employed to fit a linear law relating the thermocouple voltage and the temperature. In this way, the number of degrees of freedom of b_{Tprof} , $v_{b_{Tprof}}$, is 11 (13 minus the number of parameters used in the fitting [1, 6, 10]). The maximum systematic standard uncertainty of these thermocouples is estimated at 0.08 K in the range on which they were calibrated, and therefore $b_{T,prof} = 0.08\text{K}$. The correlation with the RPRT and repeatability of the thermocouple read-outs over 7 days were in fact below b_{Tprof} at laboratory conditions

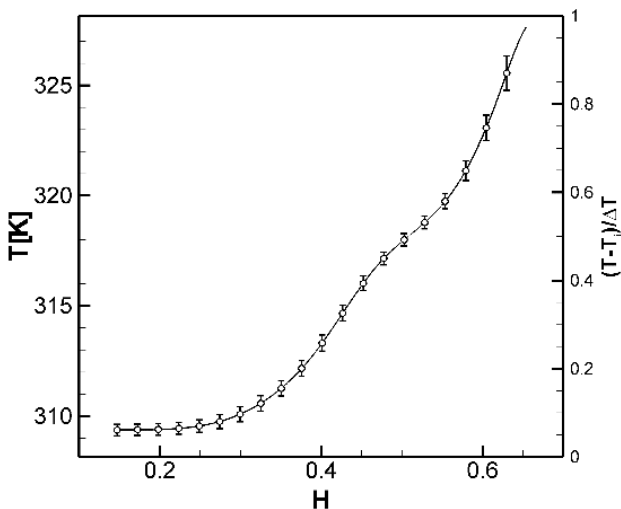


Figure 2.3 - TLC calibration curve

2.2.2.3. Random error due to the determination of the temperature profile along the calibration plate (s_{Tprof})

Since the temperature profile is obtained by interpolating linearly the temperature given by the nine thermocouples along the plate, there exists a fit error. This error, s_{Tprof} , is calculated as

$$s_{Tprof} = \sqrt{\frac{\sum_{i=1}^{i=9} (T_{TC_i} - T(x_i))^2}{9 - 2}} \quad (2.8)$$

where T_{TC_i} is the temperature given by each of the thermocouples, and $T(x_i)$ is the temperature corresponding to the obtained temperature profile at the location of each thermocouple x_i . The x direction represents the direction of the main temperature gradient in the plate. In the present case, $s_{Tprof} = 0.11$ K, whereas $\nu_{s_{Tprof}} = 9 - 2 = 7$.

2.2.2.4. Random error due to the spatial non uniformity of the Hue distribution along the plate ($s_{T,\delta H}$)

Along the calibration plate, the temperature varies linearly in the x direction. Along the plate width (y direction), the temperature is not fully uniform. This is firstly due to the non-perfect temperature distribution, and secondly, to the typical grainy distribution of the Hue value given by the TLCs. $s_{T,\delta H}$ is calculated as

$$s_{T,\delta H} = s_{H,\delta H} \cdot \frac{\partial T}{\partial H} = \frac{1}{\sqrt{M_y}} \sqrt{\frac{1}{M_y - 1} \sum_1^{M_y} (H(x, y_i) - \bar{H}(x))^2} \cdot \frac{\partial T}{\partial H} \quad (2.9)$$

where M_y is the number of points of the Hue distribution in the y direction, equal to 900 in this work. The degrees of freedom of the random error due to the spatial non-uniformities is $\nu_{s_{T,\delta H}} = M_y - 1 = 899$.

2.2.2.5. Random error due to the fitting of the original Hue distribution to a spline curve ($s_{T,fit}$)

The Hue distribution along the calibration plate is fitted to a cubic spline curve with six segments, and the fit error is calculated similarly to eq. (2.9):

$$s_{T,fit} = s_{H,fit} \cdot \frac{\partial T}{\partial H} = \sqrt{\frac{1}{N_x - (5 + 4)} \sum_1^{N_x} \left(\bar{H}(x_i) - \bar{H}_{sp}(x_i) \right)^2} \cdot \frac{\partial T}{\partial H} \quad (2.10)$$

where N_x is the number of points of the Hue distribution in the x direction, equal to 2250 in this case. The degrees of freedom in the spline fit are equal to N_x minus the number of knots (5) plus 4, and therefore, $\nu_{s_{T,fit}} = 2241$.

Figure 2.4 represents the temperature measurement standard and expanded uncertainties (95% CL), as well as the value of standard uncertainties due to the considered error sources, as a function of the temperature. In the range $308\text{K} < T < 320\text{K}$, the expanded uncertainty presents values between 0.30 and 0.37 K, whereas for $T > 320\text{K}$, the expanded uncertainty reaches values as high as 0.81K. On the other hand, the averaged value of the expanded uncertainty shown in Figure 2.4 is equal to 0.45 K. The average value of the degrees of freedom is equal to 109, leading to a Student's t value for 95% CL equal to 1.98. At 318.15K, the degrees of freedom are equal to 17, resulting in a Student's t value for 95% CL equal to 2.11. The increase of the measurement uncertainty at high temperatures is mainly due to the given quantization error of the RGB values and the increase of temperature sensitivity with

respect to Hue. Therefore, it seems convenient to operate the TLCs at temperatures below 320 K in the present setup. Figure 2.5 shows the Pareto chart of the systematic and random error contributions to the temperature measurements, for a temperature level equal to 318.15 K, and in average. At 318.15 K, the sensitivity of the Hue with respect to the RGB values is the minimum, resulting in a very low value of the systematic error due to camera digital resolution. At this temperature, the largest uncertainty sources are the systematic and random errors in the determination of the temperature profile along the calibration plate. Possible actions to reduce these uncertainties are the use of more accurate thermocouples and the increase of the accuracy in the localization of the thermocouples within the calibration plate. However, if the TLC measurement uncertainty needs to be reduced all over the active range, cameras with larger dynamic range must be used in order to reduce the measurement uncertainty due to the quantization of the RGB levels.

2.2.3. Heat transfer data reduction and uncertainty estimation

The methodology to obtain the heat transfer - Nusselt number distributions on the ribbed wall is similar to the one explained by Çakan [11]. The first step to measure the Nusselt number is to calculate the Joule heating power per unit area, q_J , that is applied to the Inconel foil. A power supply provides the voltage and current intensity needed to increase the foil temperature. In practice, the voltage is varied until the foil temperature (typically monitored with a surface thermocouple) reaches a level around 318 K, where the uncertainty is the minimum. In the present experiments, q_J is obtained as a function of the voltage generated by the power supply, V_{GEN} , for which an in-situ calibration is needed. In the calibration, the values of the current (I) and voltage (V_S) across a given surface (S) of the Inconel foil are acquired, for different levels of V_{GEN} :

$$q_J = \frac{V_S \cdot I}{S} = f(V_{GEN}) \quad (2.11)$$

In this case, the heated surface is the one represented in grey in Figure 2.2, with an area equal to $1.39 \times 0.138 \text{ m}^2$. Details about the connection between the generator and the heater are given in Ref. [8]. The voltages at the generator's output and across the heated surface are monitored with two voltmeters, whereas the current is measured by means of a standard clamp type current meter. On the other hand, the uncertainty introduced by any of the dimensions involved in the analysis is negligible in the present work. In total, 15 calibration points were fitted to obtain the relationship $q_J [\text{W/m}^2] = 5.9965 \cdot V_{GEN}^2 [\text{V}^2]$, with a fit error (considered as random uncertainty) equal to 0.13 W/m^2 . The systematic error due to the voltage and current in the calibration is calculated to be equal to 2.7 W/m^2 . The value of q_J was equal to 518.1 W/m^2 during the experiments. The temperature effects on q_J have been neglected in the present analysis, due to the low dependency of the resistivity with respect to the temperature, and the similar foil temperatures during the calibration and experiments. Next, the conduction losses to the solid ribbed wall are calculated point by point as

$$q_{cond} = \frac{k_{XPS}}{t_{XPS}} (T_w - \bar{T}_{w,ext}) \quad (2.12)$$

where k_{XPS} is the conductivity of the XPS (equal to $0.033 \text{ W/(m}\cdot\text{K)}$), t_{XPS} the thickness of the XPS (equal to 0.03 m), T_w the wall temperature given by the TLCs, and $\bar{T}_{w,ext}$ the average temperature given by the thermocouples placed on the external face of the XPS. A 1-D analysis of the heat transfer through the solid layers represented in Fig. 2.2, has indicated that the temperature difference across the layers is negligible with respect to the measured value at the wall. A 3-D heat transfer model was considered to account for the lateral conduction losses at each point (Ref. [8]), but the difference in the results was negligible. During the experiments, T_w presented an overall value equal to 316.4 K , and $\bar{T}_{w,ext}$ was equal to 300.1 K . On the other hand,

the standard deviation of the temperatures given by the 5 external thermocouples was equal to 0.57K. For this reason, the uncertainty in the $\bar{T}_{w,ext}$ measurement needs to consider not only the error due to the calculation of the mean value (equal to $0.57/5^{1/2}$ K), but also the error due to the assumption of a uniform value (equal to the typical temperature non-uniformity, 0.57 K). Therefore, the random uncertainty due to the external wall temperature is equal to $0.57+0.57/5^{1/2}=0.82$ K.

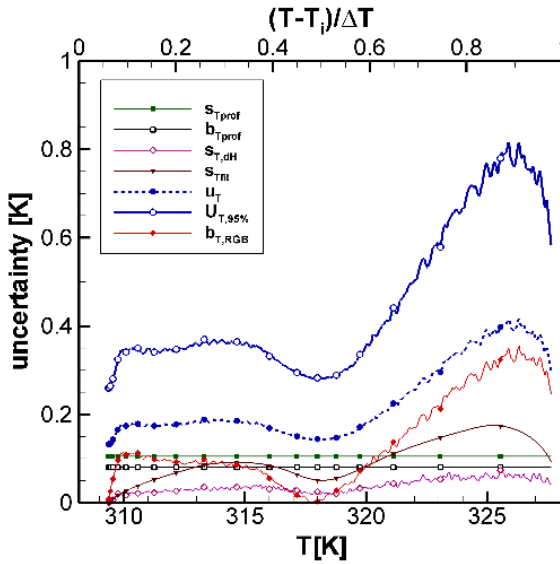


Figure 2.4 - TLC uncertainty as a function of the measured temperature

Additionally, the systematic error in $\bar{T}_{w,ext}$ is equal to the thermocouple systematic uncertainty, that is, 0.08K. On the other hand, the uncertainty related to T_w is given in Figure 2.7, whereas the uncertainty due to t_{xPS} is neglected. The radiation heat losses need also to be calculated in order to be able to carry out the heat flux balance in the heating foil. The radiation heat losses per unit area are calculated point by point as

$$q_{rad} = \varepsilon_{TLC} \sigma (T_w^4 - T_s^4) \quad (2.13)$$

where ε_{TLC} represents the TLCs emissivity, σ the Stefan-Boltzmann constant, and T_s the temperature of the surrounding walls. In the present analysis, the ratio of the radiation heat losses to the Joule heating per unit area is about 14.5%, which implies that any uncertainty in q_{rad} may have a large impact in the evaluation of the convective heat transfer. The value of the TLCs emissivity has been taken equal to 0.90, similarly to [12, 13].

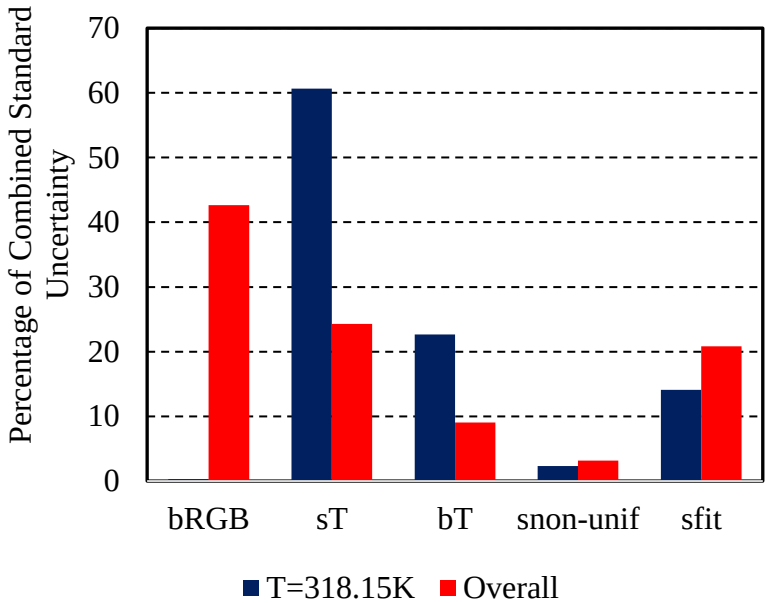


Figure 2.5 - Pareto chart of the systematic and random uncertainty component contributions to the combined standard uncertainty of the temperature measurement by means of TLCs

Nevertheless, larger values of ε_{TLC} are reported in literature, such as 0.96 in [14]. For this reason, a systematic error $b_{\varepsilon_{TLC}}$ equal to 0.06 has been considered as an uncertainty source in the analysis. The value of T_s has been also considered as uniform, and equal to the fluid inlet temperature (303.8 K). This value would

correspond to the minimum possible wall temperature, since it represents the fluid ambient temperature. Larger values are expected due to the convective heat transfer from the hot boundary layer near the ribbed wall to the cold lateral and top walls. In this way, the maximum value of T_s would correspond to the wall temperature near the bottom wall. In order to estimate the random uncertainty due to the measurement of T_s , a uniform probability distribution, with minimum and maximum values respectively equal to 303.8K and 314.3K, has been chosen. Since the minimum value has been selected to represent T_s , a truncation error is introduced. The corresponding random uncertainty has been calculated similarly to [6] (section B.7.2): $s_{T_s} = (314.3 - 303.8)/3^{1/2} = 6.1\text{K}$. A better estimation of s_{T_s} could be achieved by sampling the temperature distribution on the non-heated walls, for instance with a set of thermocouples, and obtaining the real wall temperature distribution. The systematic uncertainty due to T_s is taken equal to the systematic uncertainty in the temperature measurement by means of the present thermocouples, that is, $b_{T_s} = 0.08\text{ K}$. The uncertainty on the Stefan-Boltzmann constant has been neglected. Once q_J , q_{cond} and q_{rad} are known, the convective heat transfer per unit area is calculated point by point as:

$$q_{conv} = q_J - q_{cond} - q_{rad} \quad (2.14)$$

where q_J is assumed to be uniform on the surface due to the low variance of the heater resistivity with respect to the temperature. The heat transfer coefficient is calculated as:

$$h = \frac{q_{conv}}{T_w - T_{ref}} \quad (2.15)$$

where T_{ref} is the reference temperature defining the driving temperature potential, $T_w - T_{ref}$. The reference temperature is calculated by interpolating between the

values of the inlet and outlet thermocouples (Fig. 2.2), at the location of the investigated surface area. Since the systematic uncertainty of the temperatures given by the inlet and outlet thermocouples is estimated to be equal to 0.08K, the systematic uncertainty in the measurement of T_{ref} is equal to 0.08K. Finally, the Nusselt number is calculated as:

$$Nu = \frac{h \cdot D_H}{k_f} \quad (2.16)$$

where D_H is the hydraulic diameter, and k_f the air thermal conductivity at the reference temperature. The expression employed to calculate the air thermal conductivity at the film temperature is the one given by Stephan and Laesecke [15], which is reported to have a standard random uncertainty equal to 1.1%. Additional systematic uncertainties in the evaluation of k_f need to be considered, due to the uncertainty on T_{ref} .

Figure 2.6 shows the Nusselt number distribution along the symmetry line ($z = 0$) of the investigated surface area represented in Figure 2.1, together with the local expanded uncertainty. Two sets of data have been represented: the reference one corresponding to a first set of experiments, and the second one corresponding to repeatability tests after one week of use of the facility, with the wall heated about 9 hours a day. It can be seen that the data collapses very well, and therefore the TLC aging is not appreciable. The surface area-averaged Nusselt number in the investigated region is equal to 200, with an average expanded uncertainty equal to ± 34 (95% C.L.). Therefore, the overall relative measurement uncertainty is equal to 17%. A summary of the data with the respective standard systematic and random uncertainties is given in Table 2.1, whereas the Pareto chart summarizing the impact of the systematic and random uncertainties is represented in Figure 2.7.

The sensitivities θ_i shown in Table 2.1 correspond to the partial derivative of the Nusselt number with respect to the given parameters. In the case of q_J , for instance, the sensitivity is calculated as:

$$\theta_{q_{Joule}} = \frac{\partial Nu}{\partial q_J} = \frac{\partial Nu}{\partial h} \cdot \frac{\partial h}{\partial q_{conv}} \cdot \frac{\partial q_{conv}}{\partial q_J} \tag{2.17}$$

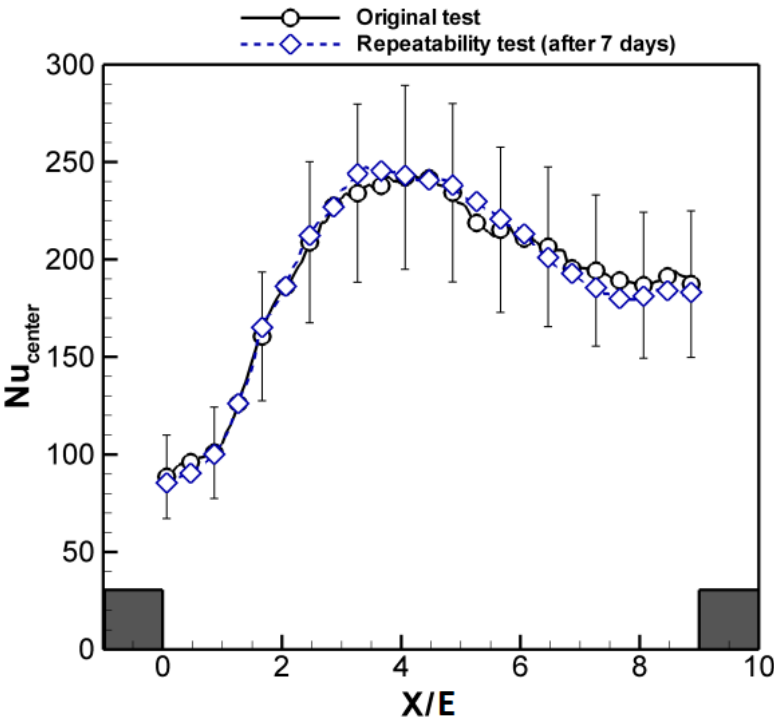


Figure 2.6 - Nusselt number axial profile and corresponding expanded uncertainty at the symmetry line of the investigated area

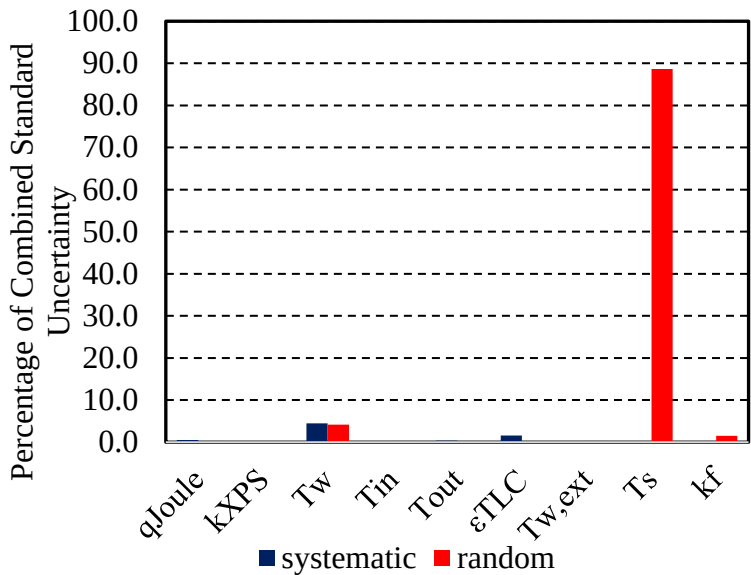


Figure 2.7 - Pareto chart of the systematic and random uncertainty component contributions to the combined standard uncertainty of the Nusselt number measurement in the rib-roughened channel by means of TLCs

where the partial derivatives can be calculated from eqs. (2.14), (2.15) and (2.16). It can be observed that the largest contribution to the measurement uncertainty of the Nusselt number is due to the random error related to the temperature of the surrounding walls used in the determination of the radiation losses: a 89% of the Nusselt number uncertainty is caused by the random uncertainty in T_s . If this random uncertainty is reduced e.g. from 6.1 K to 1K, the final Nusselt number expanded uncertainty would be reduced from the current 17% to a value of 6%.

Symbol	Units	Nominal Value	Absolute	Absolute	Absolute	Absolute	Absolute
			Systematic	Random		Systematic	Random
			Standard	Standard		Standard	Standard
			Uncertainty,	Uncertainty,		Uncertainty	Uncertainty
			b_{Xi}	s_{Xi}		$(b_{Xi}\theta)^2$	$(s_{Xi}\theta)^2$
q_J	W/m ²	518.1	2.67	0.13	0.46	1.54	0.0036
k_{XPS}	W/(m·	0.033	5×10 ⁻⁴	0	1.76	7.76×10 ⁻⁷	0
	K)						
T_w	K	316.5	0.15	0.15	-23.71	13.22	12.34
T_{in}	K	303.8	0.08	6.5×10 ⁻³	4.97	0.16	1.04×10 ³
T_{out}	K	304.4	0.08	6.5×10 ⁻³	13.53	1.17	7.73×10 ³
ε_{TLC}	-	0.9	0.06	0	-36.44	4.78	0
$T_{w,ext}$	K	300.9	0.08	0.82	0.51	0.0017	0.18
T_s	K	304	0.08	6.06	2.67	0.05	262.03
k_f	W/(m·	0.0261	1.5×10 ⁻⁵	3×10 ⁻⁴	-7340.69	0.01	4.44
	K)						

Table 2.1 - Summary of the overall data for the ribbed channel [$Nu = 200 \pm 34$ (95% C.L.)]

The second and third largest contributions correspond to the systematic and random uncertainties in the measurement of the wall temperature, about 4.5% and 4.2% of the Nusselt number expanded uncertainty. From this analysis, it clearly appears that special attention must be paid to the radiative heat transfer losses, and to T_s in particular. On the other hand, Table 2.1 provides the sensitivities of the Nusselt number uncertainty with respect to the primary measured quantities. This is particularly useful to rank the different primary variables regarding their potential impact on the final uncertainty, for the given nominal values. In the present case, the sensitivity of the uncertainty with respect to that of the fluid thermal conductivity is clearly the largest compared with the rest of the primary variables. Nevertheless, the

relatively low uncertainty in k_f does not lead to a large Nusselt number error distribution. The opposite occurs for T_s : the sensitivity is low with respect to the rest of primary variables, but the large uncertainty in T_s results in a large Nusselt number uncertainty. Table 2.1 also demonstrates that the emissivity of the TLC layer and the wall temperature are variables to be carefully controlled due to the high sensitivity of the Nusselt number with respect to these parameters.

2.3. Case B: Cylindrical smooth pipe

2.3.1. Geometry and experimental setup

The second case considered in this chapter concerns the heat transfer in a cylindrical smooth pipe; the entire wall of the test section is heated up. This setup allows the suppression of the radiation heat transfer losses, leading to a reduction of the corresponding measurement uncertainty. Figure 2.8 illustrates the experimental facility in which the test section is installed. The mass flow is generated by a centrifugal fan placed at the outlet of the facility, operated in a suction mode. In this way, the inlet flow is not affected by the highly turbulent/swirling flow produced by the fan. The working fluid (air) first enters the inlet duct, of 2250 mm length and 150 mm diameter, corresponding to a pipe length to diameter ratio equal to 15. The test section is an acrylic tube of 150 mm internal diameter and 980 mm length, placed after the inlet duct, where the heating foil is installed. The test section is divided in two semi-circular parts, in order to allow the preparation of the heating surface and the calibrations required before the tests.

A layer of TLCs is first applied on the inner wall of the pipe, superseded by the black matt paint and the Inconel foil. A sketch of the different layers in the cross section is given in Fig. 2.8 (b). In this way, the color distributions of the TLC layer can be observed from outside when the Inconel foil is heated, as shown in Figure 2.9.

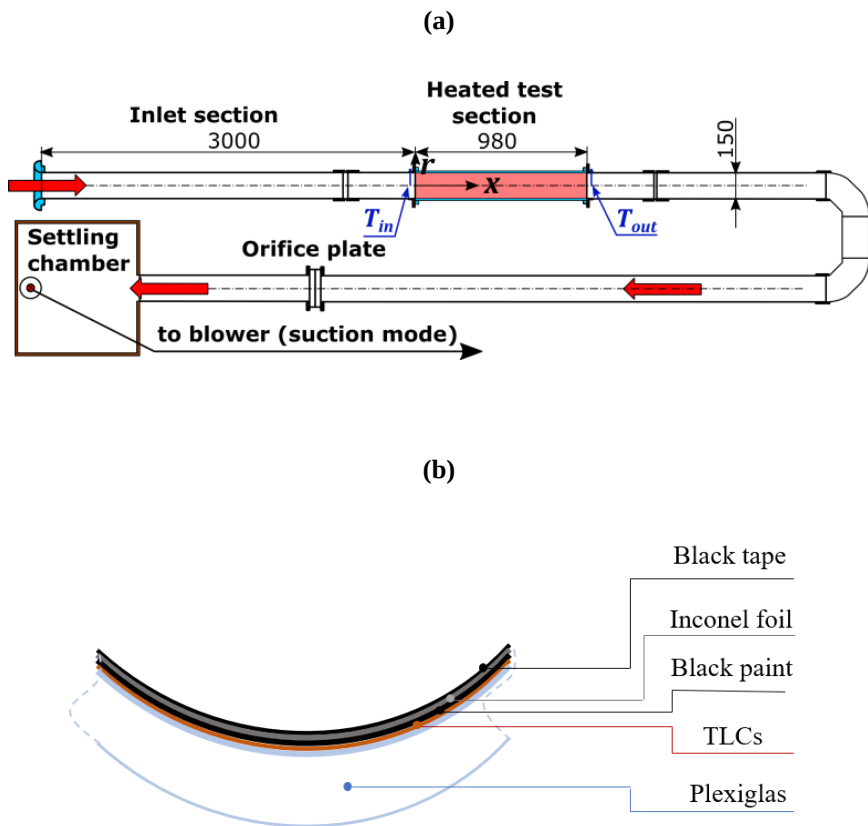


Figure 2.8 - Schematic of the test facility for the investigation of the heat transfer and uncertainty in the smooth pipe (dimensions given in [mm]) (a), layers in the cross section of the duct, from Ref. [16](b)

A black-painted box surrounds the experimental setup to avoid the influence of external light sources. Two thermocouples are placed at the inlet and outlet of the test section in order to obtain the reference temperature potential to calculate the transfer coefficient along the measurement surface area, located 600 mm downstream of the inlet of the test section. The mass flow exiting the test section is directed through a U-bend towards the exit pipe. An orifice plate is placed in the exit pipe at a distance of 30 pipe diameters downstream the U-bend to monitor the mass flow. The rotational

speed of the fan is varied in order to adapt the pressure drop across the orifice plate to the required Reynolds number. Downstream of the mass flow meter, the fluid is finally discharged into a settling chamber, to which the blower is directly connected. The inner surface temperature distribution (T_w) of the test section is measured by using the same type of TLCs as in the previous case, active from 35 to 55°C (308.15 to 328.15 K). To increase the wall temperature and to activate the TLCs, a DC voltage is applied to the Inconel foil, inducing the heat transfer between the foil and the fluid.

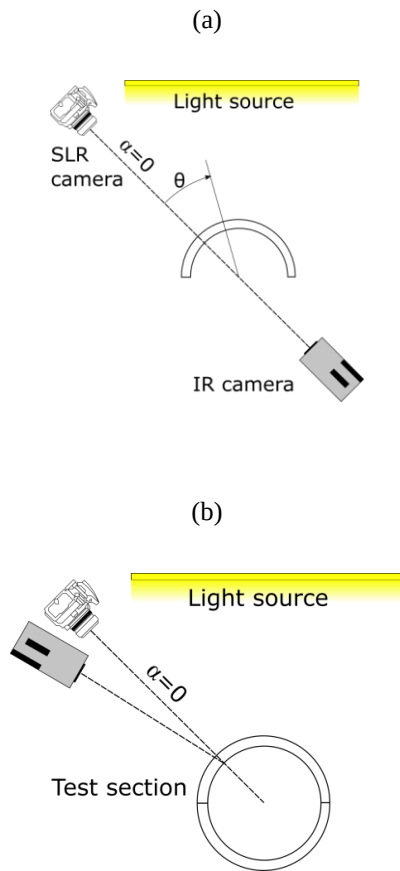


Figure 2.9 - Optical setup in the smooth pipe facility: TLCs in situ calibration
(a), experimental case (b)

The optical access through the transparent Plexiglas wall allows a SLR camera to acquire RGB intensities scattered by the TLCs. The SLR camera is focused on the test section wall from a distance of about 850 mm with an angle of 45° with respect to the horizon, as sketched in Figure 2.9. Further details on the experimental facility and the optical setup can be found in Ref. [16]. The part of the image processed is called Region of Interest (ROI) which consists on a rectangle extending 80 mm in the stream-wise direction and 30° in the azimuthal direction. In this case, an in-situ calibration was carried out in order to relate the Hue with the temperature. To account for the dependency of the TLCs color response with respect to the view angle, the calibration curves are organized depending on the viewing angle, and therefore $H = H(T, \theta)$, where θ is the azimuthal angle. As in the previous case, the heat balance in each surface element of the heating foil is carried out to evaluate the heat transfer by convection at the wall.

2.3.2. TLC calibration and wall temperature uncertainty

The function that relates the TLCs color properties and the temperature is obtained by performing a calibration in situ. During the calibration, the lower half of the cylinder is removed, to provide optical access to infrared thermography, as shown in Figure 2.9. An IR camera is employed to measure the temperature field on the inner surface of the pipe. By applying different current intensities to the heating foil, the inner surface temperature varies, as well as the color distribution reflected by the TLCs. On the other side, the SLR camera records the behavior of the TLCs layer on the inner surface of the acrylic glass. With this approach, it is possible to match locally the temperature measured by the IR camera with the colors observed by the SLR camera in the ROI. From the RGB distribution in the picture, the Hue value is obtained according to eq. (2.4). To avoid the reflection of any spurious thermal signal from the metal foil to the IR camera, a high emissivity vinyl covers the inner surface of the cylinder. The IR camera itself has been calibrated both for the Plexiglas and for the

vinyl materials, providing respectively ± 0.5 K and ± 0.4 K expanded uncertainties (95% C.L.) in the calibrated range. The final calibration curves of the TLCs are built for each row of the ROI and they consist in spline curves fitted to 18 calibration points. The calibration curves for the angles $\theta = -15^\circ$, 0° and 15° are represented in Figure 2.10, together with the corresponding measurement uncertainty at a 95% C.L.

It is important to note that, for a given Hue, a maximum difference of 2 K is detected between different calibration curves. The absence of an in-situ calibration accounting for the investigation surface curvature ($\pm 15^\circ$) would therefore result in a LCTG measurement error of ± 1 K that is 5% of the liquid crystals active temperature range (20K). On the other hand, the maximum values of the expanded uncertainty result to be between 0.93 K (for $\theta = -15^\circ$) and 0.98 K (for $\theta = 15^\circ$), again about 5% of the measurement range. In this case, the error sources taken into account are:

2.3.2.1. Systematic uncertainty due to the digital resolution of the SLR camera ($b_{T,RGB}$)

Since the SLR camera employed is the same as in the previous case, $b_{T,RGB}$ is calculated considering eqs. (2.6) and (2.7). However, the partial derivatives in eq. (2.7) are calculated from the calibration curve at each azimuthal location θ . Similarly, the degrees of freedom are considered equal to infinity, $\nu_{T,RGB} = \infty$.

2.3.2.2. Systematic uncertainty due to temperature measurement uncertainty by the IR camera ($b_{T,IR}$)

The IR camera has been calibrated up to a temperature of 328 K in a separated rig consisting in a heated plate equipped with surface thermocouples, similarly to Ref. [17]. During the calibration of the IR camera, $b_{T,IR}$ was found to be equal to 0.2 K, with $\nu_{b_{T,IR}} = 17$.

2.3.2.3. Random uncertainty due to the spatial variation of the temperature provided by the IR camera in the stream-wise direction ($s_{T,IR}$)

During the calibration, the temperature distribution is averaged in the stream-wise direction for a given azimuthal angle. $s_{T,IR}$ is calculated by computing the standard deviation of the temperature given by the IR camera for a fixed α , and considering the number of points in the stream-wise direction, P :

$$s_{T,IR}(\alpha) = \sqrt{\frac{1}{P-1} \sum_{i=1}^P (T(\alpha, x_i) - \bar{T}(\alpha))^2} \quad (2.18)$$

with $P = 137$. In this case, the degrees of freedom are $\nu_{s_{T,IR}} = P - 1 = 136$.

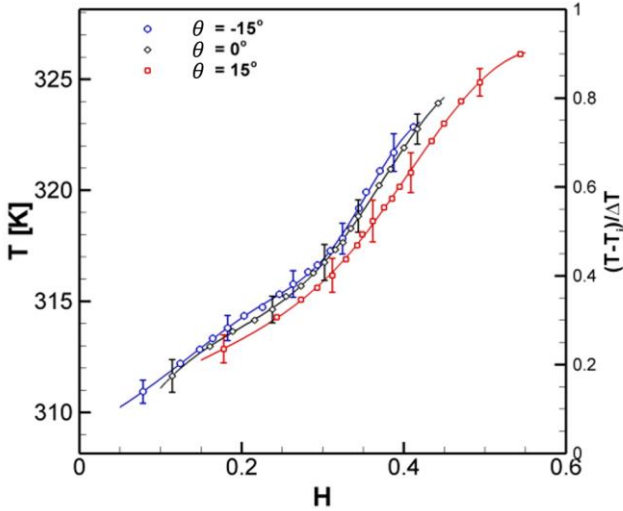


Figure 2.10 - TLC calibration curves for the angles $\Theta = -15^\circ$, 0° and $+15^\circ$, with the respective expanded uncertainty

2.3.2.4. Random uncertainty due to the spatial variation of the Hue in the streamwise direction for a given angle ($s_{T,\delta H}$)

Similarly to the previous paragraph, the value of the Hue is averaged in the streamwise direction in order to find the representative Hue for a given azimuthal location. Therefore, $s_{T,\delta H}$ is calculated from the standard deviation of the Hue value for a fixed θ :

$$s_{T,\delta H}(\theta) = \sqrt{\frac{1}{Q-1} \sum_1^P (H(\theta, x_i) - \bar{H}(\theta))^2} \cdot \frac{\partial T}{\partial H} \quad (2.19)$$

Q is the number of points in the x direction (equal to 1368). In this case, the degrees of freedom are $\nu_{s_{T,\delta H}} = Q-1 = 1367$.

2.3.2.5. Random uncertainty due to the fit ($s_{T,fit}$)

In this case, a cubic spline curve with 4 knots is employed to fit 18 calibration points to obtain the final relationship $T = T(H; \theta)$. The number of corresponding degrees of freedom are $\nu_{s_{T,fit}} = 18-4+4=18$.

This calibration methodology provides a resulting overall expanded uncertainty on the temperature measurement between ± 0.56 K and ± 0.84 K, depending on the row position inside the ROI (Figure 2.11). The expanded uncertainty also depends on the temperature, and it results to be ± 0.58 K in the red region at 313 K ; ± 0.75 K in the green region at 318.9 K and ± 0.62 K in the blue region at 324 K, values evaluated at $\theta = 0^\circ$. The Pareto chart (Figure 2.12) shows the overall standard uncertainties at an angular position $\theta = 0$, as well as the average in the entire region of interest. It can be observed that the largest uncertainty is caused by the non-uniformity of the Hue distribution within the region of interest, causing almost 50% of the total uncertainty. A possible way to reduce this uncertainty would be the division of the domain also in

the x direction, similarly to the work of Kodzwa and Eaton [12]. The second most important uncertainty source is the systematic error due to the temperature measurement by the IR camera, representing about 30% of the total uncertainty. In this case, the error is difficult to be reduced, due to the limited accuracy of the present IR cameras.

2.3.3. Heat transfer data reduction and uncertainty estimation

The methodology to obtain the heat transfer coefficient and Nusselt number distributions along the smooth pipe wall is the same as the one presented in the previous case. A summary of the nominal values and corresponding uncertainties is shown in Table 2.2, for the case of a Reynolds number equal to 78330. Firstly, it is necessary to calculate the heat transfer per unit area by Joule effect (eq. (2.11)). In this case, the zone considered to perform the calibration of the relationship $q_J = f(V_{GEN})$ is equal to one half of the cylinder surface, that is, $\pi \times 0.98 \times 0.075 \text{ m}^2$. Twelve points were used to find the relation $q_J [\text{W/m}^2] = 5.1841 \cdot V_{GEN}^2 [\text{V}^2]$, with a random uncertainty due to the fit error equal to 0.32 W/m^2 . The systematic uncertainty due to the limited accuracy of the instruments measuring the voltage and the intensity across the calibrated surface is equal 4.1 W/m^2 . For a Reynolds number equal to 78330, the nominal value of q_J is equal to 991.8 W/m^2 .

The conduction heat transfer losses per unit area are calculated locally as

$$q_{cond} = \frac{k_p}{t_p} (T_{TLC} - T_{w,ext}) \quad (2.20)$$

where k_p is the thermal conductivity of the acrylic glass (PMMA, $0.19 \text{ W/(m}\cdot\text{K)}$), t_p the thickness of the tube wall (0.01 m), T_{TLC} is the temperature of the internal face of the tube (given by the TLCs), and $T_{w,ext}$ is the temperature of its external face. In this

case, the temperature distribution along the external wall is measured with an IR camera as depicted in Figure 2.9. Therefore, the conduction losses can be computed locally, at any stream-wise and azimuthal location.

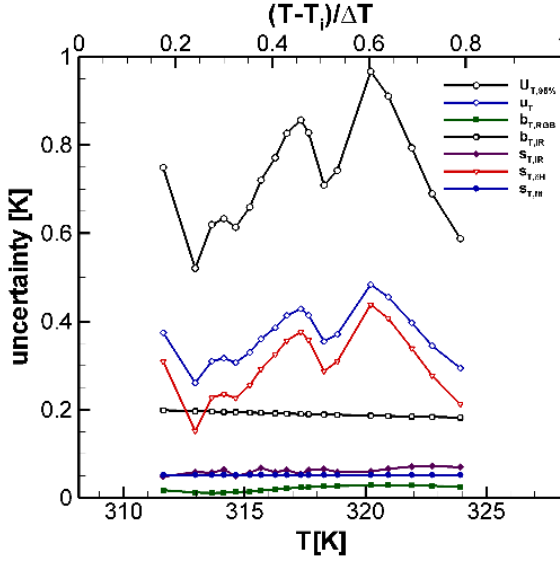


Figure 2.11 - TLC uncertainty as a function of the measured temperature for $\Theta = 0^\circ$

The overall nominal value of T_{TLC} at $Re = 78330$ is equal to 320.1 K, with systematic and random uncertainties equal to 0.23 K and 0.34 K, respectively. The overall nominal value of the external temperature is equal to 308.2 K, with overall systematic and random uncertainties equal to 0.31 K and 0.03 K, respectively. On the other hand, the lateral conduction losses have been estimated by applying a finite volume approach to the heating foil. They resulted to be negligible with respect to the conduction losses in the radial direction (eq. (2.20)). In this case, the radiative heat losses are negligible, since the entire internal wall surface is heated up. Therefore, the convective heat transfer per unit area is calculated at any stream-wise and azimuthal position as

$$q_{conv} = q_J - q_{cond} \quad (2.21)$$

which allows the calculation of the heat transfer coefficient by using eq. (2.15). The reference temperature used in eq. (2.15) is obtained by interpolating linearly the temperature from the inlet to the outlet of the test section at the region of interest. The thermocouple system employed in this case consists in a National Instruments NI-9214 module that was also calibrated against the RPRT in a liquid bath. The calibration revealed a systematic measurement uncertainty equal to 0.04 K in the calibration range. As a result, the systematic error in the reference temperature T_{ref} is equal to 0.04 K. The correlation with the RPRT and repeatability of the thermocouple read-outs were checked over seven days, showing to be below the estimated systematic error (0.006 K at 298 K, and 0.03 K at 313 K).

Finally, the Nusselt number is calculated by applying eq. (2.16), calculating the conductivity of the fluid (k_f) with the Stephan-Laesecke correlation [15] at the reference temperature, with a resulting nominal value equal to 0.0256 W/(m·K). The systematic uncertainty corresponding to k_f is due to the accuracy in the measurement of the reference temperature, whereas the random uncertainty is due to the 1.1% fit error in the correlation by Stephan and Laesecke [15]. On the other hand, an additional systematic uncertainty was considered in the calculation of the uncertainty in the wall temperature measurement. The actual temperature of the wall in contact with the fluid T_w is not the one given by the TLCs (see Fig. 2.8 (b)), and a systematic error is introduced if T_w is assumed to be equal to T_{TLC} .

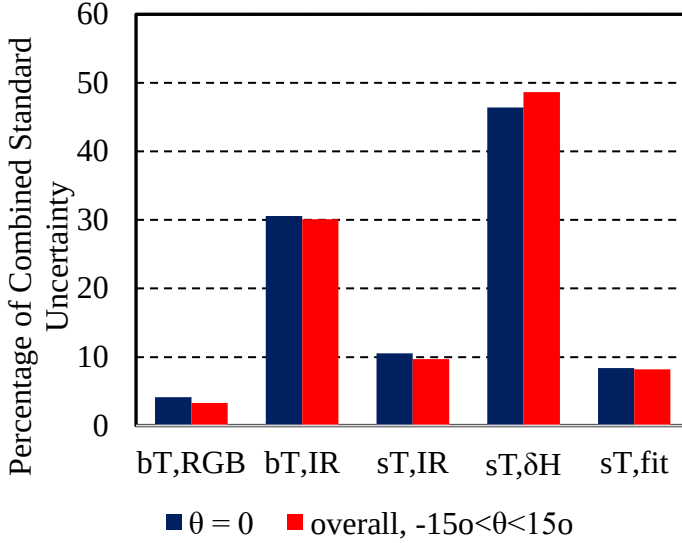


Figure 2.12 - Pareto chart of the systematic and random uncertainty component contributions to the combined standard uncertainty of the wall temperature measurement in the smooth pipe by means of TLCs

Considering a 1-D heat transfer model for the layers presented in Fig. 2.8 (b), the difference between T_w and T_{TLC} is about 0.15 K. Note that this temperature difference is proportional to the conduction heat losses, which are significant in this case. In order to introduce this systematic error at the standard uncertainty level of the analysis, its value has been divided by 2. The resulting standard systematic error in the measurement of T_w is therefore $b_{T_w} = \sqrt{\left(\frac{0.15K}{2}\right)^2 + b_{T_{TLC}}^2}$, where $b_{T_{TLC}}$ is the systematic error of the temperature given by the TLCs. For a typical value of $b_{T_{TLC}}$ equal to 0.23 K, the systematic error in T_w , b_{T_w} , is equal to 0.24 K.

Symbol	Units	Nominal Value	Absolute	Absolute	Absolute	Absolute	Absolute
			Systematic	Random		Systematic	Random
			Standard	Standard		Standard	Standard
			Uncertainty,	Uncertainty,		Uncertainty	Uncertainty
			b_{xi}	s_{xi}		Contribution,	Contribution,
					θ_i	$(b_{xi}\theta)^2$	$(s_{xi}\theta)^2$
q_J	W/m ²	991.8	4.13	0.19	0.216	0.799	1.65×10 ⁻³
k_P	W/(m·K)	0.195	5×10 ⁻⁴	0	257.05	1.65×10 ⁻²	0
T_w	K	320.1	0.24	0.34	6.08	2.13	4.28
T_{in}	K	292.6	0.04	6.5×10 ⁻³	-2.95	1.39×10 ⁻²	3.68×10 ⁻⁴
T_{out}	K	293.4	0.04	6.5×10 ⁻³	-3.35	1.79×10 ⁻²	4.74×10 ⁻⁴
$T_{w,ext}$	-	308.2	0.034	0.31	4.25	2.12×10 ⁻²	1.70
k_f	W/(m·K)	0.0256	1.50×10 ⁻⁵	2.82×10 ⁻⁴	-6.42×10 ³	9.28×10 ⁻³	3.27

Table 2.2 - Summary of the overall data for the smooth pipe [$Nu = 166 \pm 12$ (95% C.L.)]

At a Reynolds number equal to 78330, the surface area-averaged Nusselt number is equal to 166. The overall uncertainty at a C.L. of 95 % is equal to 12, which leads to a 7.3% relative uncertainty. Figure 2.13 displays the Pareto chart comparing the contributions of the primary measurement magnitudes on the overall Nusselt number uncertainty, showing that the largest contribution to the final uncertainty is due to the wall temperature measurement by the TLCs. The systematic uncertainty in the measurement of T_w amounts to a 19% of the total uncertainty, whereas the random uncertainty reaches 41%. It is important to note that the contribution of the random uncertainty in k_f to the Nusselt number uncertainty is equal to 31%. This is again due to the large sensitivity of the Nusselt number with respect to the fluid thermal conductivity, and despite the relatively low uncertainty in k_f , the final contribution is considerable. Additionally, the sensitivity of the Nusselt number uncertainty with respect to the thermal conductivity of the wall is very high, for the present experimental setup and nominal values. The conductive heat transfer losses are about

a 23% of the Joule heating power, hence any uncertainty in the parameters ruling the conductive heat losses can be translated into an increase of the final uncertainty. A possible solution to cope with the high sensitivity with respect k_p is to reduce conductive heat losses, for instance by increasing the external temperature (ambient temperature) artificially.

Finally, additional measurements at Reynolds numbers ranging from 19500 to 59600 were carried out in order to compare the measured Nusselt number values with those given by the Dittus-Boelter correlation given by McAdams [18] at the same Reynolds and Prandtl numbers:

$$Nu_0 = 0.023Re^{0.8}Pr^{0.4} \tag{2.22}$$

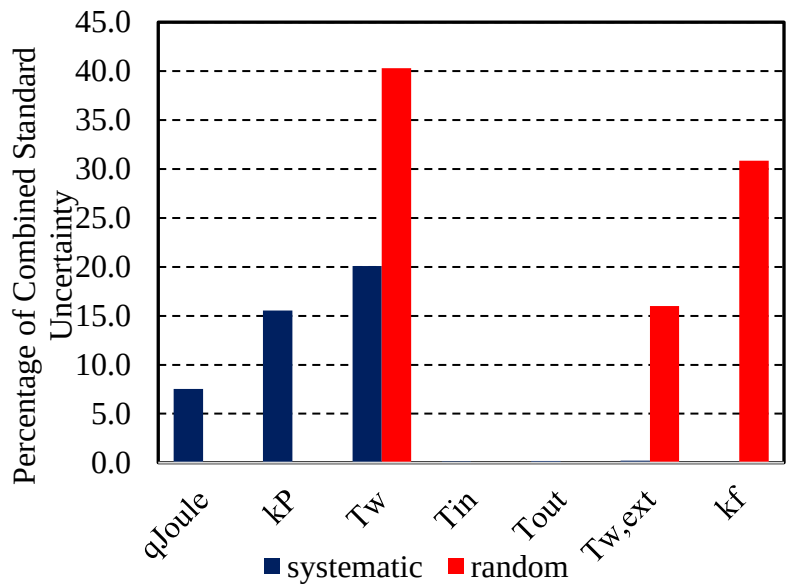


Figure 2.13 - Pareto chart of the systematic and random uncertainty component contributions to the combined standard uncertainty of the Nusselt number measurement in the smooth pipe by means of TLCs

<i>Re</i>	<i>Nu</i>	<i>U_{Nu,95%}</i>	<i>Nu₀</i> (eq. (2.22))
19520	53.6	±6.6	54.4
39990	95.5	±8.3	96.4
59600	132.8	±9.8	132.9
78330	165.6	±11.7	165.4

Table 2.3 - Measured Nusselt number and corresponding expanded uncertainty in the smooth pipe as a function of the Reynolds number

The averaged values in the region of interest for every Reynolds number and the corresponding uncertainty are compared with the correlation in Table 2.3, showing an excellent agreement. It is important to note that the relative uncertainty increases as the Reynolds number decreases. This is due to the fact that the heat transfer by convection is reduced, and the conduction losses get more and more important. Again, it is clear that the reduction of the heat losses would drive to a significant reduction in the global measurement uncertainty.

2.4. Conclusions

A detailed analysis of the uncertainty sources in temperature and heat transfer measurements by means of TLCs has been carried out. Two different experimental setups, both designed for fundamental experimental heat transfer research, were evaluated: the first considering a ribbed rectangular channel, and the second a circular smooth pipe.

In the first experimental setup, a calibration rig was employed to obtain the relationship between the Hue value derived from the TLCs and the corresponding wall temperature distribution. The average temperature expanded uncertainty (95% CL) is estimated to be equal to 0.45 K (2.3% of the TLC active range) in the activation range

of the TLCs, although the maximum value of the expanded uncertainty reaches 0.81 K (4% of the TLC active range) when the wall temperature is near the TLC maximum working temperature. The main source of error comes from the quantization error of the SLR camera, followed by the error due to the determination of the linear temperature profile in the calibration rig and the fit error of the color-temperature relationship to a spline curve. When measuring the Nusselt number in the ribbed channel at a Reynolds number equal to 53300 by means of the calibrated TLCs, a relative overall expanded uncertainty of 17% (95% CL) was reached. This high value is mainly due to the estimation of the radiation losses, which cause almost 90% of the total uncertainty. A second major uncertainty contributor is the temperature measurement given by the TLCs, representing a 6% of the total uncertainty. This analysis demonstrates the importance of controlling the heat losses and their distribution in the test section in order to reduce the measurement uncertainty.

In the cylindrical pipe test case, an in-situ calibration was performed to consider the angular dependency of the color received by the SLR camera. Instead of using thermocouples to monitor the temperature of the TLC surface during the calibration, an IR camera was employed. In this example, the largest contribution to the uncertainty of the TLC temperature evaluation is given by the non-uniformity of the color in the stream-wise direction, followed by the uncertainty due to the accuracy of the IR camera. The resulting expanded uncertainty in the temperature measurement under these conditions was between 0.56 and 0.84 K, depending on the azimuthal location. During the study of the Nusselt number, the heat losses by radiation were suppressed due to the heating of the entire pipe wall, which lead to a drastic uncertainty reduction: the Nusselt number relative uncertainty was equal to 12.3% (95% CL) at a Reynolds number equal to 19500 and 7% (95% CL) at a Reynolds number equal to 78300. In this case, the main uncertainty is related to the wall temperature measured by the TLCs and the thermal conductivity of the air obtained from literature correlations.

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3

Effects of 2-D and 3-D helical inserts on the turbulent flow in pipes

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Abstract

A constant search for more efficient heat exchangers motivates the use of innovative turbulators. Original flow-field measurements were performed with stereo-PIV to gain a better understanding of the role of 3-D wall obstacles in pipe flows. Such effects are compared with those of 2-D helically corrugated tubes, which are already provided in the literature. This work further investigates the impact of both continuous (2-D corrugation) and discontinuous (3-D corrugation) obstructions on the surrounding velocity field. The analysis of the flow structures is made by comparing two types of helicoidal turbulators. The continuous geometry has a pitch-to-diameter-ratio of $p/e=11$ and an inclination angle of 80° . An obstacle height-to-diameter-ratio of $e/D=3.6\%$ is used, which is constant throughout the helix-wise direction. The discontinuous geometry is equivalent to the continuous one except that it has a varying obstacle height throughout the helix. The reattachment and redevelopment of the flow are determined by time-averaged velocity analysis, at $Re=20000$. Notably, the aim of this study is to examine the effects of the discontinuity in the tube corrugation on the flow separation/reattachment, turbulence statistics, vorticity (ω) and strain-rate (S_{xy}) of the flow particles, and compare them with a previously studied continuous turbulator. As a result, the discontinuous rib has a lower impact on the pressure losses, on the azimuthal swirl, and on the turbulent kinetic energy generation. In the field close to the wall, the changing height of the discontinuous rib generates a turbulent kinetic energy and an azimuthal velocity, respectively, 50% and 35% lower than the continuous configuration. The irregularity of the discontinuous obstacles provides a reduced skin friction coefficient (C_f) which is 0.57 times the one given by the continuous 2-D helical turbulator. On the other hand, the observed less intense wall-bounded turbulence levels associated with the 3-D helical turbulator suggest a lower local heat transfer enhancement.

3.1. Introduction

Tubular reactors are widely applied in the chemical industry. In steam cracking processes, those type of reactors are employed to break saturated hydrocarbons into lighter chemical structures. As an example, during thermal cracking of hydrocarbons for olefin production, tubular chemical reactors with short residence times and high heat transfer rates are used. Moreover, for maximizing light olefin selectivity, the radial temperature gradients should be as small as possible [1]. One possible solution is the use of turbulators, as shown in Figure 3.1, with the purpose of disrupting the boundary layer at the wall. The subsequent reattachment of the shear layer increases the local Nusselt number (Nu) when compared with systems without wall corrugation [2, 3, 4, 5].

The wall-bounded swirling flow and the impingement of the reattaching boundary layer decrease the wall temperature, leading to a decrease in coke deposition on the wall and a longer operating time. Moreover, due to the ribbed structure, flow is ejected from the hot wall to the cooler center of the channel, reducing radial temperature and concentration gradients, positively influencing selectivity on specific products during the chemical reactions [6]. In short, the implementation of these structures allows a higher production of ethylene and propylene [7]. The increase in heat transfer rate is also beneficial in other fields, e.g., turbomachinery. High-pressure turbine blades are cooled by internal ribbed channels [8].

Recently, considerable attention has been devoted to optimizing the geometry of the turbulators for cooling applications [10]. The implementation of such obstacles provides higher local wall heat transfer but also generates a significant increase in pressure drop. Hence, it is essential to find the best performing geometry, which minimizes pressure losses [11].

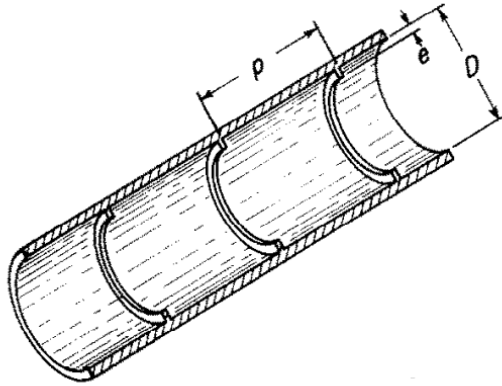


Figure 3.1 - Example of a ribbed tube [9]

Zhu et al. [12], Van Cauwenberge et al. [6] and Campet et al. [13] studied the turbulent flow in a helicoidally ribbed pipe, for petrochemical applications, using Wall-Resolved Large Eddy Simulation (WR-LES) with a kinetic energy sub-grid scale model. In the work mentioned above, the effects of a turbulator on both the heat transfer and the friction losses are shown by the authors. Experimental investigations on the effects of wire coils, with different configurations, on the heat transfer and friction coefficient were provided by several authors [14, 15, 16]. The turbulent structures in the vicinity of the obstacles were already investigated with planar PIV by Wang et al. [17] and Coletti et al. [18] for flows in rectangular cross-section channels. These authors highlighted the flapping shear layer phenomenon; the latter generates a significant intermittence inside the whole flow field. Cardwell et al. [19], using time-resolved Particle Image Velocimetry (PIV), investigated the turbulent flow field inside a rectangular cross-section channel with transverse square obstacles. The turbulence statistics were provided for flow regimes in the Reynolds number (Re) range 2,500-20,000 with a blockage ratio of 12% and a pitch-to-rib height of 8. The authors showed that the rib wake region is governed by a periodic process of bursting vortices which generate an intermittent ejection of the structures from the wall to the core of the flow. Considering that the flow in helically ribbed

pipes has a non-negligible velocity component in the azimuthal direction, stereo-PIV (s-PIV) must be considered to provide useful 3D data. The use of two cameras allows to obtain the out-of-plane velocity component, and hence to lower the perspective error as explained by Prasad [20]. A turbulent water flow in a smooth pipe with a Reynolds number equal to 5300 was measured with stereo-PIV by van Doorne & J. Westerweel [21]. Their stereo-PIV snapshots consisted of velocity fields in the circular cross-section of the pipe with the two cameras looking at angles of $+45^\circ$ and -45° to the light sheet. The results, in terms of mean flow and turbulence statistics, perfectly matched those provided by Direct Numerical Simulations (DNS) results [22].

The effects of a helically corrugated pipe on both the flow field and heat transfer were shown experimentally in the work of Mayo et al. [23] at the flow regime of $Re=20000$. The authors characterized with stereo-PIV the flow physics and statistics, analyzing their influence on the local Nu . The 2-D turbulator investigated by the authors is a continuous helicoidal wall obstacle used to enhance the local heat transfer with the drawback of generating more pressure losses. Therefore, there is a significant interest in finding new geometries which both improve the mixing inside the flow and enhance the convective heat transfer at the wall, affecting as little as possible the friction coefficient. The use of obstacles with a variable height through the helix pattern is a potential solution to the problem stated before. As a matter of fact, the turbulent boundary-layer theory [24] predicts that a surface with less roughness leads to lower friction. For this reason, this work aims to provide and analyze the pressure losses, the time-averaged flow field features and the turbulence statistics of a discontinuous helicoidal wall obstacle, utilizing stereo-PIV. The effects of the rib discontinuity on the friction coefficient, on the three-dimensional flow separation-reattachment phenomenon and the turbulent kinetic energy production are compared with those given by the continuous helix, already investigated by Mayo et al. [23] at a Reynolds number value equal to 20,000.

3.2. Experimental Methodology

The velocity measurements were conducted in the low-speed water tunnel (Figure 3.2) of the “TU - Heat and Mass Transfer Laboratory” at the von Karman Institute. The water is pumped into a settling reservoir where its level is kept constant by means of an overflow tube working as a by-pass. In such a way, the unsteadiness that the pump may introduce is avoided, and the total inlet pressure can be considered as constant. The water is discharged from the settling reservoir into the inlet duct; the latter has an internal diameter (D) of 150 mm and a length of 2250 mm. The choice concerning the pipe diameter size was taken to have a spatial resolution, which was enough to capture the turbulent structures generated by the obstacles. A more detail explanation is given in Appendix A. The resulting pipe length to diameter ratio is equal to 15, which allows the development of a turbulent flow in the test section [25]. A honeycomb is placed in the entry region of this development duct to suppress any swirling motion generated in the connection between the settling chamber and the duct entrance. The test section is made of acrylic glass to provide full optical access to the area of interest. It is placed downstream of the inlet duct. The first part is smooth, whereas, the second part is equipped with the obstacles. The turbulators of the test section consist of obstacles glued on the inner surface of a bare channel. The material of the turbulators is ABS (Acrylonitrile Butadiene Styrene). They are 3D-printed by stereolithography. Two different kinds of helical turbulators, named RIB-1 and RIB-2 in the following (Figure 3.3), are successively inserted in the test section and tested with stereo-PIV. RIB-1, which is the continuous helicoidal turbulator investigated by Mayo et al. [23], consists of a semi-circle section swept throughout a helical pattern tangentially to the inner surface of the test section. The helix has a pitch of 63 mm repeated for nine times, the angle of attack (i.e., the angle between the axial and helix-wise directions) is 80° , and the semi-circle section has a constant height-to-pipe diameter ratio of 3.6%. The other geometry (RIB-2) consists of wall obstacles glued

throughout the same helix path of RIB-1. The main difference is that the RIB-2 section's height of the semi-circles is changing along the helical pattern direction. (Figure 3.3). The maximum semi-circle height is the same as for RIB-1. This height decreases continuously to a zero value after an angular displacement of 11° in both the directions of the helix. There are six obstacles equally spaced over a single pitch of the helix, for a total of 9 full pitches. A detailed description of how the obstacles of RIB-2 are obtained is given in Chapter 4.

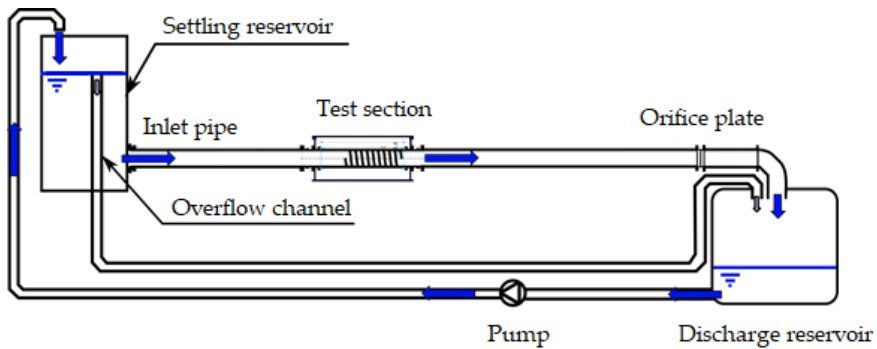


Figure 3.2 - Low-speed water tunnel layout [23]

The effects of the ribs on the main flow statistics are compared for a similar regime. The Reynolds number is defined with the bulk velocity (U_b) and the pipe diameter. Values of 24,400 and 21,100 for RIB-1 and RIB-2 are respectively applied. The stereo-PIV images are recorded for the flow region between the 7th and the 8th pitch. This zone is located at a distance of 20 pipe diameters downstream of the settling reservoir because the typical development length in low blockage ratio cooling channels is about 6 rib pitches [26]; the blockage ratio is defined as the height of the rib over the diameter of the channel. The latter is cut longitudinally in two parts. This allows the spatial calibration of the field of view for both cameras. The test section material (Plexiglas) has a refractive index of 1.49. In order to decrease the image distortions that the PIV would encounter with the use of air, whose refractive index is about 1.00, water was chosen as working fluid.

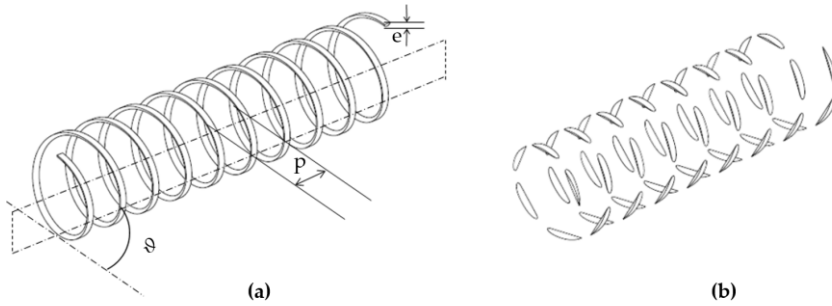


Figure 3.3 - Schematic of continuous and discontinuous helicoidal turbulators:
(a), RIB-1; (b), RIB-2

Despite this choice, there is a difference of 10% in the refraction index between the working fluid and the test section wall. The resulting image distortion is corrected with an in-situ calibration of the stereo-PIV system by means of a mapping function. In order to maintain a flat surface interface between the ambient air and the solid external wall of the facility, the test section is inserted into a transparent reservoir filled with water [21]. The mass flow rate is inferred by measuring the static pressure drop generated by an orifice plate, designed and located according to the ISO 5167 standard. The flow meter is positioned 20 pipe diameters downstream of the test section exit, and the orifice dimensions are chosen to minimize the relative measurement uncertainty. The latter is 1.5 % of the reading. To allow a control of the flow regime, a spherical valve is placed downstream of the orifice plate. The fluid is finally collected in a large discharge reservoir and driven again to the settling chamber by means of the pump.

The investigated area is illuminated by a double pulse Nd:YAG laser Quantel Tween BSL 200. It provides 70 mJ per pulse, which is about 33% of its nominal power. A telescope (focal length range of 300mm-2m), a cylindrical lens (20mm) and a mirror generate the focused light sheet (thickness of 1.5 mm) to cover the plane of interest between two obstacles. Two double-shutter CCD cameras with a

sensor of $2360 \times 1776 \text{ pix}^2$ are placed on the same side of the test section and equipped with Nikon Nikkor lenses of 50 mm focal length set to $f/8$. This PIV system layout results in image particle sizes in the range of 3-7 pixels, to minimize any error due to peak locking, which might influence, in particular, the turbulence statistics [27]. In order to maintain a focused image across the complete measurement area, a Scheimpflug mount is adapted to each camera. Even though the angle between the cameras optical axes and the horizon is often set at 45° [21] this value was reduced to 23° (Figure 3.4), as suggested by Lawson and Wu [28]. Generally, larger angles provide increased sensitivity to the velocity component perpendicular to the laser sheet. On the other hand, the uncertainty on the two other velocity components increases. For this reason, the angle between the cameras in the plane parallel to the pipe section was chosen lower than 90° . Besides, between the plane of the cameras and the normal to the laser plane, there is another angle of 15° (Figure 3.5) which allows better optical access because of the obstruction by the helix-shaped turbulators. The resulting overall Magnification Factor is about 29 pix/mm.

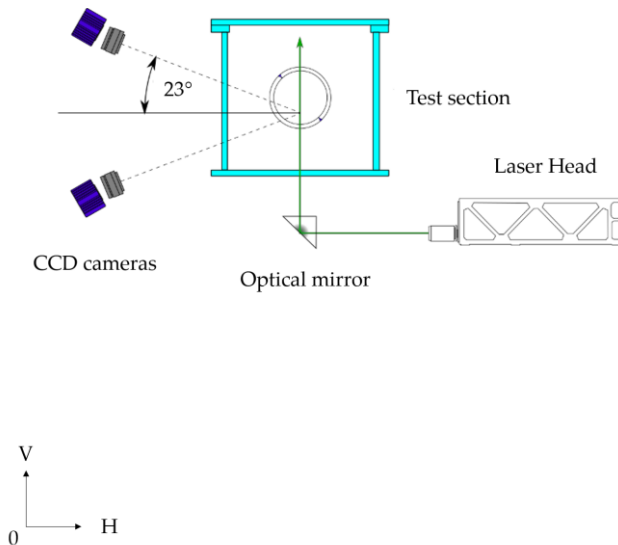


Figure 3.4 - Layout of the s-PIV equipment

The in-situ calibration of the image acquisition system is carried out by following the method proposed by Soloff et al. [29]. A dotted calibration grid (Figure 3.6) is put inside the test section and adjusted in the middle of the laser sheet. The dots have a radius of 1 mm, and they are displaced uniformly in x and y directions with a step of 2.5mm. This calibration grid is positioned by means of a linear translation stage equipped with a micrometer, whose resolution is 0.01 mm. The calibration plate is then displaced in two steps of ± 0.25 mm in the out-of-plane direction. The goal is to obtain a standard 3-D mapping function which matches the positions in the imaging system and real frames reference. The partial derivatives of the functions $F_{camera,1}^X$, $F_{camera,1}^Y$, $F_{camera,2}^X$ and $F_{camera,2}^Y$ with respect to x , y and z are then computed by the commercial software Davis 8.2 (LaVision), to retrieve the real particle displacement Δx , Δy , Δz by solving the following linear system [30]:

$$\begin{pmatrix} \Delta X_{camera,1} \\ \Delta Y_{camera,1} \\ \Delta X_{camera,2} \\ \Delta Y_{camera,2} \end{pmatrix} = \begin{bmatrix} \frac{\partial F_{camera,1}^X}{\partial x} & \frac{\partial F_{camera,1}^X}{\partial y} & \frac{\partial F_{camera,1}^X}{\partial z} \\ \frac{\partial F_{camera,1}^Y}{\partial x} & \frac{\partial F_{camera,1}^Y}{\partial y} & \frac{\partial F_{camera,1}^Y}{\partial z} \\ \frac{\partial F_{camera,2}^X}{\partial x} & \frac{\partial F_{camera,2}^X}{\partial y} & \frac{\partial F_{camera,2}^X}{\partial z} \\ \frac{\partial F_{camera,2}^Y}{\partial x} & \frac{\partial F_{camera,2}^Y}{\partial y} & \frac{\partial F_{camera,2}^Y}{\partial z} \end{bmatrix}_{z=0} \cdot \begin{pmatrix} \Delta x \\ \Delta y \\ \Delta z \end{pmatrix} \quad (3.1)$$

where $\Delta X_{camera,1}$, $\Delta Y_{camera,1}$, $\Delta X_{camera,2}$ and $\Delta Y_{camera,2}$ are the displacements obtained by the cross-correlation of the particles images. More in detail, Davis 8.2 solves the system of 4 linear equations in 3 unknowns using the normal equation. This mathematical tool is used in linear algebra to distributes the errors evenly over the 3 components [31]. Dividing Δx , Δy , and Δz by the time interval between consecutive frames, it is possible to compute the velocity components U , V , and W . To reach the statistical convergence, 1,500 PIV realizations are acquired at a frequency of 1 Hz, which is comparable to the lowest flow frequency (U_b/D). In

order to guarantee a maximum particle displacement of about ten pixels, the separation time between the two frames of each PIV realization is set at 2.3 ms.

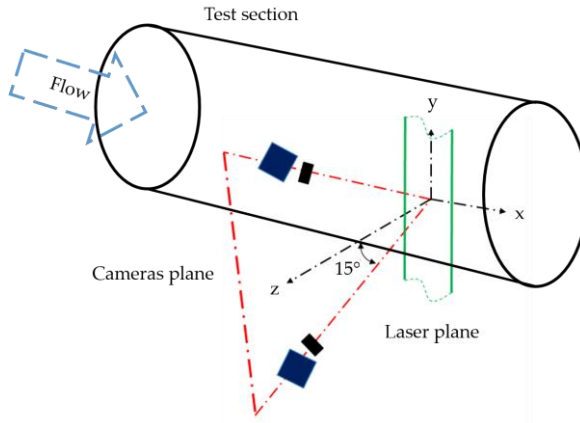


Figure 3.5 - Angle between the z-axis (perpendicular to the laser plane) and the plane of the cameras

The algorithm processes the images in three refinement steps, with a final interrogation window size of $32 \times 32 \text{ pix}^2$ and an overlap of 75% providing a resolution of $1.1 \times 1.1 \text{ mm}^2$. In this way, the total number of data points in space results to be 185×160 , respectively, in the axial and the radial direction. Fluorescent Chospheric polyethylene spheres (diameter of $10\text{--}45 \text{ }\mu\text{m}$) are used for seeding purposes. A 532 nm notch filter is placed in front of each camera sensor to cut off the laser light wavelength and to record only the light scattered by the fluorescent particles. This allows minimizing the background reflections and increasing the signal-to-noise ratio [32]. The concentration of the particles (density of 0.995 kg/dm^3) is about 20 g/m^3 in order to have at least ten particles per window. In Appendix B, the effects of the selected imaging parameters on the reconstructed error and correlation map are shown for both RIB-1 and RIB-2.

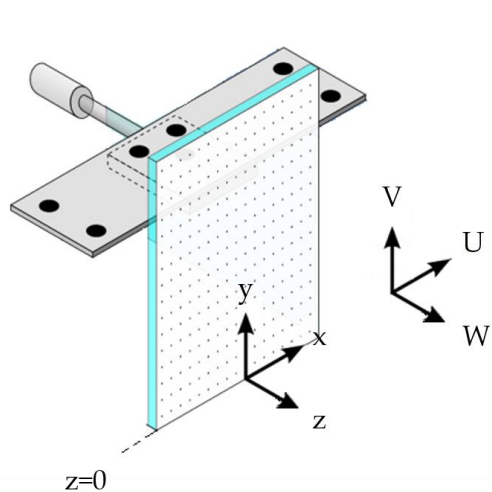


Figure 3.6 - Calibration plate and transverse system

The pressure losses measurements are conducted in a different experimental facility (Figure 3.7) with air as working fluid. The choice of air, rather than water, allows having a higher differential pressure across the turbulator. This is linked with the possibility to have pressure measurements with lower uncertainty. A centrifugal fan, which is placed at the outlet of this low-speed wind tunnel, generates the mass flow through the test section. The choice of operating in a suction mode allows having an inlet flow not affected by the turbulent swirling components produced by the fan. The air enters first the inlet section which corresponds to a pipe with a length of 20 channel diameters. Both the diameter of the pipe and the test section are the same as for the water facility. One k-type thermocouple is placed at the inlet of the test section in order to obtain a reference temperature of the flow. The mass flow exiting the test section goes through a U-bend, which connects the test section with the exit pipe. At this side of the facility, an orifice plate is placed, at a distance of 30 pipe diameters downstream the U-bend, to measure the mass flow. The required mass flow and Reynolds number are obtained by changing the rotational speed of the fan. Downstream of the exit pipe, the flow enters into a settling chamber to

which the blower is connected. The facility is equipped with one pressure tap at the inlet of the ribbed test section, and the other one placed at $7D$ far from the outlet. Although several configurations were tried for the measure of the pressure losses, no significant differences in the results were observed. At both the axial positions, the pressure taps are located at the top of the pipe, and they are connected to a high sensitivity differential pressure calibrator DPI 610 LP (Druck). This calibrator can measure the static differential pressure Δp_{static} in the range 0-250 Pa with an extended uncertainty of $\pm 0.05\%$ of the span.

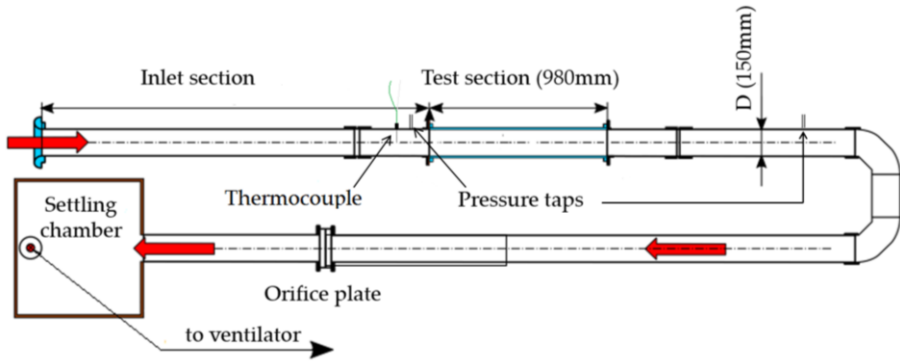


Figure 3.7 - Layout of the air facility employed for the pressure losses measurements

By solving the momentum balance considering the test section as the control volume, the wall shear stress averaged over the ribbed surface can be computed with the equation:

$$\bar{\tau}_{wall} = \frac{\Delta p_{static} \times R}{2 \times L} \quad (3.2)$$

where R and L are respectively the inner radius of the test section and the distance between the two wall pressure taps. In Appendix C, the evaluation criteria adopted for Δp_{static} and its assumptions are explained in detail. The surface averaged friction coefficient C_f is computed as follows:

$$C_f = \frac{\bar{\tau}_{wall}}{0.5 \times \rho \times U_b^2} \quad (3.3)$$

where ρ is the flow density, and U_b is the bulk velocity deduced from the mass flow measurements. To validate the skin friction evaluation of the two different turbulators, the same procedure is applied for the bare pipe, without any obstacles, and compared with the Blasius correlation for turbulent fully developed flows in smooth pipes [33]:

$$C_{f,smooth} = \frac{0.079}{Re^{0.25}} \quad (3.4)$$

3.3. Results and discussion

The surface averaged friction coefficients of the flow inside the bare channel are plotted in Figure 3.8 for several Reynolds numbers in the range $Re=20000-80000$. The results are close to the values obtained by using the Blasius correlation with a maximum deviation of 15%. The extended uncertainties on the C_f result to be 82%, 24.4%, 15.2% and 10.1% relatively for $Re = 20500, 40400, 60100$ and 80000 . The insertion inside the pipe of the turbulators generates a friction loss increase, as expected. The flow across the continuous obstacles (RIB-1) is characterized by a friction coefficient which is seven times higher than the one of the smooth pipe.

The same pattern is observed for the discontinuous geometry (RIB-2), where the skin friction coefficient is at a constant value of about four times the Blasius correlation predictions, at the same Reynolds number. The high uncertainties on C_f evaluation shown at $Re=20000$ are justified by the fact that, at this flow regime, the static pressure differences are 1.30 Pa, 1.85 Pa and 3.03 Pa for respectively the

smooth pipe, RIB-1 and RIB-2. As expected, the discontinuity of the turbulator RIB-2 alters the flow in such a way that a lower increase of the pressure losses is measured, when compared with the continuous geometry (RIB-1).

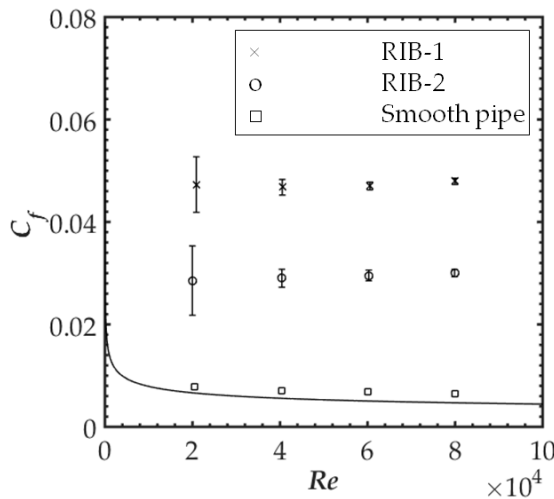


Figure 3.8 - Skin friction coefficient of the two turbulators as a function of the Re : solid black line, Blasius correlation; squares, smooth channel; crosses, RIB-1; circles, RIB-2

The two stereo-PIV experimental campaigns focus on the area between 40% and 100% of the radius since the primary interest is the effect of the ribs on the flow close to the wall. The three components U , V and W are respectively the projections of the velocity vector on X , Y and Z respectively (see Figure 3.6). The streamlines deduced from the time-averaged flow field, and the contours of the azimuthal velocity component, generated by RIB-1 and RIB-2 are compared in Figure 3.9. In both configurations, the presence of the obstacles forces the flow to separate and to impinge on the wall, generating swirling coherent structures. The reattachment of the shear layer is followed by a flow re-development, which consequently impacts the leading edge of the next obstacle. In the case of RIB-1, the reattachment is at a distance of about 4.15 ± 0.25 times the rib height (e) from the rib trailing edge. The

reattachment is defined as the position where the radial gradient of the axial velocity is equal to zero at the wall, where the shear stress, $\tau_{yx}|_{wall}$ has a zero value. A recirculation bubble, whose height is 1/10th of the pipe radius, and smaller vortices are generated across each obstacle. Since the height of RIB-2 changes along the helix-wise direction, the global blockage is expected to have a weaker influence on the separation. For this reason, the reattachment occurs more upstream than for RIB-1 (reattachment length of 3.5 times e), and the main coherent structure presents a smaller size. According to the authors Mayo et al. [23], the local peak of the Nusselt number between the obstacles is just downstream the reattachment point of the shear layer. Therefore, since the discontinuity of RIB-2 provides a smaller reattachment length, the heat transfer peak moves slightly closer to the 7th obstacle trailing edge if compared with RIB-1.

The main vortex in RIB-1 and RIB-2 is characterized by a positive azimuthal velocity, which is respectively about 15% and 5% of the bulk velocity for both configurations. The intensity of the azimuthal velocity component is higher in the presence of the continuous obstacle (RIB-1), where the highest values are confined between the wall and 85% of the pipe radius. For RIB-2, the recirculating bubble is bounded in a smaller region, closer to the wall. The peak value, for both the geometries, is about 25% of the bulk velocity and it is located just upstream of the obstacles, at $X/e \sim 9$, where e is the rib height, and X is measured from the rib trailing edge, see Figure 3.9. Generally speaking, in RIB-2, the portion of the fluid which is moving tangentially to the wall has a lower W magnitude since the helicoidal flow structures are broken by the discontinuity of the obstacle.

This phenomenon also is supposed to lower the wall heat exchange at the leading edge area of the 8th obstacle, since for RIB-2 the intensity of the velocity component in z direction (W) is lower than that one provided by RIB-1. The authors Gee & Webb [2] in their work show the dependency of the Stanton number with the angle of attack of the helicoidal turbulators, θ in Figure 3.3. According to their results, the

heat transfer increases with the helix angle ϑ at the expense of having increased pressure losses.

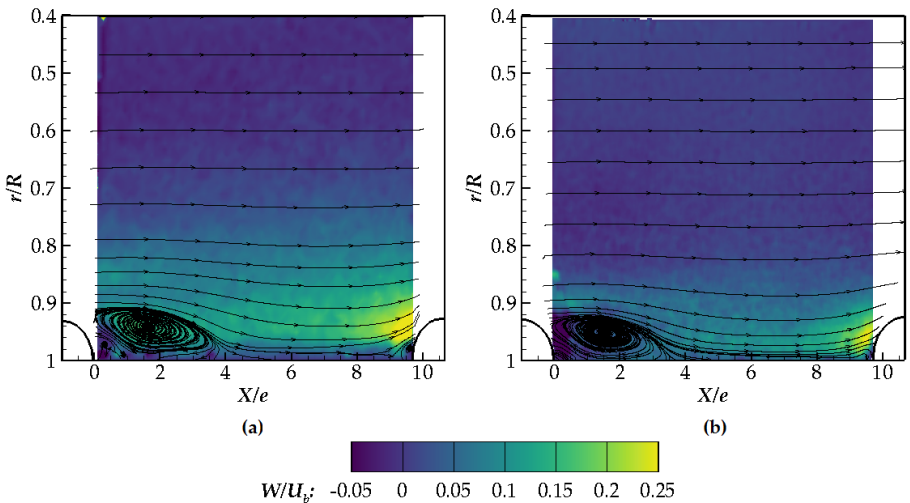


Figure 3.9 - Time-averaged azimuthal velocity (W) field at the plane $z=0$ between the 7th and the 8th obstacles: (a), RIB-1; (b), RIB-2

Therefore, the helical flow structures generated by RIB-2 break the continuity of the flow angle, potentially decreasing the surface heat transfer. It is also substantial to analyze how the discontinuity of RIB-2 influences the mean flow, in comparison with that one measured by Mayo et al. [23]. The normalized average axial velocity profiles are presented in Figure 3.10 for different stream-wise positions. The lower global blockage introduced by the discontinuity of RIB-2 decelerates less the flow axial velocity closer to the wall. From 70% of the radius towards the axis of the pipe, the axial velocity for RIB-2 is lower in comparison with the one for RIB-1. This is in accordance with the fact that the two experiments were done at similar Reynolds numbers. Figure 3.11 compares the azimuthal components of the velocity field (W) generated by the two geometries at different stream-wise locations. As

expected, the continuity of RIB-1 leads to a stronger W/U_b in the observed wall-normal region (between 70% and 100% of the radius).

Moreover, since the two generated flows between the obstacles of RIB-1 and RIB-2 have different velocity components and turbulent structures, the fluid particles rotate with a different intensity (Figure 3.12). Figure 3.12 shows the distribution, in the surrounding flow fields of RIB-1 and RIB-2, of the vorticity, defined as:

$$\omega = \frac{1}{2} \cdot \left(\frac{\partial u}{\partial y} - \frac{\partial v}{\partial x} \right) \quad (3.5)$$

The separated layer generates clockwise vorticity of the flow, whereas the shear between the rotating bubble and the wall produces counter-clockwise structures. It is also interesting to note that such fluid structures are not deformed in the $z=0$ plane with the same intensity, given by the x - y component of the dimensionless strain tensor shown in Figure 3.13. The x - y component of the strain rate tensor is defined as:

$$S_{xy} = \frac{1}{2} \cdot \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \quad (3.6)$$

Therefore, there is not a significant influence of the RIB-2 geometry on the gradients of the flow. Only one exception is closer to the leading edge of the 8th obstacle, where the vorticity is slightly higher for the flow generated in RIB-2 than for RIB-1. Obstacles in channel flows add a turbulence intensity which depends on the turbulator shape. Stronger coherent structures promote both the momentum and the energy transfer by means of the fluctuating velocity field. In this sense, the two turbulators are also compared in terms of first-order statistics, to understand how the discontinuity of RIB-2 provides a degree of influence.

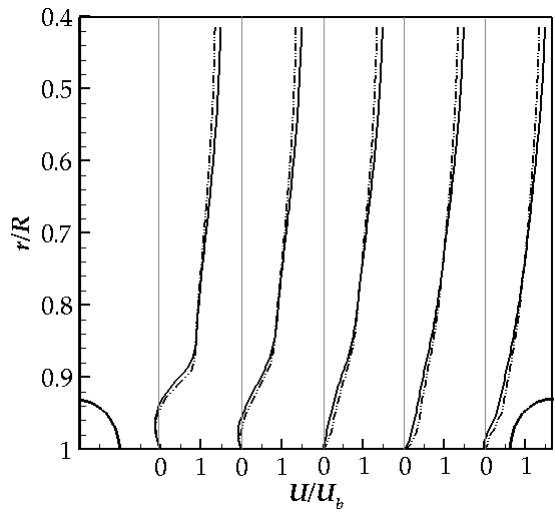


Figure 3.10 - Normalized profiles of the mean axial velocity at five different axial positions between the 7th and the 8th obstacles: black solid, RIB-1; black dash-dot-dot, RIB-2

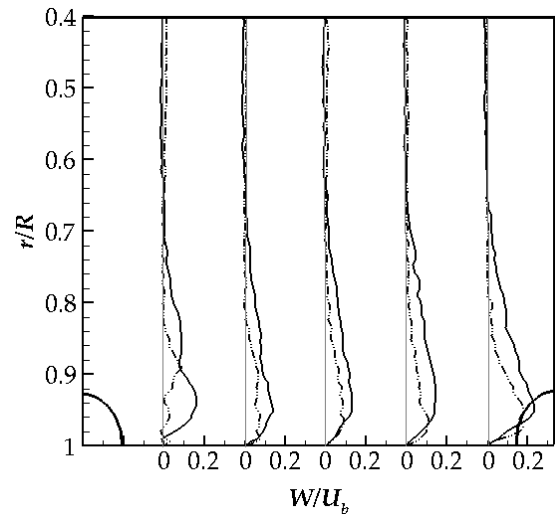


Figure 3.11 - Normalized profiles of the mean azimuthal velocity at five different axial positions between the 7th and the 8th obstacles: black solid, RIB-1; black dash-dot-dot, RIB-2

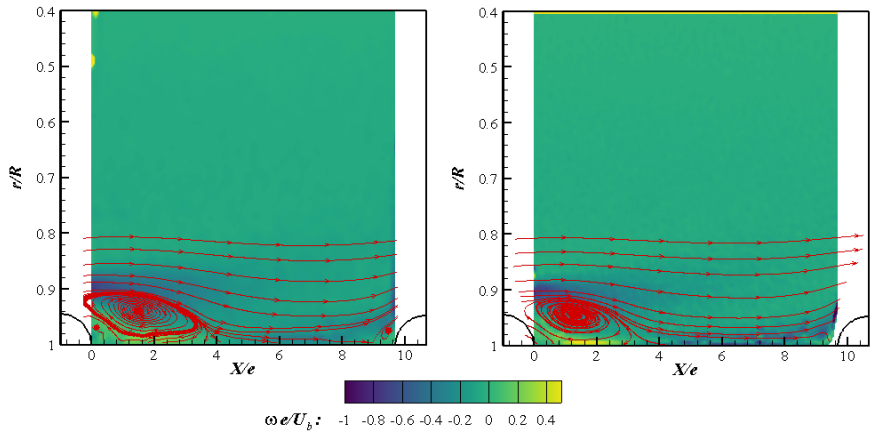


Figure 3.12 - Time-average of the azimuthal component of the dimensionless vorticity at the plane $z=0$ between the 7th and the 8th obstacles: (a), RIB-1; (b), RIB-2

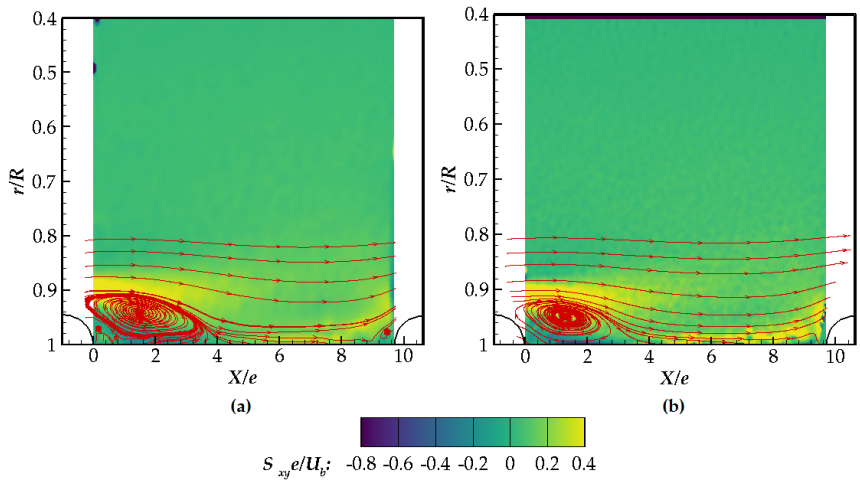


Figure 3.13 - Time-average of the x-y component of the dimensionless rate-of-strain tensor at the plane $z=0$ between the 7th and the 8th obstacles: (a), RIB-1; (b), RIB-2

The dimensionless turbulence statistics, in terms of variance and covariance of the axial and radial velocity fluctuations, shown in Figure 3.14, are defined respectively as:

$$\langle uu \rangle = \frac{\sum_{i=1}^N (U_i - \bar{U}) \cdot (U_i - \bar{U})}{N} \quad (3.7)$$

and

$$\langle uv \rangle = \frac{\sum_{i=1}^N (U_i - \bar{U}) \cdot (V_i - \bar{V})}{N} \quad (3.8)$$

The axial turbulent intensity generated by RIB-1 is slightly higher in all the investigated streamwise and wall-normal range. Both the normal, Figure 3.14-a, and the dimensionless shear stresses, Figure 3.14-b, proportional to the variance $\langle uu \rangle$ and covariance $\langle uv \rangle$, show higher values for RIB-1 than for RIB-2. Maximum values of the streamwise component $\langle uu \rangle_{max}$ of about 7% of the bulk kinetic energy are generated just downstream of the obstacles, while values not greater than 2.5% are reached for the shear stress $\langle uv \rangle$. Those profiles of turbulent flows are influenced by the errors from the s-PIV, particularly, for the measure of the particles displacement in z (direction perpendicular to the calibration plate). Notably, the standard errors associated with the azimuthal component of the velocity and the axial turbulence intensity generated by the two turbulators are shown in Figure 3.15. The statistical method applied to compute these random uncertainties is the bootstrap, which is a resampling algorithm used for turbulence research [34]. Bootstrap samples are drawn from the entire original measurements in order to provide a finite number of other new datasets. For example, if we consider in one spatial point (x, y) the N time measurements of the azimuthal velocity (W), the original dataset is:

$$\mathbf{W}(x, y) = [W_1(x, y), W_2(x, y), \dots, W_N(x, y)] \quad (3.9)$$

According to the bootstrap method, it is possible to rebuild B new datasets

$$\mathbf{W}^*(x, y)_j = [W_{boot,1}(x, y), W_{boot,2}(x, y), \dots, W_{boot,N}(x, y)]_j \quad (3.10)$$

with $j = 1, 2, \dots, B$

where each vector component $W_{boot,i}(x, y)$ is always drawn randomly from the original dataset $\mathbf{W}(x, y)$. The authors chose to resample 25 times the original datasets in line with the work of Efron & Tibshirani [35]. The random sampling with replacement allows evaluating the uncertainty of the estimates with a confidence interval of 95%. In fact, the random component of the extended uncertainty in the measure of $W(x, y)$ is computed as:

$$\pm 1.96 \sqrt{\text{var}(\mathbf{W}^*(x, y))} \quad (3.11)$$

where $\text{var}(\mathbf{W}^*(x, y))$ is the variance of the B bootstrap new datasets:

$$\text{var}(\mathbf{W}^*(x, y)) = \frac{1}{B-1} \sum_{j=1}^B (\overline{\mathbf{W}^*(x, y)}_j - \overline{\overline{\mathbf{W}^*(x, y)}}_j)^2 \quad (3.12)$$

with

$$\overline{\mathbf{W}^*(x, y)}_j = \frac{\sum_{i=1}^N W_{boot,i}(x, y)_j}{N} \quad (3.13)$$

$$\overline{\overline{\mathbf{W}^*(x, y)}}_j = \frac{\sum_{j=1}^B \overline{\mathbf{W}^*(x, y)}_j}{B} \quad (3.14)$$

The standard extended uncertainties in Figure 3.15 are given in terms of percentage of the estimated maximum value, inside the region of interest. In the field far from the wall, between 40% and 75% of the radius, the two measurement

campaigns show the same uncertainty on W , in the range 2-5% (Figure 3.15-a). Moving closer to the wall, the measurements for the RIB-2 show a higher uncertainty, which is because the amplitude of the measure (W) is lower. In the region between 75% of the radius and the wall, such uncertainties have values higher than 5%. These values are about 8% in the shear zone and inside the recirculation bubble, 6-7 % in the zone closer to the leading edge of the obstacles. A peak of 10 % in the uncertainty value is located downstream the reattachment at 95% of the radius. Concerning the estimates of the axial turbulence intensity $\langle uu \rangle$, in Figure 3.15-b, they are given with a random extended uncertainty of about 1% in the zone far from the wall. Values of about 2% are estimated in the wake behind the obstacle, where the flapping shear layer takes place. Those last observations are mainly the same for both geometries. Only close to the shear layer and inside the recirculation bubble, the measurements of $\langle uu \rangle$ for RIB-2 are characterized by a slightly higher value. As further information, the extended uncertainties related to the $\langle uv \rangle$ fields have been added in Appendix D.

The Turbulent Kinetic Energy (TKE) generated by the two ribs is made dimensionless by dividing by the square of the bulk velocity ($k=TKE/U_b^2$), similarly as for the Reynolds stresses. Figure 3.16-a and Figure 3.16-b show that for both geometries, just behind each obstacle, close to the trailing edge, there is an area where the time-averaged TKE is equal to zero. In this region of the flow field, the energy exchange between fluid elements is low and high unwanted temperature gradients are expected. Because of a lower azimuthal swirl, RIB-2 provides lower values of the TKE (Figure 3.16-b). Moreover, it is confined in the region above 90% of the radius. Hence, with RIB-2, there is a lower energy exchange in all the directions, indicating that the temperature profile is less uniform than with RIB-1 (Figure 3.16-a).

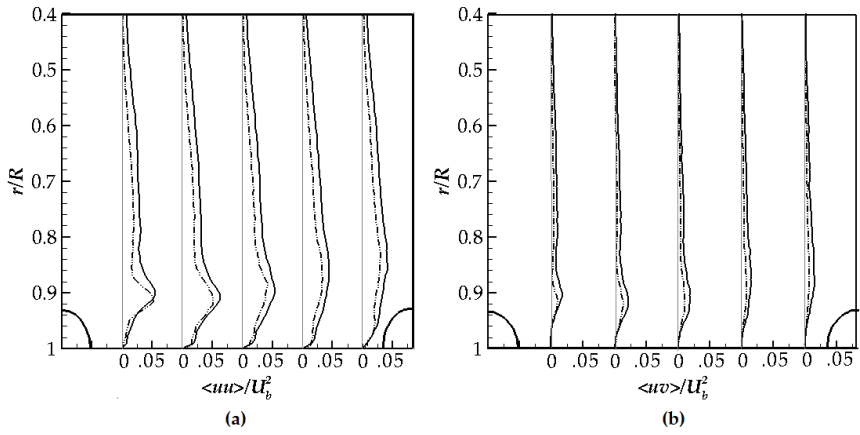


Figure 3.14 - Normalized profiles of the turbulent normal stress $\langle uu \rangle$ (a) and shear stress $\langle uv \rangle$ (b) at five different axial positions between the 7th and the 8th obstacles: black solid, RIB-1; black dash-dot-dot, RIB-2

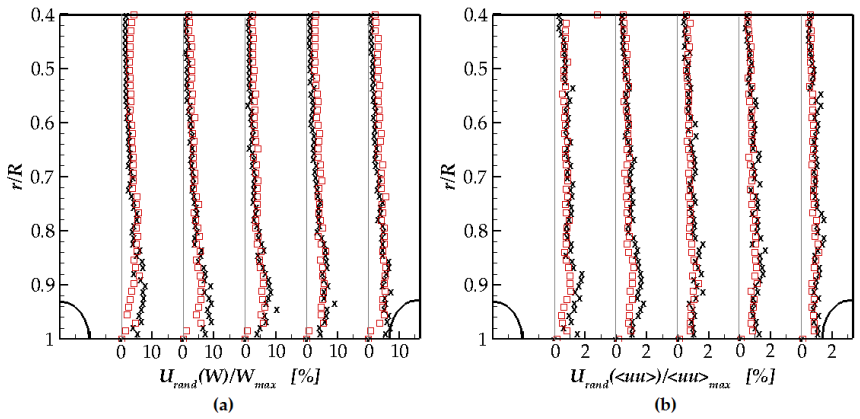


Figure 3.15 - Normalized profiles of the random uncertainty in the measurements of the azimuthal velocity component (a) and normal stress $\langle uu \rangle$ (b) at five different axial positions between the 7th and the 8th obstacles: red squares, RIB-1; black crosses, RIB-2

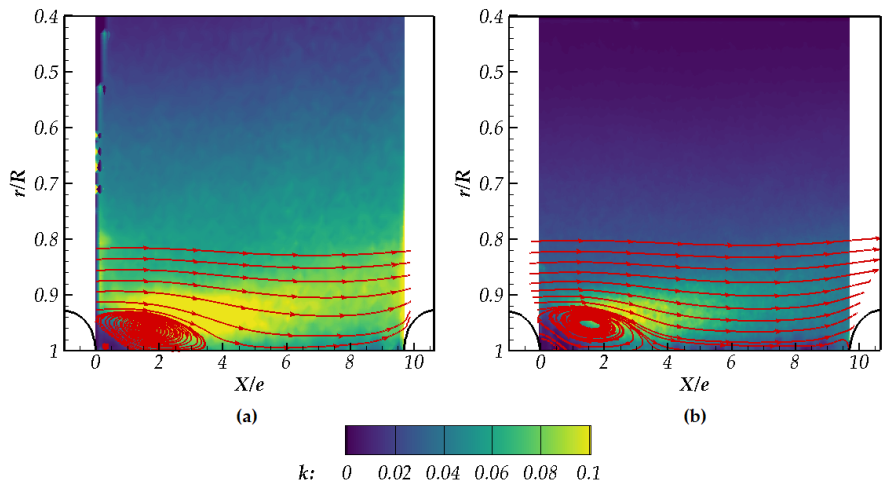


Figure 3.16 - Dimensionless Turbulent Kinetic Energy between the 7th and the 8th obstacles: (a), RIB-1; (b), RIB-2

This is in accordance with the fact that higher turbulent intensities let the turbulent diffusivity increase. As a consequence, a reacting flow with such turbulence has smoother temperature profiles which are required in chemical reactors to have a higher production of light olefins. In the region close to the leading edge of RIB-2, the low TKE values are combined with low azimuthal velocities and low axial fluctuations. Therefore, it is concluded that its effect on the local cooling rate at the wall has a smaller impact on the leading edge of the obstacles, differently from the case of RIB-1. Such an impact of the flow topology on the local heat transfer was already discussed by Wang, Salewski, & Sundén [17] and Mayo et al. [23].

3.4. Conclusions

This investigation is aimed at providing experimental data concerning the aerodynamics of complex swirling flows in ribbed tubes. The ribbing technique consists of using similar wall bounded helices as turbulators for the disruption of the thermal boundary layer. Stereo-PIV was used to analyze the flow between the obstacles, at a Re of the order of 20,000. The geometrical difference between the two helicoidal turbulators is that one presents a continuously constant obstacle height (2D obstacle) while the second one presents obstacles whose height depends upon the azimuthal coordinate (3D obstacle). The measurements mainly show that:

1. For both the turbulators, C_f results to be not function of the flow regime in the range $Re=20000\div80000$. The skin friction coefficients of the continuous and discontinuous turbulators are, respectively, 7 and 4 times higher than the case of a smooth pipe, at the same Re .
2. The turbulators induce a three-dimensional separation, with a generation of a recirculating bubble downstream the 7th obstacle. These phenomena lead the turbulence of the surrounding flow field to increase.
3. Because of the different shape, the reattachment lengths of the 2 separated flows are different. In RIB-1 the reattachment occurs at $X/e=4.15$, which is higher than that one in RIB-2.
4. Vorticity and strain rate component are not affected by the discontinuity of the helicoidal turbulator.
5. The continuous geometry generates a higher TKE distribution inside the flow and a more significant azimuthal swirl in comparison with the discontinuous one.

The emphasis was put on investigating the effect of the wall turbulator discontinuity on the friction losses and the turbulent characteristics, known to

influence the local heat transfer and the radial mixing [24]. The induced separation-reattachment phenomenon is favorable in terms of thermal efficiency, particularly for chemical reactors used for olefin production where a combination of a low-pressure drop and a high heating rate are required. The given experimental dataset is therefore of great importance for the validation of CFD codes meant that can be used for design and optimization.

3.5. References

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4

Influence of obstacles on the wall heat transfer for 2D and 3D helically ribbed pipes

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Abstract

Obstacles placed on the inner walls of pipes lead to flow separation and subsequent reattachment with an overall beneficial effect on the local heat transfer. In this work, the effects of both a continuous and a discontinuous helicoidal turbulator on the local heat transfer are investigated and compared for Reynolds numbers in the range from 20000 to 80000. Both turbulators are cylindrical helices with constant angle and fixed direction. The continuous geometry has a pitch-to-diameter-ratio (p/e) equal to 11, an inclination angle of 80° and a uniform obstacle height-to-diameter-ratio (e/D) equal to 3.6%. The discontinuous geometry is equivalent to the continuous one except that it has a varying obstacle height throughout the helix. Steady-state heat transfer measurements are carried out by means of Liquid Crystals Thermography (LCT). A recently developed calibration method for the narrow-band Thermochromic Liquid Crystals (TLCs) is successfully modified to lower the angular effects on the measurement uncertainty. The local wall heat transfer rates and the temperature distributions are strongly influenced by the flow reattachment and thus by the geometry of the investigated turbulators. Compared to a smooth pipe at the same Reynolds number, the continuous and the discontinuous turbulator increase the average Nusselt number by factors of approximately 1.8 and 1.5 respectively. These heat transfer measurements with respective uncertainties are meant to be a valuable dataset for the validation of Large Eddy Simulations (LES) of turbulent flows in pipes equipped with turbulators.

4.1. Introduction

In the process industry, there is an interest in using tubes equipped with artificial roughness elements to improve heat transfer and create better mixing [1, 2, 3, 4, 5].

Similarly in turbomachinery, the internal cooling channels of blades is based on ribbed walls to provide a higher thermodynamic efficiency [6]. Pressurized air is extracted from the compressor to cool down the turbine blades of the high-pressure stages. In this way, the expansion of the exhaust gas inside the turbine can be done at higher total inlet temperatures (TITs). Since the thermal efficiency of the engine is a function of the TIT, the higher the temperature, the better the gas turbine performance [7]. As stated above, one of the techniques used for gas turbines as a cooling system, for instance, is the use of internal ribbed channels. Likewise, tubular chemical reactors used in endothermic processes like steam cracking or ethylene dichloride (EDC) pyrolysis are often equipped with ribs to improve the overall heat transfer between process gas side and the furnace side. Moreover, due to the ribbed structures, flow is ejected from the hot wall to the cooler center of the pipe, reducing radial temperature and concentration gradients, positively affecting selectivities [8]. As an example, implementation of these structures allows a higher production of propylene [9].

The effects of the ribs are explained by the fact that the turbulent boundary layer is disrupted by the presence of the obstacles. It separates and reattaches on the wall at a certain distance downstream of the ribs. In this way, the impingement of the shear layer and the increased turbulent kinetic energy augment over compensates the effects of the flow detachment, leading to wall Nusselt numbers (Nu) above values for smooth pipe flows [10]. In the open literature, heat transfer data are often provided in terms of averaged wall heat transfer [11, 12, 13], rather than a detailed Nu distributions. This occurred even though large variations of convective heat flux exist between the trailing edge of one obstacle and the leading edge of the following one. This Nusselt number profile is a function of the geometry shape, which influences the generated turbulent flow structures [14]. On the other hand, the repeated-rib roughness has the drawback to let the wall shear stresses, and the skin friction factor, increase. Friction factor and heat transfer correlations are given by P.

R. Chandra et al. [13], D. L. Gee and R. L. Webb [15] for the design of cooling channels respectively with rectangular and circular cross-sections.

The main question is thus how to optimize this rib shape, i.e. minimize the friction losses whilst keeping the wall convection elevated. Typically, numerical methods are employed as they allow more reliable predictions of local wall heat transfer than semi-empirical correlations. These methods require thorough comparison with experimental data to prove their validity and accuracy. Comparison between experimental data and different numerical flow solvers for ribbed channels are provided by H. Iacovides and M. Raisee [16]. The closures used by the authors ranged from eddy-viscosity $k-\varepsilon$ to algebraic and differential stress closures. Since the large-scale motion in ribbed channels is not simulated sufficiently in $k-\varepsilon$ models, and the use of wall functions might cause discrepancies in the heat transfer computation [17], the focus of numerical investigations shifted towards Large Eddy Simulations (LES) [8, 18, 19] based on filtered energy equations.

To verify how the filtered energy equation and the corresponding sub-grid scale model represent the local heat transfer, a validation is requested by comparing the numerical solutions with measurements. An example of an experimental investigation was done by P. G. Vicente et al. [11] for the characterization of spiral corrugated pipes, using point-wise techniques. The authors exhibit overall values of the augmented Nusselt number as a function of the Reynolds number (Re) for different working fluids in corrugated pipes. G. Rau et al. [14] measured the Nu profile over the wall between the obstacles in a straight channel. Their measurements of the heat transfer enhancement, at the inner wall of the channel, were made by means of Liquid Crystals Thermography (LCT). This technique consists in the application on a surface of a paint that can reflect different visible light wavelengths in function of its surface temperature. Practically, the temperature state of the liquid crystals leads the orientation of the molecules to change direction. This phenomenon occurs at several different multi-layers, stack one on top of each

other. The different orientation of the molecules generates helicoidal structures with a geometry dependent on the temperature. Since the light's wavelength reflected by the TLCs is a function of the helicoidal geometry, at different temperatures correspond to different scattered colours. The Thermochromic Liquid Crystals (TLCs) scattering of different colors is reversible, i.e., their calibration function is unique [20]. With the use of LCT, it is possible to measure the temperature field of the inner wall of a pipe, and to solve the local convective heat transfer, without any thin-wall approximation [21]. Additionally, in order to have an optical access to the inner wall, the pipe under investigation has to be made out of a transparent material (e.g., glass, Plexiglas and acrylic plastics).

An issue concerning the TLCs is their application on curved surfaces. The light scattered by the TLCs depends not only on the temperature but also on the angle between the normal to the surface and the direction of viewing [22]. An up-to-date application of Thermochromic Liquid Crystals for heat transfer measurements is provided by I. Mayo et al. [23] in the case of a pipe ribbed with one helicoidal turbulator [24, 25] at the flow regime of $Re=20000$. The authors show experimentally the connection between peaks in Nusselt number and turbulent kinetic energy of the wall-bounded swirling flow generated by the helicoidal obstacles. In view of the complexity of LCT on curved surfaces, the results concerning the Nu field between the obstacles were limited to the region extended in 1 helix pitch and only 5° . Under those circumstances, a new experimental approach on the use of liquid crystals needs to be provided to the literature, where the effects of obstacles are highly influencing the temperature distribution over an extended surface.

The helicoidal turbulator investigated by the authors is a continuous wall obstacle with a constant height throughout the helix path. To lower the friction losses given by uninterrupted ribs, there is a significant interest in the use of discontinuous helicoidal turbulators [26](Fig. 4.1). Although this configuration is

associated with a lower roughness, its effects on the wall heat transfer still need to be evaluated and compared with those given by continuous geometries.

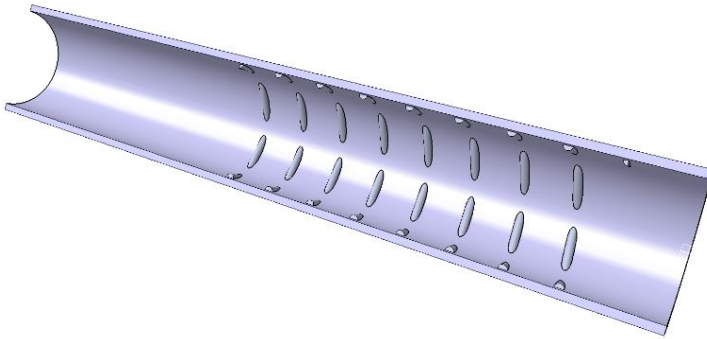


Figure 4.1 - Section of the pipe equipped with a discontinuous helicoidal turbulator

Therefore, the main objectives of the present work are to provide a detailed characterization of 2 helically corrugated cooling channels effects on the local heat transfer and to contribute with a well-documented test case for high-fidelity computations. The effects of the rib discontinuity on the wall convective heat transfer are compared with those given by the continuous helix, investigated by Mayo et al. [23], at the flow regimes in the range $Re=20000-80000$. The measurements are carried out using steady-state LCT, presenting the enhancement factor (EF) distribution on the inner pipe wall. The EF is defined as the ratio between the local Nusselt numbers of the corrugated and bare pipes. Values higher than the unity show where the effects of the turbulators provide an increase in the local convective heat transfer. Furthermore, the authors show a methodology to overcome the issue of angular dependency associated with this temperature measurement technique.

4.2. Description of experiments

4.2.1. Geometries under investigation

The ribbed pipes under investigation were obtained by gluing wall obstacles on the top of the smooth inner surface of a cylinder. Two different types of turbulators, named RIB-1 and RIB-2 (Fig. 4.2), are successively inserted in the test section. RIB-1, which is the continuous helicoidal turbulator investigated by Mayo et al. [23], consists of a semi-circle section swept through the turbulator coil, tangentially to the inner surface of the test section. The helix has a pitch of 63 mm repeated for 10 times, and the semi-circle section has a constant height-to-pipe diameter ratio (i.e., blockage-ratio) of 3.6%.

The second geometry, RIB-2, has the same helix parameters as RIB-1 except for the rib section's height of the semi-circle, which changes along with the coil pattern (Fig. 4.2). The flow obstacles are obtained out of a solid intersection of a 150 mm diameter cylinder with a smaller one (10.8 mm diameter) as in Fig. 4.3. The axis of the small cylinder is tangent to the external surface of the bigger one, and it is set at an angle (ϑ) of 80° with respect to the axial direction of the big cylinder. The value of the coil angle was the same as for the continuous turbulator.

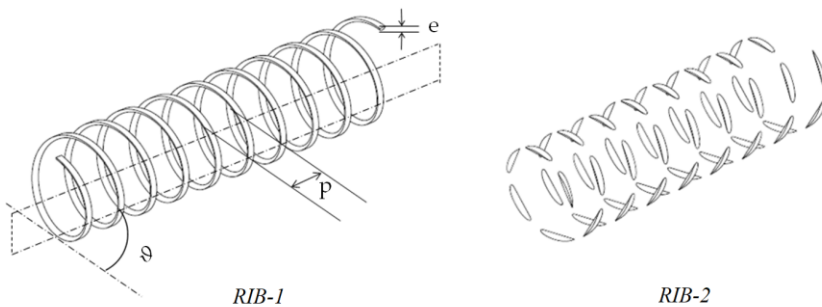


Figure 4.2 - Continuous and discontinuous turbulators

Therefore, the Boolean intersection between the cylinders mentioned above was chosen in a way that the obstacles have circular segments. One of those profiles is a half circle with a maximum height of 5.4 mm. This value corresponds to the constant height of the RIB-1. Such a solid represents the shape of each wall obstacle of the discontinuous geometry, which was put throughout the rib helix pattern. In one pitch of the helix, which is 63 mm, there are 6 obstacles equally spaced, for a total of 9 full pitches. Each obstacle is made of Acrylonitrile Butadiene Styrene (ABS), manufactured by stereolithography.

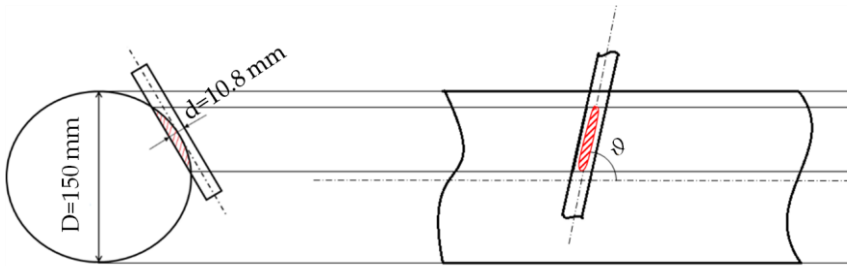


Figure 4.3 - Sketch of a single obstacle of RIB-2 obtained as a solid intersection of two cylinders with different diameters

4.2.2. Experimental setup

The heat transfer measurements of the corrugated pipes were done in the low-speed wind tunnel (Figure 4.4) located at the “Heat and Mass Transfer Laboratory” of the von Karman Institute for Fluid Dynamics. The facility consists of a pipe circuit, with a constant diameter of 150mm. The air mass flow is generated by a centrifugal fan. To avoid that the inlet flow would be affected by the discharge fluctuations of the blower, the facility operates in suction mode. The air enters first the inlet duct which has a length of 15 diameters. With such an extent [27], a turbulent fully developed boundary layer is provided at the inlet of the downstream test section (see Appendix E).

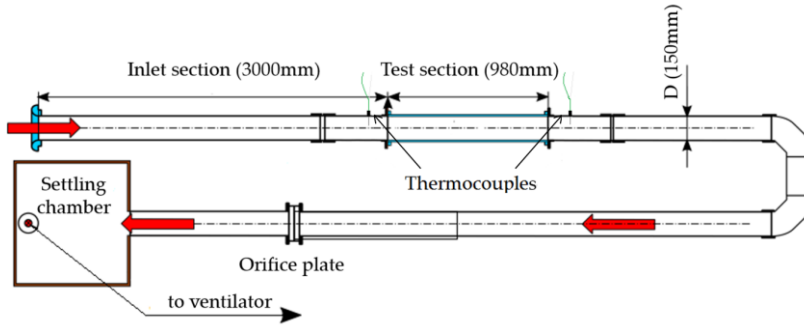


Figure 4.4 - Low-speed wind tunnel for heat transfer measurements

Two k-type thermocouples were inserted respectively at the inlet and at the outlet of the test section to measure the air temperature at the pipe axis position. The transparency of this pipe is guaranteed by the acrylic glass material of the pipe, whose length is 980mm. The test section is divided into two halves to have full access to its inner surface where both layers of different material and obstacles are displaced. The first layer of micro-encapsulated wide-range TLCs was sprayed on the inner wall of the test section to be able to measure its surface temperature distribution. On top of it, there are respectively a black matt paint and an Inconel 600 foil which can heat the test section with a uniform heat flux when a DC voltage is applied to the Inconel foil. One final layer of black matt paint (Rust-Oleum 2178 black topcoat) was sprayed onto the metal sheet. In this fashion, the surface has an emissivity close to the unity, and the heat reflections from the Inconel foil are lowered. Furthermore, the whole test section was put inside a black box illuminated by a diffusive LED light panel (Pan-70W), to avoid any external reflections on the acrylic glass (Fig. 4.5). The color distribution of the TLC layer can be recorded as RGB intensities by an SLR camera. The digital camera is focused on the section from a distance of about 850 mm with an angle of 45° with respect to the horizon, as in Figure 4.5. The portion of the inner cylindrical surface under investigation is called Region of Interest (ROI), which is extended over 73.5 mm and 60° in the axial and azimuthal directions, respectively (Fig. 4.6). The ROI was chosen in the

way that the inner surface is disposed in the area between 7 and 8 rib-pitches. According to F. Coletti et al. [28], this distance was chosen to have a typical development for low blockage ratio channels. An infrared camera (ThermaCAM SC3000 – Flir System) was put on the same side of the photographic equipment to measure the outer surface temperature of the cylinder.

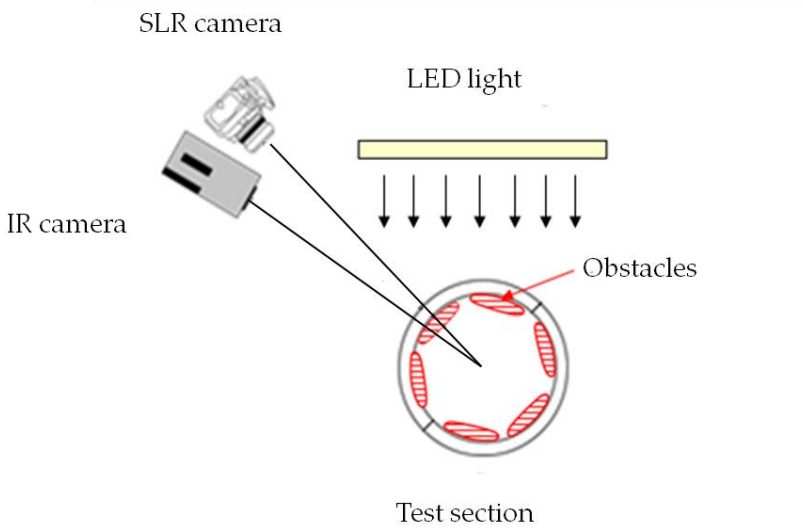


Figure 4.5 - Cameras disposition during the heat transfer measurements

Downstream of the test section, the mass flow enters first a U-bend and then a pipe with a length of 30 diameters. According to the normative ISO [29], such a length is required to mount an orifice plate as a pressure differential device inserted in a circular cross-section duct.

The mass flow meter is specifically an orifice plate with corner tapings, whose differential pressure is read by a water manometer (resolution of 0.1 mm H₂O). From the reading of the pressure drop, the mass flow rate and hence the Reynolds number can be evaluated (according to the normative ISO 5167) with an uncertainty of ± 300 . The last component of the facility is a settling chamber which is directly connected to the blower. It has the function of damping the flow fluctuations

produced by the fan, which may influence the pressure downstream of the orifice plate.

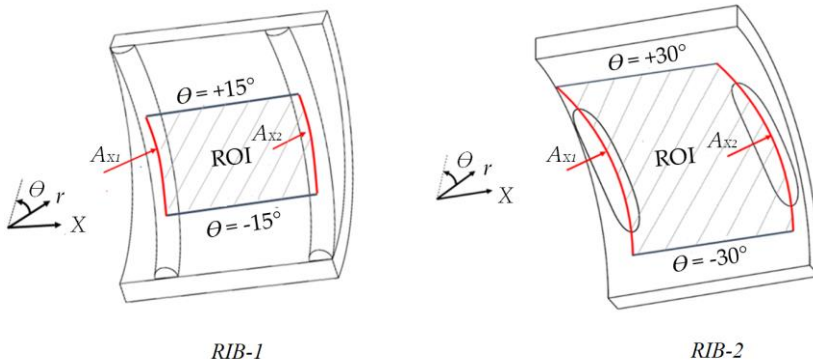


Figure 4.6 - ROI between two obstacles of the turbulators RIB-1 and RIB-2

4.2.3. TLCs Calibration in situ

One layer of TLCs is applied to the inner wall surface of the transparent test section. The temperature level when the liquid crystals start to be optically active, which is 308.15 K, and the range where they can be operated (20 K) are called red-start and color play, respectively. From the same distance as in the measurements, a Nikon D7100 camera (CMOS sensor with 6000×4000 pixels) records the scattered light of TLCs with a 105 mm Nikon AF DC-Nikkor, as optical lens. The ISO sensitivity was set at 100 to enhance the signal to noise ratio of the images. The aperture and the shutter speed were chosen at $f/14$ and 3 seconds, to have pictures with the RGB intensities as high as possible. The reason is that playing with highly saturated colors, the temperature uncertainty related to the LCT gets minimized [22]. The white balance of the camera was fixed at 4170 K, which is the same value as for the light source. Hence, the range of the RGB space and the sensitivity of the SLR camera are optimized [30]. With the use of the Hollingsworth equation [31], the

RGB intensities of each pixel inside the ROI can be translated into a different tri-value combination, called Hue-Saturation-Intensity. Along these lines, it is possible to extrapolate the value referring only to the wavelength of the TLCs color, eliminating the intensity and the saturation components. During the in-situ calibration, half of the test section is removed to have optical access for the infrared camera. Accordingly, the inner surface temperature is measured by displacing the thermo-camera in line with the digital one but on the other side, as shown in Figure 4.7. The infrared (IR) camera was calibrated twice with surface thermocouples in the limited temperature range of 303-333 K, focusing on two different targets separately. One of them is the external pipe surface, made in Plexiglas, while the other target is the inner surface of the test section whose final layer is a black matt paint. The need for two different calibrations arises from the fact that the employment of the infrared camera is for both the TLCs calibration (Fig. 4.7) and for the measure of the external pipe temperature. In these two different situations, two different targets are seen by the camera. The in-situ calibrations of the infrared camera provide an extended uncertainty of ± 0.4 K, with a level of confidence of the interval equal to 95% (see Appendix F).

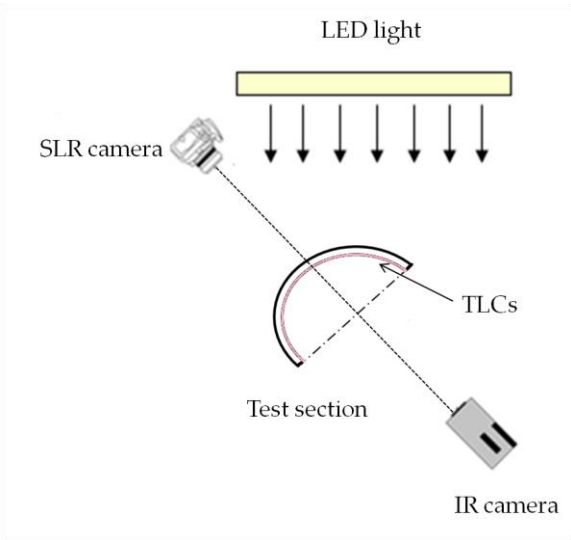


Figure 4.7 - Cameras disposition for the TLCs in-situ calibration

The Hue distribution on the inner surface of the test section is a function not only of the temperature but also of the azimuthal coordinate. As a consequence of such angular dependency of the TLCs [22], the in-situ calibration is made in a way that each pixel row of the image's ROI, has a related calibration curve, (Hue vs Temperature). A single calibration procedure combines the mean temperature (given by the IR camera) to the mean value of Hue for a selected row of the ROI. The method is providing a spline curve fitted to 18 different temperature points. With the purpose of calibrating all the ROI over the inner surface of the test section, the before mentioned algorithm is applied for all the angular positions (see Fig. 4.6). The outcome of this procedure is shown in Figure 4.8, where three calibration curves are illustrated at three different angular positions, at 0° , -30° and $+30^\circ$.

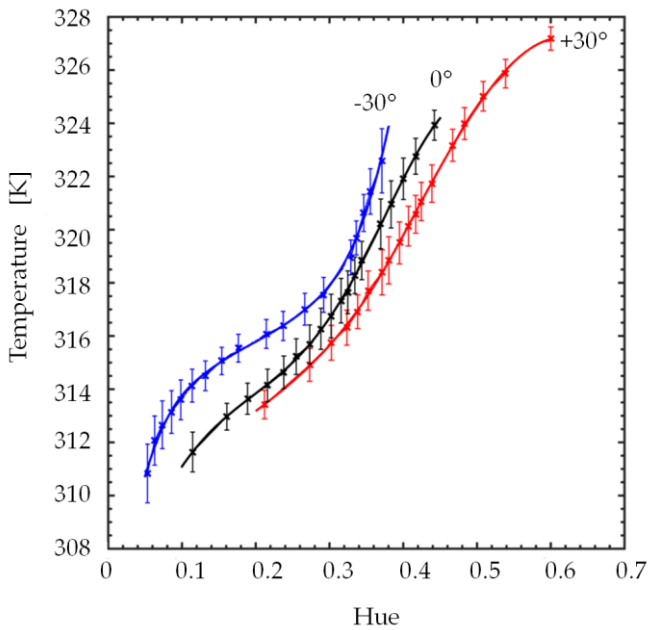


Figure 4.8 - TLCs calibration curves at different angular positions

The uncertainty values, shown in Figure 4.8, take into account the spatial variation of Hue for each row; the spatial variation of the temperature provided by the infrared camera; the uncertainty on the temperature given by the thermal camera calibration; and the uncertainty in the regression functions. The procedure to calculate the uncertainty of the experimental method is reported by Mayo et al. [32]. With a level of confidence of 95%, the TLCs calibration methodology provides a resulting extended uncertainty on measurement of the inner wall temperature in the range $\pm 0.5\text{K}$ and $\pm 0.8\text{K}$. The error value depends on the angular position to which the calibration curve is referred to, and also on the temperature values. It turns out that the highest values of the measurement uncertainty are where the partial derivatives $\partial T / \partial \text{Hue}$ are at the highest (see Fig. 4.8).

4.3. Data Reduction and Uncertainty Analysis

The outcome of the experimental campaign is presented as the Nusselt number divided by the same dimensionless number for a smooth pipe (Nu_s) at the same flow regime (Re) with uniform heat flux as a boundary condition. The Nusselt number reference, for bare pipes, is given by the Dittus-Boelter correlation for the case of heated pipe flows:

$$Nu_s = 0.023 \cdot Re^{0.8} Pr^{0.4} \quad (4.1)$$

This ratio, called “Enhancement Factor” (EF) is provided as a scalar field on a cylindrical surface between two consecutive obstacles, that is the ROI. Hence, for a given Reynolds number, the EF contour is a function of the axial (X) and azimuthal (θ) cylindrical coordinates as follows:

$$EF(X, \theta) = \frac{Nu(X, \theta)}{Nu_s} = \frac{h(X, \theta) \cdot D}{k_f(T_b) \cdot Nu_s} \quad (4.2)$$

where k_f is the air thermal conductivity, which is a function of the bulk temperature (T_b) [33]; D is the pipe diameter, and h is the convective heat transfer coefficient. By definition, h is given by the formula:

$$h(X, \theta) = \frac{\dot{q}_{conv}(X, \theta)}{[T_w(X, \theta) - T_b(X)]} \quad (4.3)$$

where $\dot{q}_{conv}$ is the wall heat flux transferred by convection and T_w is the temperature of the inner wall, inside the ROI. The bulk temperature is assumed to be equal to the temperature of the flow at the center of the pipe, which inappreciably increases from the inlet to the outlet section. Most compelling evidence is that the difference between the inlet and outlet temperatures is at the same order as the thermocouples uncertainty. The before mentioned assumption was made for the sake of its simplicity, without computing the integral energy balance equation:

$$\dot{m}c_p(T_b(X_{ROI}) - T_c(X_{inlet})) = \int_{X_{inlet}}^{X_{ROI}} \int_0^{2\pi} \dot{q}_{conv}(X, \theta) r d\theta dX \quad (4.4)$$

The use of this equation to compute the bulk temperature introduces further uncertainties in the final results for the Nu number. Nevertheless, in the measurements, a bias is introduced, and it can be estimated. The center of the ROI is placed at 4 diameters downstream of the test section inlet. The temperature difference ΔT , between T_b at the ROI's inlet ($X = 4 \cdot D$) and T_c at the test section inlet ($X = 0$), can be computed using equation 4.4, resulting in:

$$\Delta T = T_b(X = 4 \cdot D) - T_c(X = 0) = \frac{\bar{\dot{q}}_{conv}}{\dot{m} \cdot c_p} \cdot 2\pi \cdot r \cdot (4 \cdot D) = 1.03 \text{ K} \quad (4.5)$$

where, for the sake of simplicity, the heat flux is considered equal to its surface average inside the ROI, $\bar{\bar{q}}_{conv}$. Considering that no significant variations of the air temperature, T_c at the centreline of the pipe are expected, the error computing the flow properties with T_c instead of T_b is:

$$T_b(X = 4 \cdot D) - T_c(X = 4 \cdot D) \cong \Delta T \quad (4.6)$$

With such a bias in the evaluation of the bulk temperature, the relative error in Nu results to be in percentage:

$$\frac{\Delta Nu}{Nu(Re = 80000)} = \frac{\frac{\partial Nu}{\partial T_b} \cdot \Delta T}{Nu(Re = 80000)} \cong 5\% \quad (4.7)$$

with:

$$\frac{\partial Nu}{\partial T_b} = \left(\frac{\partial Nu}{\partial h} \cdot \frac{\partial h}{\partial T_b} + \frac{\partial Nu}{\partial k_f} \cdot \frac{\partial k_f}{\partial T_b} \right) \cong 11.5 \frac{1}{K} \quad (4.8)$$

As a consequence, the introduced bias was quantified, and it can be added to the other error sources for the uncertainty analysis.

The convective heat flux ($\dot{q}_{conv}$ [W/m^2]) was evaluated by solving the energy balance at each finite volume (Figure 4.9) of the Inconel sheet (whose thermal conductivity is k_{inc}), where the heat production ($\dot{q}_j$ [W/m^3]) is generated by Joule effect. Since the metal foil is considered to be at the same temperature as the TLCs layer, it is possible to solve the energy balance:

$$\dot{q}_{conv}(X, \theta)A_c = \dot{q}_j V_j - \dot{q}_r(X, \theta)A_r - \partial \dot{q}_x(X, \theta)A_x - \partial \dot{q}_\theta(X, \theta)A_\theta \quad (4.9)$$

where the heat fluxes:

$$\partial \dot{q}_X = \frac{\partial \dot{q}_X}{\partial X} \cdot dX = \frac{\partial}{\partial X} \left(-k_{Inc} \frac{\partial T_w(X, \theta)}{\partial X} \right) \cdot dX \quad (4.10)$$

$$\partial \dot{q}_\theta = \frac{1}{r} \frac{\partial \dot{q}_\theta}{\partial \theta} \cdot r d\theta = \frac{1}{r} \frac{\partial}{\partial \theta} \left(-k_{Inc} \frac{1}{r} \frac{\partial T_w(X, \theta)}{\partial \theta} \right) \cdot r d\theta \quad (4.11)$$

$$\dot{q}_r = -k_p \frac{\partial T_{solid}(X, \theta, r)}{\partial r} \quad (4.12)$$

are approximated as finite differences of the measured temperature of the inner surface.

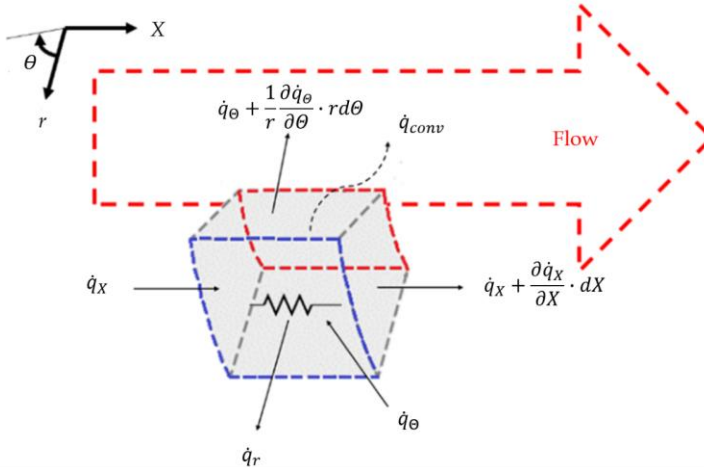


Figure 4.9 - Heat balance for a generic Inconel finite volume

In particular, the two lateral conductive heat fluxes ($\partial \dot{q}_\theta$, $\partial \dot{q}_X$) were estimated by using the central difference scheme for the points of the ROI. Considering that the contour of the measured wall temperature is a scattered data affected by errors, it was smoothed with a 2-D Gaussian filter, as for images post-processing [31]. The size of the filter was chosen in order to have a new temperature field whose standard

deviation from the original data is less than or equal to the standard uncertainty of the TLCs [21]. The radial heat flux ($\dot{q}_r$) is calculated numerically with an upwind scheme which allows computing the temperature derivatives at the inner surface of the test section, i.e., the ROI. For this achievement, the 3D temperature field (T_{solid}) inside the thickness of the test section was solved with the Energy Balance Method for a steady-state condition [27]. The procedure expects to generate a mesh inside the solid for the portion of cylinder delimited by the ROI (Fig. 4.6) and to apply the conservation of energy for each node of the mesh. In the case of M equations with M temperature unknowns, the coefficient matrix A of the linear system is of size M×M. Particularly, the coefficients of the temperature unknowns ($a_{i,j}$) are given by the nodes distances and the heat-exchange areas between the nodal volumes (Eq. 4.13). The hexahedral structured mesh is obtained dividing the pipe thickness (10 mm), the axial length (73.5 mm) and the azimuthal direction (60°) of the ROI in respectively 5, 500 and 500 equally spaced parts, for a total of 1075000 cells. The equation for internal nodes becomes:

$$\begin{aligned}
 & \left[\frac{T_{(i-1,j,k)} - T_{(i,j,k)}}{\Delta i} \times S_{i-1}(i,j,k) \right] + \left[\frac{T_{(i+1,j,k)} - T_{(i,j,k)}}{\Delta i} \times S_{i+1}(i,j,k) \right] \\
 & \quad + \\
 & \left[\frac{T_{(i,j-1,k)} - T_{(i,j,k)}}{\Delta j} \times S_{j-1}(i,j,k) \right] + \left[\frac{T_{(i,j+1,k)} - T_{(i,j,k)}}{\Delta j} \times S_{j+1}(i,j,k) \right] \\
 & \quad + \\
 & \left[\frac{T_{(i,j,k-1)} - T_{(i,j,k)}}{\Delta k} \times S_{k-1}(i,j,k) \right] \\
 & \quad + \left[\frac{T_{(i,j,k+1)} - T_{(i,j,k)}}{\Delta k} \times S_{k+1}(i,j,k) \right] = 0
 \end{aligned} \tag{4.13}$$

with i, j , and k defining the position of the nodes inside the 3-dimensional space (Fig. 4.10). If with $\bar{b}$ we denote the vector of the well-known terms of the nodal energy equations, the vector containing the temperature values inside the solid $T_{solid}(X, \theta, r)$ may be computed with the matrix inversion method [27]:

$$\begin{bmatrix} T_{solid}(1) \\ \vdots \\ T_{solid}(M) \end{bmatrix} = \begin{pmatrix} a_{11} & \cdots & a_{1M} \\ \vdots & \ddots & \vdots \\ a_{M1} & \cdots & a_{MM} \end{pmatrix}^{-1} \cdot \begin{bmatrix} b_1 \\ \vdots \\ b_M \end{bmatrix} \quad (4.14)$$

The equations of the linear system respect the axial periodicity at the planes A_{X1} and A_{X2} (Fig. 4.6)) and the helix-wise periodicity at $\theta = -30^\circ$ and $\theta = +30^\circ$. Besides, the boundary conditions at both the inner and outer wall are the measured temperature fields.

For this reason, radial temperature gradients computed at the inner wall for nodes which are closer than half of the pipe thickness are more affected by the temperature errors given by the liquid crystals thermography. The uncertainty on the heat transfer measurements is mainly related to the measure of the inner surface temperature made by liquid crystals thermography. The max LCT uncertainty ($u_{T_{wall}}$) of ± 0.8 K on the temperature measurements has an influence on the Nu of 7% at $Re=80000$, which can be computed from:

$$\frac{\Delta Nu}{Nu(Re = 80000)} = \frac{\frac{\partial Nu}{\partial T_{wall}} \cdot u_{T_{wall}}}{Nu(Re = 80000)} \cong 7\% \quad (4.15)$$

with

$$\frac{\partial Nu}{\partial T_{wall}} = \frac{\partial Nu}{\partial h} \left(\frac{\partial h}{\partial T_{wall}} + \frac{\partial h}{\partial \dot{q}_{conv}} \frac{\partial \dot{q}_{conv}}{\partial T_{wall}} \right) \cong 20.2 \frac{1}{K} \quad (4.16)$$

Other less significant error sources arise from the outer surface temperature measurements made by the IR camera; the mass flow measurement given by an orifice plate; and the heat source term in the energy balance (Eq. 4.9), obtained by the Joule effect.

The mass flow meter provides an uncertainty of 1.5% of the reading, while the measures of the electrical power given by the Inconel sheet are less than 1%. A higher understating of the derivatives mentioned above is provided in the work of Mayo et al. [32], where the procedure for the uncertainty quantification for such a heat transfer measurement is also explained.

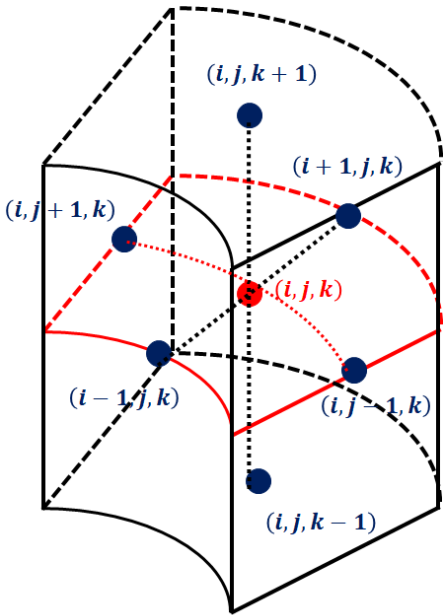


Figure 4.10 - Portion of the hexahedral mesh generated for the node in a general position (i,j,k)

4.4. Results and discussion

4.4.1. Wall heat transfer fields

The physics behind the flow over obstacles is characterized by a 3-dimensional separation and a subsequent wall reattachment [26]. As a consequence, the local heat transfer in the reattachment area increases to its highest value [23, 28]. In the case of the turbulent flow past the continuous helicoidal turbulator (RIB-1), the reattachment length is constant at a fixed Reynolds number. This length is computed from the trailing edge of the obstacles [26]. Figure 4.11-a shows such a phenomenon using EF contour plots at a flow regime with $Re=77180$. The x-axis is the dimensionless axial position, X/e_{max} , and the y-axis is the angular position, θ . The axial position, X , is divided by e_{max} , which is the height of the RIB-1 obstacles. This value is the same as the maximum height of the discontinuous helicoidal turbulator. In the reattachment area, the EF peak of 2.10 ± 0.20 is located $2.5e_{max} - 4.5e_{max}$ far from the trailing edge. This occurs at each angular position in the range $-15^\circ-15^\circ$. The EF , from a value close to the unity, gradually increases from the trailing edge to the peak area. This zone is actually where a recirculating bubble takes place [23]. The $EF=1.00\pm0.10$ just behind the 7th obstacle is where the heat transfer shows its absolute minimum. This is because, in the trailing edge area, the convection is limited by the presence of the obstacle. A slight decrease occurs from the peak area to the leading edge of the 8th obstacle, where the EF is equal to 1.60. This is the area where the flow has a significant azimuthal velocity component, which contributes to the rise in the convective heat transfer compared with a smooth pipe. Such a flow characteristic of helicoidal turbulators have been shown also by Mayo et al. [23] with the use of s-PIV at the flow regime with $Re=20000$. The distribution of the EF on the wall between the obstacles of the discontinuous turbulator is shown in Fig. 4.11-b at the flow regime with $Re=78850$. The flow reattachment doubles the rate of heat transfer compared to heat transfer in smooth

channels ($EF = 2.00 \pm 0.30$), at the same conditions. Differently from the continuous geometry, the area where the flow reattaches is not parallel to the trailing edge of the obstacle. This is due to the discontinuity of the obstacles, with a height changing through the helical path of the turbulator. Therefore, the reattachment length at $\Theta = 0^\circ$ is comparable with the one in RIB-1. At the other angular positions in the range -22° – 22° the reattachment length is proportional to the rib height. In such a way, the reattachment zone has a parabola-like shape where the local EF peak changes in function of the angular position. At $\Theta = 0^\circ$ the EF peak is equal to 1.90 ± 0.30 , which results to be lower than that one obtained in the continuous turbulator. The absolute maximum value of EF is 2.35 ± 0.36 , located at $\Theta = 18^\circ$. This area is close to the obstacle edge where the flow accelerates and enter inside the inter-rib area, with an azimuthal component given by the helicoidal shape of the turbulator. On the other hand, at $\Theta = -18^\circ$, the local EF maximum value is 2.00 ± 0.30 located very close to the obstacle trailing edge. The area just downstream the trailing edge has values of EF lower than the unity, $EF = 0.90 \pm 0.10$. This is given by the fact that the region where flow recirculates is known for a decrease in convective heat rate, as for RIB-1.

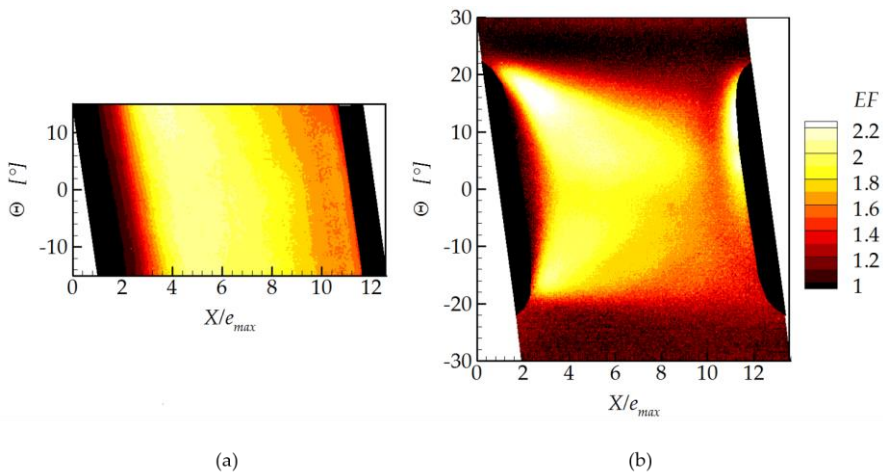


Figure 4.11 - EF distribution between the 7th and the 8th obstacles: left, RIB-1 at $Re=77180$; right, RIB-2 at $Re=78850$

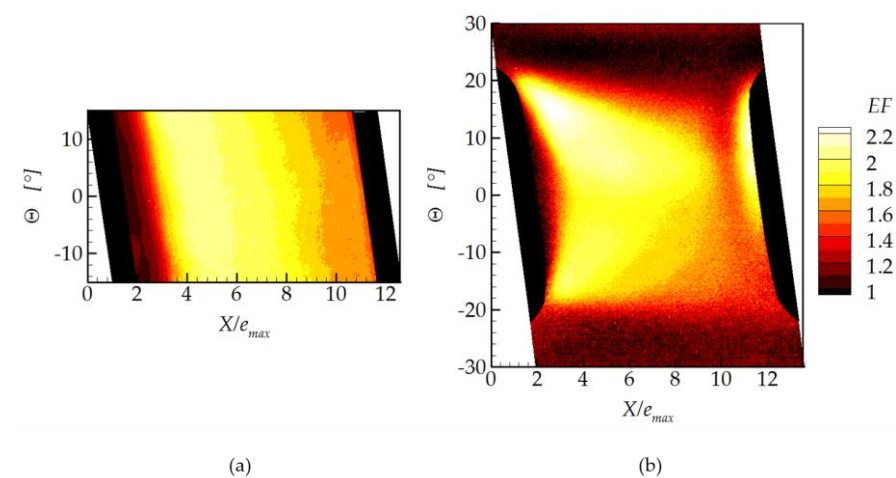


Figure 4.12 - EF distribution between the 7th and the 8th obstacles: left, RIB-1 at $Re=59730$; right, RIB-2 at $Re=61850$

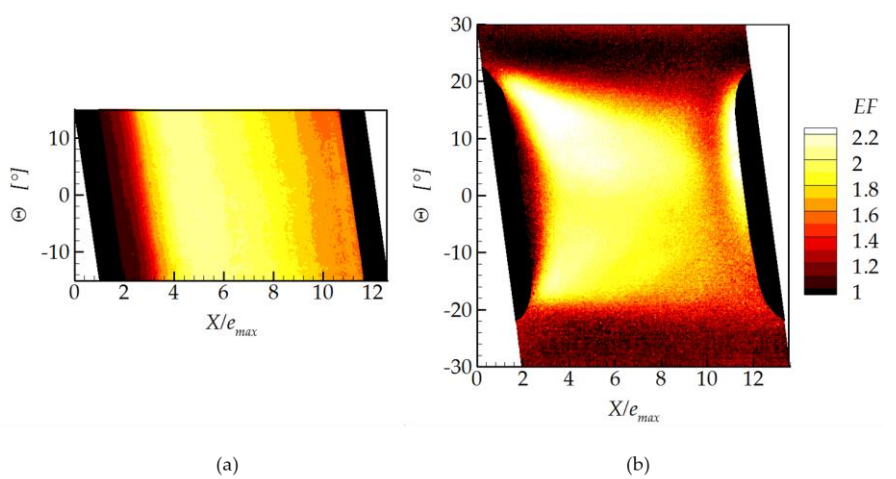


Figure 4.13 - EF distribution between the 7th and the 8th obstacles: left, RIB-1 at $Re=40020$; right, RIB-2 at $Re=41530$

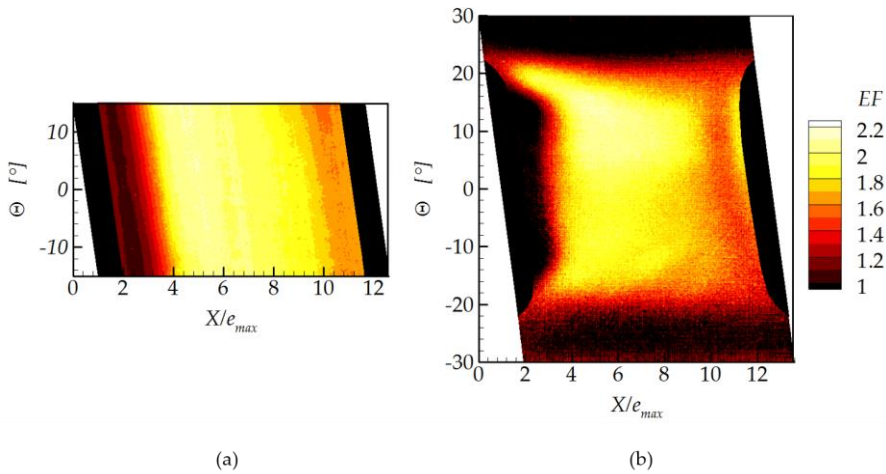


Figure 4.14 - EF distribution between the 7th and the 8th obstacles: left, RIB-1 at $Re=19980$; right, RIB-2 at $Re=21160$

This effect appears to be stronger in RIB-2, in comparison with the continuous turbulator, since the discontinuity of the turbulator lowers the azimuthal component of the recirculating bubble velocity. Hence, in the trailing-edge of the discontinuous obstacles, there is a lower swirl advection that reduces the Nusselt number in that area. Moving from the reattachment towards the next obstacle, the boundary layer develops again, and therefore, the EF decreases from its maximum values. Downstream, the flow bumps against the leading-edge of the subsequent obstacle. The rise in pressure pushes the hot flow from the wall upwards letting the radial heat exchange by convection increase [34]. This is given mainly in the area attached to the leading-edge of the obstacle at the position $\Theta=10^\circ$. In this region, the flow has a large azimuthal velocity component [32], which also contributes to the convective heat transfer. To point out, the flow which impinges on the obstacle leading edge at $\Theta=20^\circ$ moves through the leading-edge following the helix. This influences the EF distribution in the zone upstream the leading edge of the 8th obstacle. Particularly in the area upstream the leading-edge at $\Theta=14^\circ$, the local values of EF reach their local maximum, which equals 2.25 ± 0.35 . The areas which are clear from the presence of

the obstacles show values of EF close to the unity, i.e., $EF=0.90-1.00$ in the area at $\Theta=22^\circ-30^\circ$ and $EF=1.00-1.10$ in the area at $\Theta=-22^\circ--30^\circ$.

Figures 4.12, 4.13, and 4.14 show the effect of lower Reynolds numbers on the Enhancement Factor distribution between the obstacles of both RIB-1 and RIB-2. The results in the figures mentioned above are given in the range $Re=20000-60000$ with a step of 20000, in descending order. Notably, for RIB-1, the reattachment zone gets slightly farther from the trailing-edge of the 7th obstacle, with decreasing Reynolds number. At $Re=19980$, the minimum EF value is no longer located at the trailing-edge of the obstacle, but it is moved at a distance of e_{max} downstream from it. For the case of RIB-2, a significant change in the EF field among the obstacles in not experienced. An exception is the case with $Re=21160$ (Fig. 4.14), where the shape of the reattachment zone significantly changes.

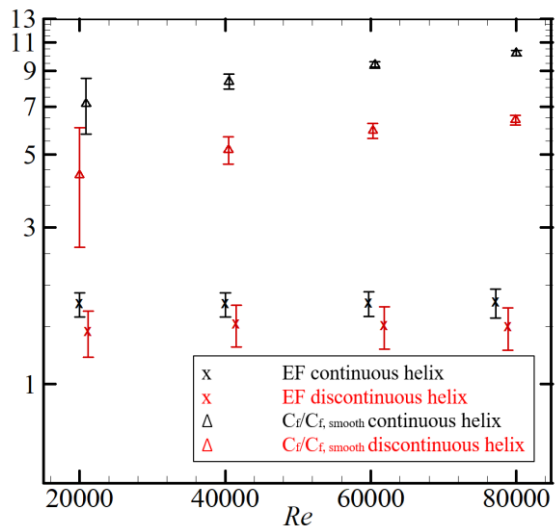


Figure 4.15 - Area-average heat transfer and friction coefficient ratios for different Reynolds numbers: black symbols, RIB-1; red symbols, RIB-2; crosses, area-averaged EF ; triangles, $Cf/Cf,smooth$ [26]

This is mainly given by the fact that at $Re=20000$, the conduction losses significantly influence the heat balance in the trailing edge region. This phenomenon has an effect on the shape of the reattachment zone (see Appendix G). A low heat transfer zone ($EF=0.8$) close to the trailing edge is clearly shown in Figure 4.14-b. Therefore, for the case of the discontinuous turbulator at low Re , the turbulence intensity and the flow structures significantly differ from the case of Reynolds number about 80000, contributing less to the rise in the local heat transfer. The area-average of the heat transfer rate over the Regions of Interest between the adjacent obstacles of RIB-1 and RIB-2 is illustrated in Figure 4.15 in the range $Re=20000-80000$. The values are given in Table 4.1, with the relative uncertainties given by the measurement technique. Since the variation of the mean EF with the Reynolds number is lower than the level of the measurement uncertainty, the two trends are considered not to depend on the flow regime. The continuous geometry has a constant heat transfer level of about $EF=1.75$ with an extended relative uncertainty of 17%, computed with a probability of 95%.

		Obstacles #7 and #8			
<i>Re</i>	<i>Turbulator</i>	$\overline{EF}$	$std(EF)$	$U_{95}(EF)$	Nu_{smooth}
19980	<i>RIB-1</i>	1.75	± 0.31	± 0.15	55.6
21160	<i>RIB-2</i>	1.44	± 0.41	± 0.23	58.2
40020	<i>RIB-1</i>	1.75	± 0.28	± 0.15	96.9
41530	<i>RIB-2</i>	1.52	± 0.39	± 0.22	99.9
59730	<i>RIB-1</i>	1.76	± 0.27	± 0.15	133.5
61850	<i>RIB-2</i>	1.50	± 0.36	± 0.22	137.3
77180	<i>RIB-1</i>	1.77	± 0.28	± 0.18	163.9
78850	<i>RIB-2</i>	1.49	± 0.37	± 0.22	166.8

Table 4.1 - Area-average EF values and their uncertainties as function of Re for RIB-1 and RIB-2

The discontinuous turbulator provides instead a lower constant thermal efficiency of about $EF=1.48$ with a higher uncertainty (27%) compared with the

measurements for RIB-1. The most compelling evidence of such a difference between the surface-averaged EF of the continuous and the discontinuous helicoidal turbulators is explained by the difference in the flow field characteristics.

At Re equal to 20000, stereo-PIV measurements of the flow between the obstacles of the two studied turbulators are provided by the authors in our previous work [26]. In comparison with RIB-1, the discontinuity of RIB-2 lowers the intensity of the tangential component of the velocity field bounded at the wall. In fact, in the inter-rib region between 70% and 95% of the pipe radius, the dimensionless azimuthal velocity component W/U_b generated by RIB-1 reaches values close to 0.2, which are twice as high as the ones observed with RIB-2. The recirculating flow structure behind the RIB-1's obstacles is advected in the azimuthal direction with a higher intensity than for RIB-2.

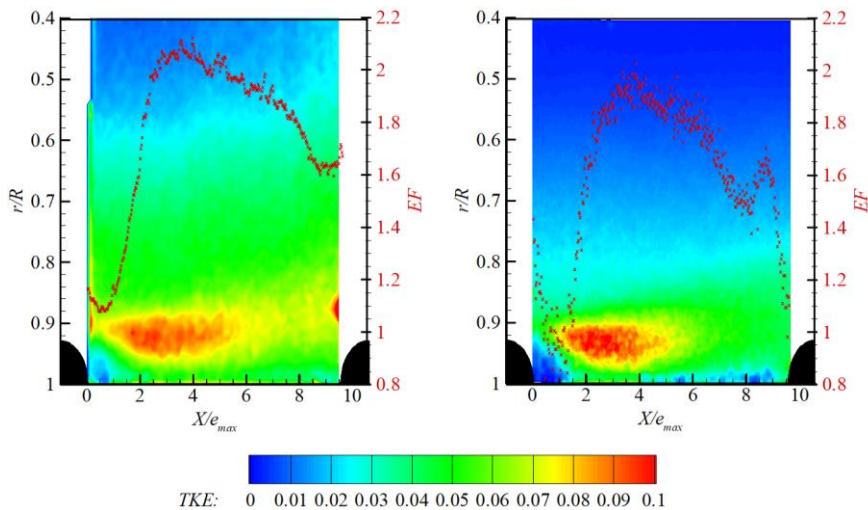


Figure 4.16 – Combination of the TKE with the EF distributions at $Re=20000$:
left, RIB-1; right, RIB-2

Moreover, the authors' results show also a higher Turbulent Kinetic Energy (TKE) in the flow field generated by the continuous turbulator. The dimensionless Turbulent Kinetic Energy is defined as:

$$TKE = 0.5 \frac{(\langle uu \rangle + \langle vv \rangle + \langle ww \rangle)}{U_{bulk}^2} \quad (4.17)$$

where $\langle uu \rangle$, $\langle vv \rangle$ and $\langle ww \rangle$ are the variances of the flow axial, radial, and tangential components, respectively, and U_{bulk} is the bulk velocity. This is shown particularly for the region where the shear layer takes place, between 80% and 95% of the pipe radius. It is shown that the extension of both the TKE's maximum and minimum zones are respectively wider and smaller for RIB-1, in comparison with RIB-2. These levels of the wall-bounded flow fluctuations (TKE) influences the wall heat transfer distribution positively, as shown in Figure 4.16. Therefore, both the higher azimuthal advection of the turbulent structures and the higher TKE shown between the obstacles of the continuous helicoidal turbulator explain the higher level of forced convection at the wall. In addition, the results on the pressure losses provided by the authors [26] are also shown in Figure 4.15. At $Re=20000$, both the turbulators cause a rise in the ratio $C_f/C_{f, smooth}$, at values equal to 7 and 4 for RIB-1 and RIB-2, respectively. The skin friction coefficient for a smooth pipe $C_{f, smooth}$ is evaluated by using the Blasius correlation:

$$C_{f, smooth} = 0.079 \cdot Re^{-0.25} \quad (4.18)$$

For flow regimes with Re higher than 20000, the helicoidal turbulators induce an amplification of the rise in the friction coefficient. Values of $C_f/C_{f, smooth}$ equal to 10.5, and 6.5 at $Re=80000$ are provided by the continuous and the discontinuous turbulators, respectively. This phenomenon is in contrast with the surface-averaged EF , which results to be uniform with the Reynolds number.

4.4.2. Angular dependence

Due to its discontinuity in the geometry, RIB-2 provides different EF profiles, from the trailing to the leading-edge of the obstacles, at different angular positions, Θ . This angular dependence is shown in Figure 4.17, where the EF profiles of RIB-2 at $\Theta=0^\circ$, 15° and -15° are compared with those of RIB-1. Notably, for the latter turbulator, there is no angular dependence, and the EF profile is the same at different Θ 's. The x-axis of Figure 4.17 represents x^* which is defined as

$$x^* = \frac{X/e_{max} - X/e_{max}|_{LE}}{X/e_{max}|_{TE} - X/e_{max}|_{LE}} \quad (4.19)$$

This is a scaled coordinate where 0 and 1 represent respectively, the leading-edge (LE) and the trailing-edge positions (TE). In this way, it is possible to compare the EF profiles of RIB-2 at different angular positions, since they have a different length between the adjacent obstacles $(X/e_{max}|_{TE} - X/e_{max}|_{LE})$.

Figure 4.17-a, b, c and d show the EF profile of RIB-1 at $\Theta=0^\circ$, in the range $Re=20000-80000$ in descending order. The decrease in the Re does not influence the EF neither at the trailing-edge ($EF=1.00$), leading-edge ($EF=1.60$) nor maximum value ($EF=2.00$). The significant difference is in the slope $\partial EF / \partial x^*$ at the TE, which is inversely proportional to the Reynolds number. Therefore, at low turbulent intensities, the peak of EF moves further from the trailing edge of the obstacle. Concerning the discontinuous turbulator, Figure 4.17-e, f, g and h plot the EF profiles at the angular positions $\Theta=0^\circ$ (black symbols), 15° (red symbols) and -15° (blue symbols) at different flow regimes. For the case of $Re=78850$, the heat transfer rate between the obstacles at $\Theta=0^\circ$ (Fig. 4.17-e) shows a trend which is similar to that of RIB-1 (Fig. 4.17-a). The main difference appears close to the leading-edge ($x^*=0.9$) of the next obstacle, where a small peak of $EF=1.80$ occurs. As mentioned above, this is due to the fact that the flow, which is going through the obstacle-free

zone, is impinging to the leading edge of the 8th obstacle. Moreover, the reattachment zone is mainly at a value which is slightly lower than the heat transfer levels of RIB-1 at the same flow regime. At the angular position $\Theta=15^\circ$, which is close to the upper edge of the obstacle, the slope $\partial EF / \partial x^*$ at the TE is quite higher if compared with the one at $\Theta=0^\circ$.

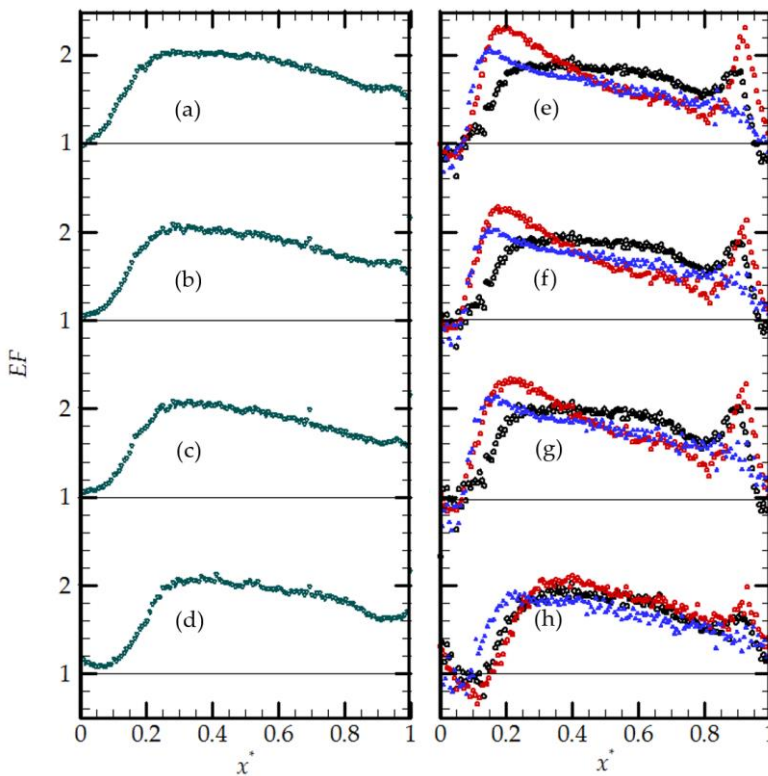


Figure 4.17 - EF profiles at different angular positions between the 7th and the 8th obstacles: left, dark-green triangles for $\Theta=0^\circ$ of RIB-1; right, black circles for $\Theta=0^\circ$, red squares for $\Theta=15^\circ$ and blue triangles for $\Theta=-15^\circ$ of RIB-2; (a), $Re=77200$; (b), $Re=59700$; (c), $Re=40000$; (d), $Re=19900$; (e), $Re=78900$; (f), $Re=61900$; (g), $Re=41500$; (h), $Re=21200$

The EF reaches a peak of 2.38 to then drastically decrease at values close to 1.40 at $x^*=0.8$. A second peak is located close to the leading-edge of the 8th obstacle, to a level higher than that obtained at $\Theta=0^\circ$. The EF profile at $\Theta=-15^\circ$ is similar to that at $\Theta=15^\circ$ without showing a crest close to the obstacle leading-edge. If compared with $\Theta=15^\circ$, the peak at the reattachment zone occurs slightly in advance and at a lower value ($EF=2.00$). In the Figures 4.17-f, g and h the effects of gradually lower Reynolds numbers on the local heat transfer are shown. With exception to $Re=21160$ (Fig. 4.17-h), the EF profiles at $\Theta=0^\circ$ and $\Theta=-15^\circ$ do not show significant variations. The main effect is on the two EF peaks of the profile at $\Theta=15^\circ$, which are lowered and shifted downstream. At $Re=21160$, the two curves at $\Theta=15^\circ$ and $\Theta=-15^\circ$ approximately collapse over the EF profile at $\Theta=0^\circ$. Notably, for this flow regime, the EF downstream the trailing-edge of the 7th obstacle results to be lower than unity. The low level of turbulence intensity [26] and the presence of the obstacle lead to a heat transfer in this area worse than the one obtained in smooth pipes with the same boundary conditions.

4.4.3. Helix-wise dependence

The continuous turbulator provides constant levels of heat transfer (EF) at each line, inside the ROI, parallel to the leading-edge. This is given by the fact that the reattachment length is uniform since the shape and the height of the obstacle is the same throughout the helix-wise direction. Figures 4.18-a, b, c and d show such a behavior of RIB-1 through 3 lines parallel to the trailing edge, which are 0.1, 4.5 and 9 times e_{max} far from the obstacle. Those positions are chosen to represent what happens respectively in the zones close to the trailing-edge (red symbols), to the reattachment (black symbols) and close to the leading-edge (blue symbols). The levels of local heat transfer at the reattachment ($EF=2.00$) and leading edge ($EF=1.60$) do not change with the Reynolds number. On the other hand, this Reynolds number dependence is relevant for the wall heat transfer level at the trailing-edge. The EF equals 1.00 for the case of $Re=77180$ (Fig. 4.18-a), and it

increases to 1.20 by lowering the Reynolds number to a value equal to 19980 (Fig. 4.18-d). This phenomenon is explained, referring to the results shown in Fig.4.17. As a piece of evidence, the EF profiles for RIB-1 are subjected to a shift towards higher values of x^* with decreasing Re . Hence, it can be inferred that at lower flow regimes, the recirculation bubble moves away from the obstacle. In such a way, there is a lower thermal resistance of the massive recirculating zone to the convective heat transfer in the region behind the wall obstacle. Differently, the discontinuity in the geometry of RIB-2 influences such EF plots in the helix-wise direction. In this case, the positions (X/e_{max}) of the three lines mentioned before, which are parallel to the helix path, are computed from the trailing edge of the 7th obstacle at $\Theta=0^\circ$. In the flow regime with $Re=78850$ (Fig. 4.18-e), the EF through the reattachment and the leading-edge lines have a mild and a significant increase respectively, moving from -15° to 15° . The two trends both show a peak at 8° of $EF=2.00$ for the reattachment and $EF=1.80$ for the leading-edge zones. In the same angular range, the wall heat transfer through the trailing-edge line has values close to the smooth channel one ($EF=1.00$).

Inside the rib-free region, i.e., $\Theta=\pm (20^\circ-30^\circ)$, the heat transfer rates of the reattachment and leading-edge lines tend to the unity. Differently, the trailing-edge line shows two EF peaks of 1.80 and 2.00, particularly at $\Theta=-17^\circ$ and 20° . The reduction in the flow velocity, i.e., Reynolds number, let the heat transfer rate at the leading-edge-line decrease. The same occurs slightly for the two peaks shown at the trailing-edge line, which also tend to move away from each other (Fig. 4.18-f, g and h). Notably, at $Re=21160$ (Fig. 4.18-h), the mean value of the EF at the trailing-edge equals 0.80 throughout the whole helix-wise direction. The discontinuity of the RIB-2 configuration contributes negatively to the wall heat transfer rate at the trailing-edge area, when compared to the values of the Dittus-Boelter equation for smooth pipes.

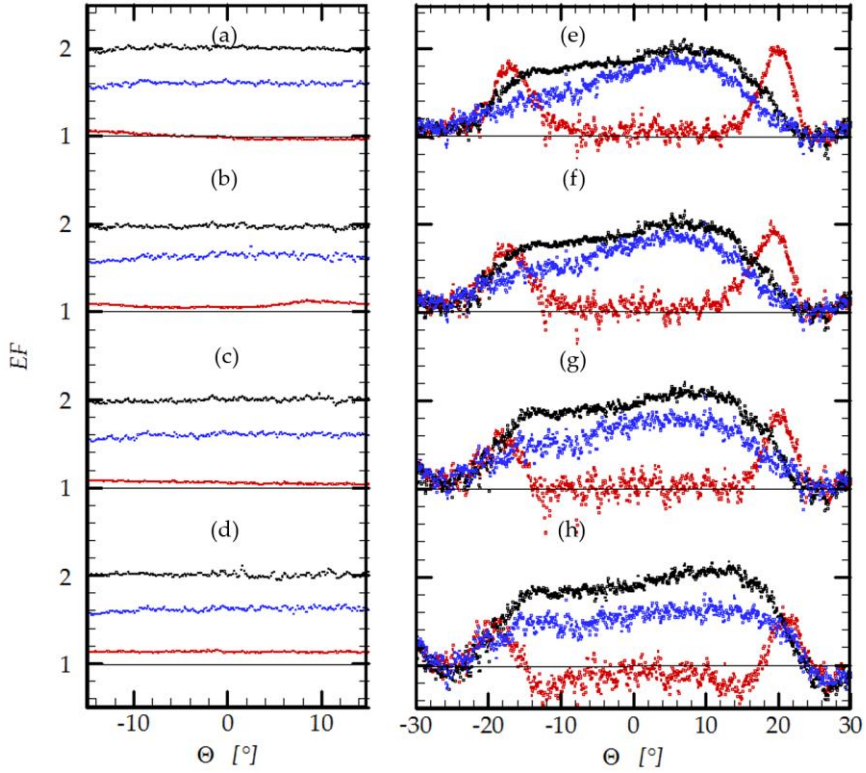


Figure 4.18 - EF profiles at different helix-wise positions between the 7th and the 8th obstacles: red squares for values downstream the trailing-edge, black squares for values close to the reattachment area, blue squares for values upstream the leading-edge; left, RIB-1; right, RIB-2; (a), $Re=77200$; (b), $Re=59700$; (c), $Re=40000$; (d), $Re=19900$; (e), $Re=78900$; (f), $Re=61900$; (g), $Re=41500$; (h), $Re=21200$

4.4.4. Heat transfer periodicity

The periodicity of the phenomenon quantifies the degree of establishment of the flow induced by the repetitive wall obstruction. This can be evaluated by comparing the 3D separation and reattachment effects on the heat transfer distribution between adjacent couples of obstacles. Figure 4.19 compares the EF profiles between the 6th

and 7th obstacles with those between the 7th and the 8th obstacles, for the case of RIB-1. For all the different flow regimes (Fig. 4.19-a, b, c and e) there is generally a reasonable periodicity.

The highest deviation is located at $Re=19980$ in the region close to the heat transfer peak, i.e., $X/e_{max}=0.2-0.4$. This value of 10% is actually of the same order of magnitude of the measurement uncertainty. Figure 4.20 shows the same comparison for the case of RIB-2 at three different angular positions, i.e., $\Theta=0^\circ$, 15° and -15° . The periodicity of the wall heat transfer field of the discontinuous turbulator is validated at the different flow regimes.

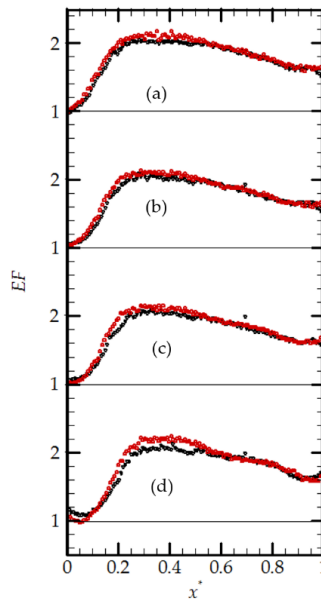


Figure 4.19 – Comparison between EF profiles at $\Theta=0^\circ$ between two adjacent pairs of obstacles of RIB-1: red triangles for values between the 6th and 7th obstacles, black triangles for values between the 7th and 8th obstacles; (a), $Re=77200$; (b), $Re=59700$; (c), $Re=40000$; (d), $Re=19900$

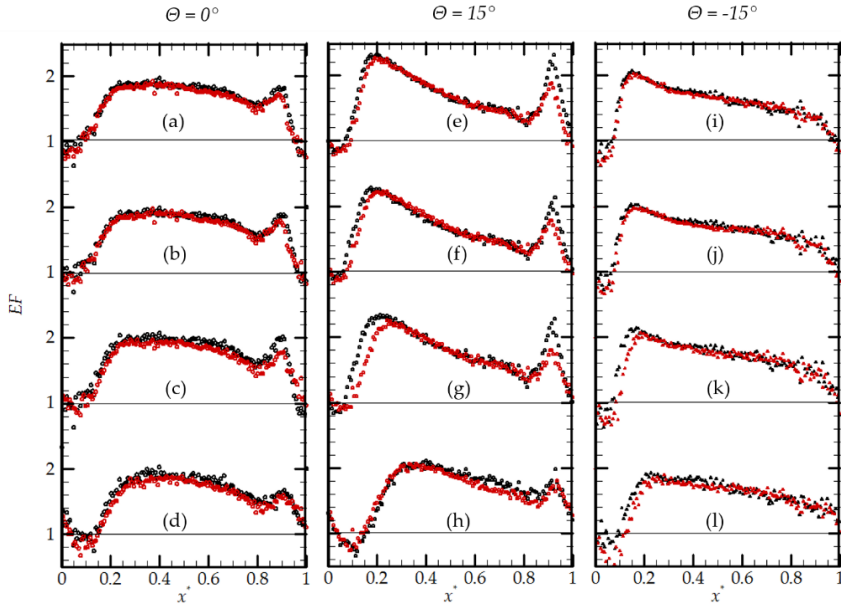


Figure 4.20 - Comparison between EF profiles between two adjacent pairs of obstacles of RIB-2 at different angular positions: red symbols for values between the 6th and 7th obstacles, black symbols for values between the 7th and 8th obstacles; (a)-(e)-(i), $Re=78900$; (b)-(f)-(j), $Re=61900$; (c)-(g)-(k), $Re=41500$; (d)-(h)-(l), $Re=21200$

4.5. Conclusions

The effects of continuous and discontinuous helicoidal ribs, used as wall turbulator in pipe flows, on the local heat transfer are presented in this work. The heat transfer characterization of the two technologies depends on the wall temperature measurements, which are based mainly on the Infrared and Liquid Crystals Thermography. Notably, the authors show the achievement in the application of an experimental technique for the measurement of temperature and Nu fields in complex 3D geometries. The use of TLCs paint over the inner surface of a

transparent pipe, equipped with turbulators, allows the measurements of the surface temperature. An improved methodology for the data processing for the TLCs calibration allows to reduce the uncertainty of the temperature measurement over curved surfaces. Notably, this method expects a calibration line-by-line where the temperature read by the IR camera is matched with the RGB intensities of the respective images at different temperature levels. Extended uncertainties of 15% and 25% are evaluated in the heat transfer measurements for the continuous and discontinuous turbulators, respectively. Comparison with a discontinuous geometry which influences the EF profiles both at different angular positions and helix-wise directions shows that the continuous rib provides a heat transfer rate, which is about 1.2 times the discontinuous one. Therefore the dataset of Mayo et al. [23] for a ROI bounded in a small angular range obtained at $Re=20000$ had to be extended to an angular range of 30° , to a flow regime up to $Re=80000$. In terms of thermal efficiency, the application of continuous helicoidal turbulators is preferred over discontinuous geometries.

4.6. References

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5

Dimples in turbulent pipe flows: experimental aero-thermal investigation

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M. Virgilio, J. N. Dedeyne, K. M. Van Geem, G. B. Marin, T. Arts, Dimples in turbulent pipe flows: experimental aero-thermal investigation, Int. J. of Heat and Mass Transfer, (2020).

Abstract

An experimental investigation is performed to study the effects of turbulent flow structures on the wall heat transfer, specifically, inside the spherical cavities of partially dimpled tubes. The use of dimples in heat exchangers allows achieving a rise in thermal efficiency with the burden of augmenting the pressure losses. Notably, the skin friction coefficient (C_f) of the dimpled pipe is provided and compared with those in ribbed tubes at several flow regimes ($Re=20000-80000$). The geometry under investigation consists of an array of spherical cavities distributed over the inner surface of half a tube. Each row has five indentations separated by an angular offset of 36° over the half circumference. The dimples are characterized by a depth-to-print diameter (d/D_p) of 0.18, a print diameter-to-pipe diameter (D_p/D) of 0.25, and a streamwise distance (x_d/D) between consecutive cavities of 0.56. At the flow regime of $Re=20000$, flow separation, reattachment, and secondary flows are captured using both planar Particle Image Velocimetry (PIV) and stereo-PIV. The time-averaged flow field, the vorticity levels, and Turbulent Kinetic Energy (TKE) fields inside the dimples are provided for the purpose of numerical validations. Besides, the local heat transfer field inside the cavity is given for $Re=20000-80000$, by using the Liquid Crystals Thermography technique (LCTG). At the same Re and Pr , an overall heat transfer enhancement of 1.8 (1.8 times the Nusselt number value (Nu) in bare pipes) has been measured. Higher levels of the aerothermal performance (high heat transfer and low losses) are achieved by employing the dimple technique for tubular heat exchangers, rather than using helicoidal wall inserts.

5.1 Introduction

In 1976, the aerodynamic forces applied to different dimpled golf balls were measured by Bearman and Harvey [1]. The surface depressions were found to induce

a critical Reynolds number at values lower than the one identified on smooth balls. The consequence is that the boundary layer does not remain laminar up to the flow separation. In detail, the turbulence generated by the cavities strengthens the boundary layer which remains attached to the wall with a higher resistance. Therefore, the wake behind the dimpled sphere is reduced without the drag penalties associated with other trips, which would protrude from the surface. This phenomenon explains why a dimpled golf ball can be driven further than a bare one. Another peculiarity of dimples is to promote the ejection of vortex pairs, which are advected in the streamwise direction, leading the local turbulent transport to increase [2]. Correspondingly, the wall reattachment of the shear layer, separated at the leading edge of the spherical indentations, also contributes to the local rise of the Nusselt number (Nu). This is given by the impingement of the separated flow to the rear zone of the dimple, which contributes in the heat removal by convection. With this “golf ball effect” in mind, a massive interest grew in adopting this technique in place of other methodologies for the enhancement of heat exchanger thermal efficiencies [3]. The surface corrugation enhancement technique, by the use of arrays of dimples, has a variety of possible applications, including heat exchangers, bio-medical devices, electronics cooling, chemical reactors [4], gas turbines [5] and microfluidics.

The separation-reattachment event, which disrupts the turbulent thermal boundary layer, is also obtained with other heat augmentation devices, such as rib turbulators [6]. As highlighted before, the main difference is to be found in the pressure drop penalties, which are lower for the case of shallow indentations. In this friction reduction investigation, Lienhart et al. [7] put in evidence, by means of both Direct Numerical Simulations (DNS) and experiments, that the contribution from the shear stress is decreased. Nonetheless, the effect on the drag is overwhelmed by the involvement of new pressure forces generated by the dimples. A similar investigation on the application of spherical cavities on flat plates was carried out by Afanasyev et al. [8]. Differently from Lienhart [7], a geometry with shallow indentations was chosen in a way to avoid flow with a breakaway, so, the depth of the dimples

($d/D_p < 0.1$) did not induce any flow separation. Nevertheless, the roughness of the wall let the thermal resistance of the viscous sublayer decrease, providing a rise in heat transfer by convection. This was shown experimentally by the maximum displacement of the axial velocity and temperature pulsations, toward the flow close to the wall. The rise in the Stanton number (St) of about 30%-40% for different Reynolds numbers (Re) was shown not to occur with any appreciable increase in friction factor. Similarly to the case of a flat plate, the study of Burgess et al. [9] analyzed the effects of the dimple depth in the application of turbulent square channel flows. The authors investigated the behavior of the local Nusselt number in the flow regime $Re=12000-70000$ of channels with a height-to-dimple print diameter ratio (d/D_p) of 1.00. In this case, deeper spherical cavities generated stronger unsteady pairs of shedding vortices and more robust shear layers. As a consequence, the local flow ejection from the deeper dimples was more persistent, resulting in turbulent structures with stronger secondary flows (Figure 5.1). Larger volumes of fluid were moved into and out of the cavity, letting the local Nu be three times higher than in the case of smooth channels at the same flow regime. The increased turbulent production led the peak of heat transfer to be located in the downstream portion of the dimples.

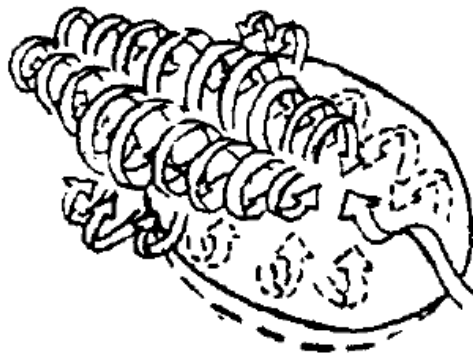


Figure 5.1 – Turbulent structures inside one dimple [10]

A more detailed picture of the unsteady turbulent flow structures was provided by Ligrani et al. [10] with smoke visualizations in a dimpled channel. The smoke line trajectories showed inside the cavities both a recirculation zone and pairs of secondary flows in the streamwise and spanwise-normal planes, respectively. At a later stage, the flow showed a strong up-wash region, where a primary pair of counter-rotating vortices were ejected upward from the inside of the cavity. Therefore, the mushroom-shaped smoke pattern was advected downstream, becoming smaller in size and stretched with a higher level of vorticity (Figure 5.1), before entering the following indentations. Additionally, two secondary pairs of vortices were generated and ejected from the region near the right and left spanwise edges of a dimple. The shedding process of the primary and secondary vortical fluid structures involved the whole array of dimples, contributing principally to the rise in the global heat transfer rate and turbulence transport in the flow surrounding the obstacles [11]. Notably, the shedding of the vortices was irregular in the sense that one of the two vortices became stronger than the other and got ejected downstream of the dimple. This phenomenon occurs alternatively, with a constant frequency, which depends on the Reynold number and the channel height-to-print diameter ratio (hc/D_p). Particle Image Velocimetry (PIV) performed by Zhou [12] is in agreement with the turbulent structures encountered by the smoke visualizations described before. The work of Tay et al. [13] defined this phenomenon as a “tornado-like vortex”, which generates a rise in both heat transfer and hydraulic resistance of the flow. Correspondingly, Turnow et al. [14] came up with the same conclusions, using Large Eddy Simulations (LES) of turbulent flow through a channel enhanced with deep dimples. Only the first mode, of a Proper Orthogonal Decomposition (POD) of the flow, shows the existence of large rotating structures, originated within the cavity, and successively expelled outward. The turbulent flow was nevertheless dominated by small scales since the POD provided a Turbulent Kinetic Energy (TKE) equally distributed across all the modes. This is not in line with what was claimed by Ligrani et al. [10], where the flow inside the spherical cavities was dominated by pairs of counter-rotating vortices. However, there is a good agreement in the levels of enhancement in heat transfer and friction coefficients,

which results to be respectively, 1.93 and 3.69 times the values seen for a smooth channel. Those results are for channels equipped with an array of dimples whose depth-to-print diameter is 0.26, for a flow regime of $Re=13000$.

Considering that the results found in the literature suggest the dimples to be the wall obstacles which provide the best thermo-hydraulic performance [15], a strong effort was put in their shape optimization [16]. Particularly, the interest was to maximize the overall heat transfer, keeping the friction losses as low as possible. For the case of turbulent channel flows, Chen et al. [17] proposed an asymmetric dimple, which intensified the strength of the pairs of vortices inside the cavity. According to the author, this can be achieved mainly by skewing the deepest point of the spherical indentation in the downstream direction. In the literature, alternative solutions, for example, the use of tear-drop shaped cavities [15] and planar inserts located inside symmetric dimples [18], are considered to modify both the separation-reattachment phenomenon and improve the heat transfer. Likewise, V-shaped arrays of dimples in channels were tested by Jordan and Wright [5] using the transient Thermochromic Liquid Crystals (TLCs) technique. The measurement of the Nu distribution over the enhanced surface confirmed the heat transfer augmentation when compared to smooth channels.

Considering the results mentioned here above, efficiencies of tubular heat exchangers are also considered for improvements, taking advantage of the wall dimpling technique (Figure 5.2). In fact, in the field of steam cracking processes, tubular reactors with corrugated walls are able to decrease their inner wall temperature during the cracking of hydrocarbons [19]. This reduces the rate of coke deposition, the latter being an unwanted product during chemical reactions. Considering that, also, pressure drop penalties negatively contribute to the selectivity of desired products, the choice of using dimples as heat transfer augmentation devices has significant potential. The tornado-like turbulent structures in the flow within the dimples are also encountered in augmented tubular reactors by Dedeyne et al. [4]. The

authors made a sensitivity analysis employing LES to simulate turbulent flows in tubular systems, dimpled in several ways, to find out the most performant configuration.

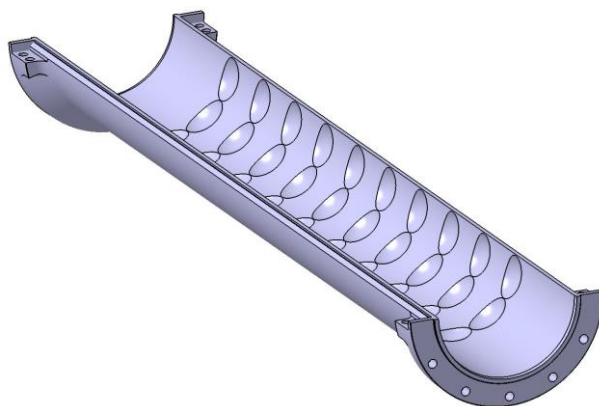


Figure 5.2 – Tube enhanced with dimples

The numerical results concerning the use of dimples in pipes for heat transfer improvements suffer from a lack of validation. To the best of the author's knowledge, while many experimental investigations were focused on differently dimpled square channels and flat plates, little aero-thermal data are provided for the case of tubes. The enhancement of heat transfer was measured for a dimpled pipe by Bunker and Donnellan [20]. Employing steady-state Liquid Crystals Thermography (LCTG), the authors measured the external temperature of six dimpled tubes. Subsequently, the convective heat transfer coefficient was obtained by removing an estimation of the heat losses by conduction, without solving the energy equation at the inner wall of the test section. Therefore, a spatially resolved heat transfer distribution across the spherical cavities was not provided. In this manner, the results suffered from a lack of accuracy, since the inner wall temperature field on the dimpled surface was not measured. Besides, quantitative experimental flow characterizations of the turbulent structures within spherical dimples inside tubes are not reported in the literature.

Starting from all these considerations, this chapter aims at providing spatially resolved experiments to investigate the aerothermal characteristics of the turbulent flow inside dimpled tubes. PIV and stereo-PIV techniques were chosen to investigate the vortical structures and turbulent statistics respectively within streamwise and spanwise planes, at the flow regime of about $Re=20000$. The effects of the dimples on the local heat transfer fields were measured with steady-state TLCs, measuring the inner wall temperature all over the enhanced tube, at the range $Re=20000-80000$. These experimental results are considered as datasets for the validation of numerical simulations.

5.2 Experimental Methodology

5.2.1 Geometry under investigation

The tube under investigation has a surface corrugation made of spherical cavities, each of them obtained by the solid intersection between a sphere (radius of 30.4 mm) and the inner surface of the pipe (diameter of 150 mm). This Boolean operation subtracts material from the pipe in a way that a spherical indentation with a depth-to-print diameter (d/D_p) of 0.18 and a print diameter-to-pipe diameter (D_p/D) of 0.25 is generated (Figure 5.3, 5.4). The hindrance given to the pipe flow by this negative obstacle has a blockage ratio (d/D) of 5%.

Only one half of the tube is dimpled, providing an array of 10 rows with 5 spherical cavities in each row (Fig. 5.4), for a total of 50 sharp-edge dimples. The streamwise distance between consecutive dimples is 80mm ($x_d/D = 0.56$), while the angular interspace among the cavities in the circumferential direction is 36° . The choice of an in-line configuration, rather than a staggered layout, allows having an optimum in terms of heat transfer efficiency and pressure losses [4].

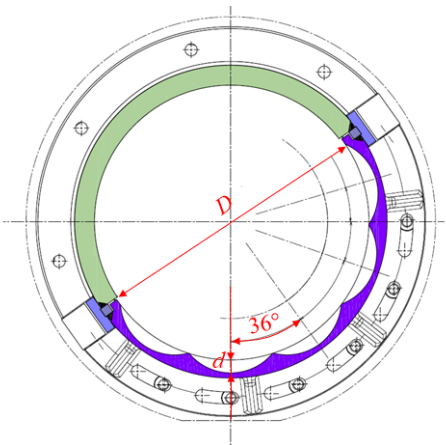


Figure 5.3 – Cross-section of the dimpled tube

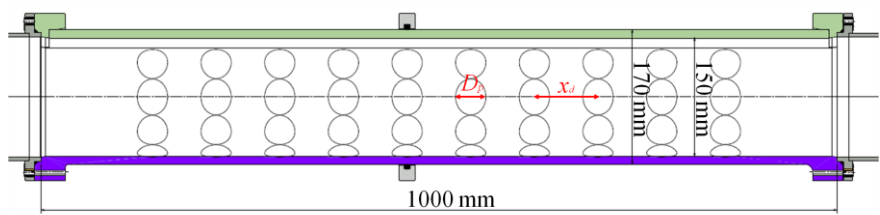


Figure 4.4 – Dimpled tube under investigation

Furthermore, a clustered circumferential disposition generates more massive secondary flows. The latter also contribute to a better mixing and turbulent production inside the entire cross-section of the tube. This is in agreement with the numerical investigation of Dedeyne et al. [4]. The author found out, by comparing LES of both fully and partially dimpled tubes, that the geometry with clustered dimples generates secondary flows of Prandtl’s second kind [4], which contribute to the whole cross-section mixing. Notably, this is generated by the level of imbalance in the Reynolds stresses distribution over the corrugated surface. Therefore, if compared to geometries

with uniformly distributed dimples, the clustered layout generates higher levels of TKE inside the pipe flow.

5.2.2 Heat transfer and pressure loss measurements

The heat transfer and pressure loss measurements of the dimpled pipe are performed in a low-speed wind tunnel (Figure 5.5), which consists of a cylindrical circuit with a uniform diameter of 150mm. This facility, located in the “TU - Heat and Mass Transfer Laboratory” of the von Karman Institute for Fluid Dynamics, allows both pressure loss and heat transfer measurements. The operating status of the facility is in a suction mode, with the airflow generated by a centrifugal fan located at the exit of the facility. This system avoids having the inlet flow affected by both discharge fluctuations and swirl components generated by the blower. The inlet section has a length of 15 pipe diameters [21] to provide a fully developed turbulent boundary layer entering the test section. To measure the temperature of the air on the centerline of the pipe, two k-type thermocouples are inserted upstream and downstream of the test section (test section length = 980mm). To ensure optical access inside of the pipe, the test section was manufactured by casting, in vacuum conditions, transparent ABS-like polyurethane material. The master model was produced from stereo lithography of the dimpled pipe, with a high-quality surface finish. Then, a mold was produced by casting silicon around the master, under vacuum, to avoid air bubbles. In this way, after curing, the master was removed, leaving a cavity ready for the casting of the transparent final material. The cylinder is made of two halves (Figure 5.3) in order to provide access to the inner surface for the application of layers of different materials. More specifically, a coating of wide-range TLCs is applied on the transparent inner surface to be able to measure its temperature field. A layer of black matt paint is sprayed on top of the liquid crystals to guarantee a high contrast of the scattered colors. One last layer of (Rust-Oleum 2178) anti-heat reflections topcoat is applied on top of the black paint, to provide the dimpled surface with an emissivity close to the unity.

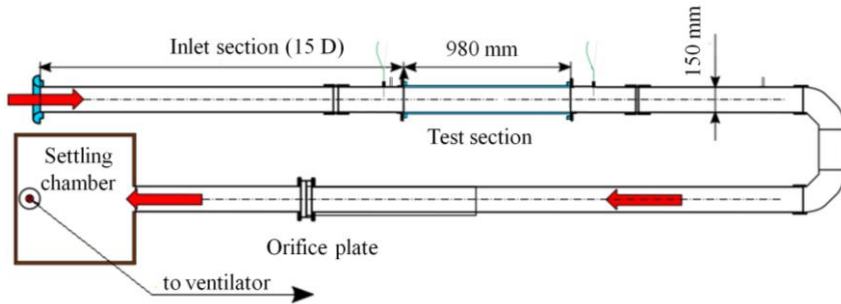


Figure 5.5 – Low-speed wind tunnel for heat transfer measurements [22]

For the pressure loss measurements, the facility is equipped with static pressure taps just upstream and downstream of the test section. Those pressure taps are located at the top of the pipe, and they are connected to an ultra-low differential pressure calibrator DPI 610 LP (Druck). In such a way, the pressure calibrator measures the static differential pressure in the range $(-250-250)$ Pa with an uncertainty of ± 0.25 Pa. The heating system consists mainly of a transparent conductive film (TCF) glued on the outer surface of the test section. This TCF (resistivity of $0.1 \Omega/\text{m}^2$) is an optical PET film (thickness of 0.1 mm) deposited with a copper grid and protected with a nickel layer. This system guarantees transparency higher than 80%. The application of a DC voltage to the TCF's extremities allows having a heating system that provides uniform heat flux as a boundary condition for the heat transfer measurements. A diffusive LED light panel (Pan-70W) illuminates the whole test section, surrounded by black panels to avoid any light reflections. The colors scattered by the TLCs are recorded as RGB intensities of the sensor's pixels, by a Nikon D7100 camera (Figure 5.6). The latter is located at a distance equal to 850 mm from the test section and at an angle of 45° to the horizon. On the same side of the photographic equipment, an infrared camera (FLIR A655 SC) is mounted to measure the temperature of the test section's external surface. Downstream of the dimpled pipe, after a U-bend and a pipe with a length of 30 D, an orifice plate with corner tappings is mounted. According to the ISO normative 5167 [23], this arrangement allows to retrieve the mass-flow via

the pressure drop, read by a water manometer (resolution of 1 Pa). Hence, the bulk velocity and the Reynolds number are evaluated with a relative uncertainty of 1.5%. As last component of the low-speed wind tunnel, a settling chamber is located 4 pipe diameters downstream of the orifice plate. Since it is connected to the blower, its purpose is to damp the fluctuations generated by the radial fan, which may affect the pressure measurements of the orifice plate.

The TLCs used to measure the temperature distribution over the dimpled surface have a red-start at 308 K and a color-play of 20 K. The red-start is the temperature value at which the liquid crystals start to be optically active, scattering mainly the lowest wave-length of the visible spectrum. The blue-stop, which is 20 K higher than the red-start, is the maximum temperature at which the liquid crystals are thermochromic. The Nikon D7100 camera records the light scattered by using an objective of 105mm (Nikon AF DC-Nikkor) from a distance of 850mm. In order to enhance the signal to noise ratio and to have highly saturated colors, the camera's parameters were set at an ISO sensitivity of 100, an $f/14$ aperture, and a shutter speed of 3 seconds. High values of the color saturation of the image allow having a Liquid Crystal Thermography with a minimized uncertainty [24].

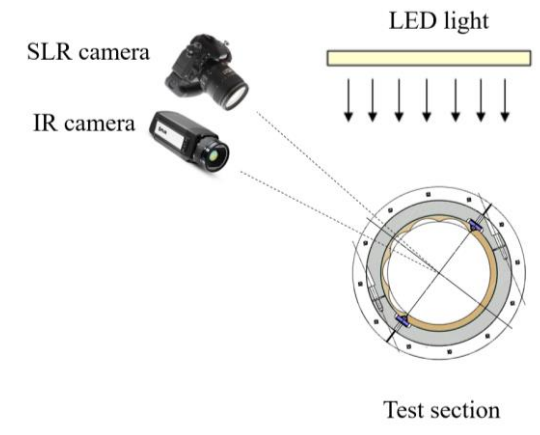


Figure 5.6 – Camera layout for the heat transfer measurements

Furthermore, the white balance of the photographic equipment was set at 4170 K, the same as the color temperature of the light source [25]. The RGB intensities of the acquired images are converted in an alternative three-value combination, called Hue-Saturation-Intensity (HSI), by using the Hollingsworth equation [26]. The Hue value is a number representing the color wavelength of the light scattered by the liquid crystals. Notably, its main property is to have a monotonic variation with the temperature. In the in-situ calibration process, the smooth half of the test section is removed to have the optical access for the IR camera, which is aligned with the digital one (Figure 5.7). The IR camera was a-priori calibrated using a target, coated with the same black matt as in the dimpled test section, and 3 thermocouples placed in different locations. This process guarantees an uncertainty on the temperature measurements by the IR camera of about ± 0.4 K, with a confidence interval equal to 95%. The Hue distribution over the dimpled inner surface of the tube is a function not only of the temperature but also of the pixels' coordinates. This is given by the Hue angular dependency, investigated by Kodzwa & Eaton [24]. Indeed, since the angle between the viewing direction of the camera and the surface normal is not uniform over the investigated surface, the Hue distribution depends also on the pixels' coordinates. To avoid such a dependency, since each point of the dimpled surface has not the same normal vector, each pixel of the RGB images has an own calibration curve [24].

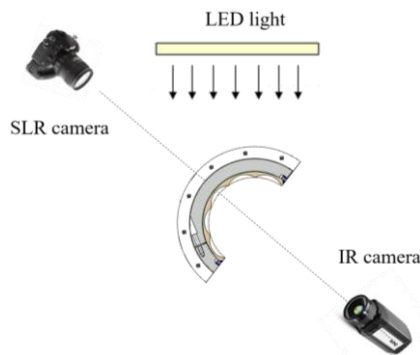


Figure 5.7 – Camera layout for the in-line calibration

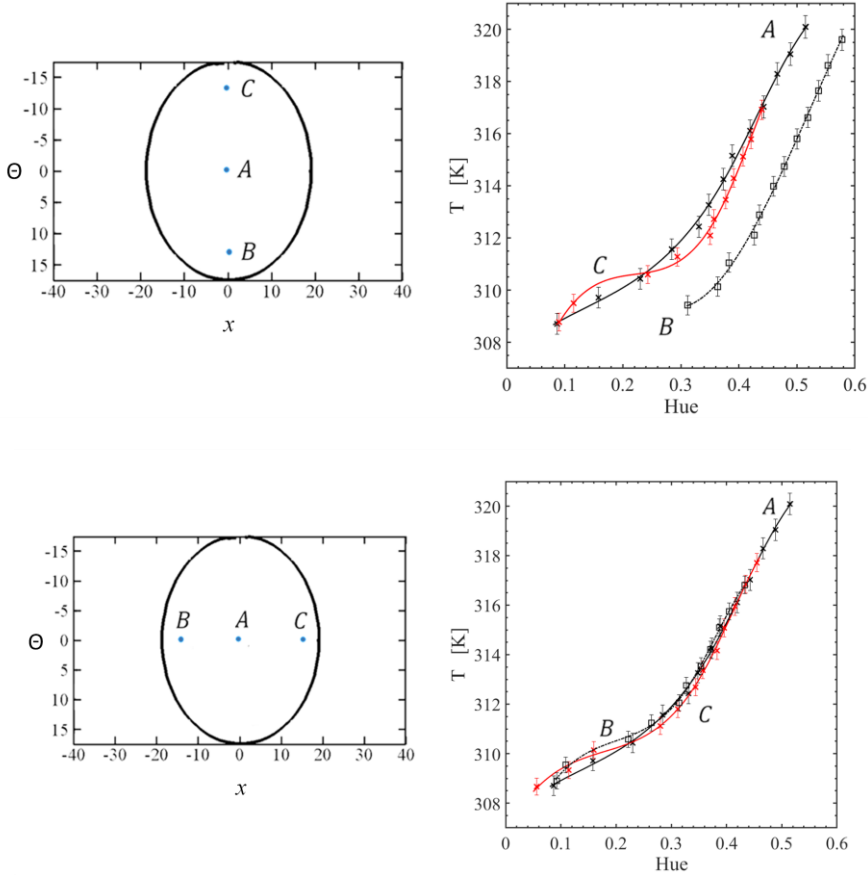


Figure 5.8 – Calibration curves for different points inside the ROI: black solid line, calibration fit in A; black dash-dotted line, calibration fit in B; red solid line, calibration fit in C; black crosses, calibration points for A; black squares, calibration points in B; red crosses, calibration points in C; up, TLCs dependency on the polar direction; bottom, TLCs dependency on the azimuthal direction

The calibration consists in matching pixel-by-pixel the Hue value with the temperature read by the IR camera, for a total of 15 calibration temperatures, in the range (309-323 K). This is done for a zone, called Region of Interest (ROI), of 80 mm

in the axial direction and 36° in the circumferential direction (Figure 5.8), with the origin located at the center of the dimple. The ROI, which is the region where the Nu field is measured, contains 1400×2400 pixels, and since all the images get resized by a factor of 10, the methodology consists of 33600 calibration curves (Figure. 5.8). This technique provides an extended uncertainty in the range $\pm (0.4-0.7)$ K, depending on the temperature value and the pixel position over the dimpled surface. The error sources and the uncertainty quantification procedure are documented by Mayo et al. [27].

5.2.3 Flow velocimetry

The flow-field measurements were performed in the low-speed water tunnel (Figure 5.9), also located in the “TU - Heat and Mass Transfer Laboratory”. A centrifugal pump delivers the flow to a settling reservoir, where the water level is kept constant by means of a by-pass overflow tube. This arrangement allows maintaining a steady state inlet total pressure, i.e., a constant operating Re . The latter is a function only of the water column height and temperature, measured by one K-type thermocouple. The Re is computed using the bulk velocity, given by an orifice plate, and the pipe diameter. From the settling chamber, the water enters the inlet duct, passing through a honeycomb, which suppresses any swirling components generated in the entry region. The inlet section has a uniform diameter of 150 mm and a length of 15 diameters. As for the heat transfer facility, this length allows the development of a turbulent flow [21] entering the following test section. The dimpled tube consists of one smooth half made of acrylic glass and the other enhanced half made of transparent plastic, which was 3D printed with stereo lithography. The transparencies of the two halves leads the laser plane to pass through the dimpled surface and the camera to have optical access inside the dimples (Figure 5.10).

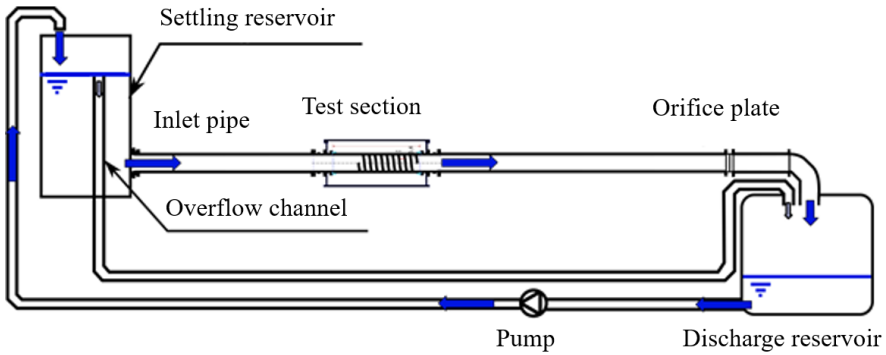


Figure 5.9 – Low-speed water tunnel for flow velocimetry [22]

The optical access for both the planar and stereo PIV configurations is provided through the smooth half of the tube, made of acrylic glass. Since its refractive index equals 1.49, the use of air as working fluid (refractive index of 1.00) would have given substantial distortions of the particles images. For this reason, water was employed, which has a refractive index ($n_R=1.33$) at an ambient temperature closer to the one of Plexiglas. This choice is in-line with the work of van Doorne & Westerweel [28], who employed water to measure with stereo-PIV the laminar state and the turbulence inside pipe flows.

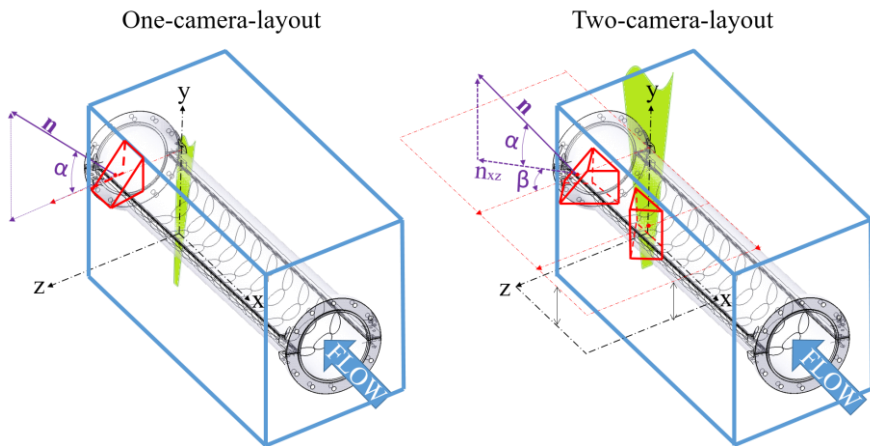


Figure 5.10 – Water prisms in the PIV (left) and stereo-PIV (right) layouts

However, to overcome the 10 % difference in the refractive indexes, the remaining image distortions are corrected with an in-situ calibration by using a 3D polynomial mapping function [29]. In both the PIV and stereo-PIV configurations, the in-situ calibration follows the method suggested by Soloff et al. [30]. The process uses a dotted grid that is placed inside the test section and adjusted in the middle of the laser sheet. Any errors generated by a misalignment are subsequently lowered by applying the self-calibration technique [31], which computes the disparity vectors and applies changes to the 3D mapping calibration for the error compensation. As shown in Figure 5.11, the calibration plate is parallel and perpendicular to the x-y plane during, respectively, the PIV and stereo-PIV calibrations. The dimpled half of the test section (Figure 5.11) is removed, and the dotted plate is inserted inside for the calibration of the camera's system. The grid is made of 50×50 circular dots with a radius of 0.5mm and a uniform displacement of 2.5mm in both the orthonormal directions of the calibration plate. The positioning of the dotted grid is adjusted using a linear translation stage (Figure 5.11), equipped with a micrometer (resolution of 0.01 mm).

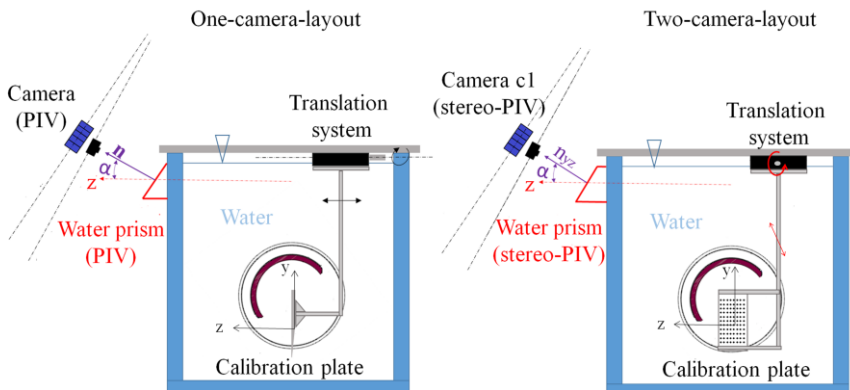


Figure 5.11 – Calibration plates layouts for PIV (left) and stereo-PIV (right)

The 2nd order polynomial mapping function is obtained by using the commercial software Davis 8.4.0 (LaVision) [32]. In the stereoscopic system, 3 different parallel positions of the calibration plate are acquired. Those are equally spaced with a

displacement of 0.3mm in the z -direction inside the thickness of the laser sheet. One of the 3 calibration positions has to be at the center of the laser sheet ($z=0$). For the one-camera-layout, the calibration plate consists of a transparent plastic sheet with printed black dots glued on a planar metal surface, which is painted in white to enhance the contrast. Differently, in the two-camera-layout, the dotted transparent film is inserted between two acrylic plastic sheets, both of them having a thickness of 0.5 mm. In such a way, the two cameras look at the same grid position with the same image distortions. The same procedure was applied by van Doorne and Westerweel [28]. To further lower the distortions that the cylindrical surface might generate, the test section is embedded in a transparent tank filled with water. As in the air facility, the mass flow is obtained by measuring the static pressure drop generated by an orifice plate, positioned 20 pipe diameters downstream of the test section outlet. According to the normative ISO 5167 [23], this mass flow meter was designed in order to minimize the relative measurement uncertainty (1.5% of the reading). The flow is finally collected in a discharge reservoir and pumped again into the settling chamber.

A double pulse ND:YAG laser (Quantel Tween BSL 200) illuminates, with a wave-length of 532 nm and an energy of 70 mJ per pulse, the region under investigation inside the dimples. A telescopic system ($f = 300\text{mm}$ - 2000mm), composed by two spherical lenses, and a cylindrical lens ($f = -20\text{mm}$) generates a focused light plane (thickness = 1.5mm), which is turned by 90° by a mirror, inserted in a gimbal mount. The turbulent flow field investigation expects the employment of PIV (one-camera-layout) to measure the 2 velocity components in two different stream-wise positions, i.e., Plane A and B (Figure 5.12). The former is the y - x plane of symmetry, while the latter is obtained by the clockwise rotation of the test section about the pipe axis of 13° , towards the dimple's edge. For a different purpose, stereo-PIV (two-camera-layout) is used to investigate the turbulent flow characteristics to quantify the 3 velocity components in two different span-wise positions, i.e., Plane C and D (Figure 5.12). The former is the y - z plane of symmetry, while the latter is obtained by a 10 mm stream-wise translation of Plane C.

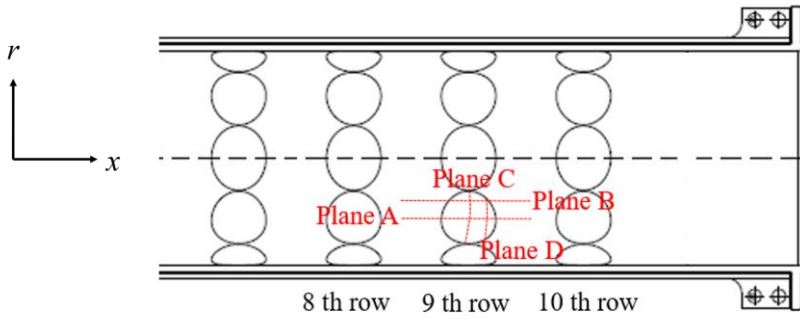


Figure 5.12 – Position of the laser planes for the particle image acquisitions

In the one-camera-layout (Figure 5.10), one double-shutter CCD camera with a $2360 \times 1776 \text{ pix}^2$ sensor is placed on one side of the test section with an angle of 30° to the horizon. This camera is provided with a Nikon Nikkor lens of 50mm, set to an aperture of $f/8$, mounted at a distance of 350mm from the target. To compensate substantial radial distortions, the photographic equipment is arranged with a Scheimpflug mount and a liquid prism. The Scheimpflug mount allows the sensor's plane rotating around one axis, in order to maintain a focused image across the whole field of view [33]. The liquid prism is a thin-walled (5mm) Plexiglas flask filled with water, which is stuck to the water tank of the test section. Its purpose is to provide an orthogonal viewing to the angular camera, reducing the radial distortions given by off-axis angles subtended by the particles to the camera. This PIV layout provides image particle sizes in the range of 3-6 pixels and a resulting overall Magnification Factor (MF) of 38 pix/mm . In the two-camera-layout (Figure 5.10), the stereo-PIV system uses two double-shutter CCD cameras placed on the same side of the test section, both with an angle of 30° to the horizon. The angle between the two cameras is 90° , i.e., each camera has an angle of 45° with respect to the azimuth (Figure 5.13), as in the work of van Doorne and Westerweel [28]. This stereo angle provides increased sensitivity to the stream-wise velocity component, perpendicular to the laser sheet plane.

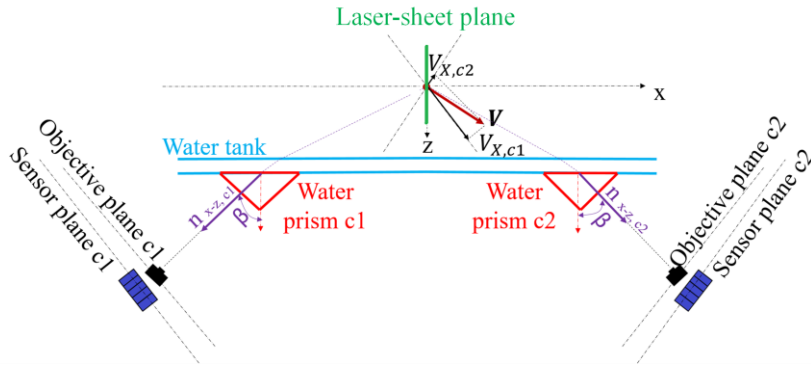


Figure 5.13 – Two-camera-layout for stereo-PIV

As in the one-camera-layout, the stereo-PIV system is equipped with two Scheimpflug adapters, one for each camera. Moreover, two water prisms are used, which take into account the angular positions of the cameras, concerning both the horizon and the azimuth. The objectives (Nikon Nikkor) have a focal of 50mm and an aperture of $f/11$. This stereo-PIV system yields particles diameter in the range of 4-10 pixels and a Magnification Factor (MF) of 33 pix/mm. In order to reach statistical convergence [34], 1000 not time-correlated PIV realizations are acquired at a sampling frequency of 1Hz (Ub/D) for both camera layouts. A separation time of 2.6ms, between two successive frames of each PIV realization, was chosen to have, on average, a particle displacement of about 10 pixels. The cross-correlation algorithm processes the images in three refinement steps with a final interrogation window size of 32×32 pix² and with an overlap of 75%. The final resolutions, linked to the interrogation window sizes are about 0.85×0.85 and 1×1 mm², for the one-camera and two-camera layouts, respectively. To minimize the background reflections of the green laser light, fluorescent polyethylene spheres (Cospheric), with a diameter in the range 10-45 μ m, are used to seed the flow. A 532 nm notch filters are placed in front of the cameras' sensors to cut off the green light. This solution allows recording only the red light scattered by the fluorescent particles. Despite the higher cost, such a technique allows augmenting the signal-to-noise ratio of the PIV realizations [35]. In

order to respect the rule of thumb of having at least 10 particles per interrogation window, the flow was seeded up to a concentration of 30 g/m³. In Appendix B, the effects of the imaging parameters on the reconstructed error and correlation map are shown for both the one and two cameras configurations.

5.3 Data Reduction

5.3.1 Skin friction coefficient

For turbulent boundary layers, the skin friction coefficient (C_f) in pipes is defined as follows [36]:

$$C_f = \frac{\bar{\tau}_{wall}}{0.5 \cdot \rho \cdot U_b^2} \quad (5.1)$$

where $\bar{\tau}_{wall}$, ρ , and U_b are the wall shear stress, the fluid density and the bulk velocity, respectively. Whereas the bulk velocity is obtained from the orifice plate, the wall shear stress is computed as follows:

$$\tau_{wall} = \frac{\Delta p_{static} \cdot R}{2L} \quad (5.2)$$

where Δp_{static} is the difference between the static pressures upstream and downstream the dimpled test section; R and L are respectively, the radius and the length of the pipe. Equation 5.2 results from solving the momentum balance in a control volume extended to the whole test section; therefore, it provides a surface averaged value of the wall shear stress. To validate the measurements, the methodology was applied to a smooth pipe, and compared with the correlation for smooth pipes [36]:

$$C_{f,smooth} = \frac{0.079}{Re^{0.25}} \quad (5.3)$$

5.3.2 Mean flow, vorticity and turbulence statistics

The three components of the mean flow result from a time-average of all the time-uncorrelated samples:

$$U(\mathbf{x}) = \frac{\sum_{i=1}^{N_t} (U_i(\mathbf{x}, t))}{N_t} \quad (5.4)$$

$$V(\mathbf{x}) = \frac{\sum_{i=1}^{N_t} (V_i(\mathbf{x}, t))}{N_t} \quad (5.5)$$

$$W(\mathbf{x}) = \frac{\sum_{i=1}^{N_t} (W_i(\mathbf{x}, t))}{N_t} \quad (5.6)$$

where U , V , and W are the velocity components in the x , y , and z directions, respectively; $\mathbf{x}$ results to be either (x,y) , for the one-camera-layout, or (y,z) for the two-camera-layout, and N_t equals the number of temporal acquisitions. Since the flow inside the cavity is characterized by recirculating flows [4, 10] which influence the wall heat transfer [2], it is also significant to evaluate the flow vorticity component ω in the z direction:

$$\omega(x, y) = \frac{1}{2} \left(\frac{\partial V(x, y)}{\partial x} - \frac{\partial U(x, y)}{\partial y} \right) \quad (5.7)$$

The partial derivatives of the time-averaged velocity components are computed by using a second-order central scheme [32]. Another important parameter to characterize the influence of the turbulent structures on the local heat transfer is the Turbulent Kinetic Energy (TKE) [36], defined as:

$$TKE = \frac{1}{2} \cdot \left(\frac{\langle uu \rangle + \langle vv \rangle + \langle ww \rangle}{U_b^2} \right) \quad (5.8)$$

For the case of the PIV measurements, since the third component (W) is not measured, an in-plane TKE is introduced [37]:

$$TKE(x, y) = \frac{3}{4} \cdot \left(\frac{\langle uu \rangle + \langle vv \rangle}{U_b^2} \right) \quad (5.9)$$

In Equations 5.8 and 5.9, the flow fluctuations in x and y are computed as the variances of the three velocity components:

$$\langle uu \rangle = \frac{\sum_{i=1}^{N_t} (U_i(\mathbf{x}, t) - U(\mathbf{x})) \cdot (U_i(\mathbf{x}, t) - U(\mathbf{x}))}{N_t} \quad (5.10)$$

$$\langle vv \rangle = \frac{\sum_{i=1}^{N_t} (V_i(\mathbf{x}, t) - V(\mathbf{x})) \cdot (V_i(\mathbf{x}, t) - V(\mathbf{x}))}{N_t} \quad (5.11)$$

5.3.3 Enhancement Factor

In the literature, the parameter assessing the local rise in the heat rate over the ROI is the Enhancement Factor (EF):

$$EF(x, \theta) = \frac{Nu(x, \theta)}{Nu_s(x, \theta)} \quad (5.12)$$

it is the ratio of the Nu over the same number for a smooth pipe (Nu_s), at the same flow regime (Re), and with identical fluid properties (Pr). The denominator of Equation 5.12 is computed using the Dittus-Boelter correlation [21]:

$$Nu_s = 0.023 \cdot Re^{0.8} Pr^{0.4} \quad (5.13)$$

The Nusselt number field over the ROI (Figure 5.8) is computed as:

$$Nu(x, \theta) = \frac{h(x, \theta) \cdot D}{k_f(T_b)} \quad (5.14)$$

where k_f is the air thermal conductivity, function of the bulk temperature (T_b) [38], x and θ are the axial and tangential coordinates inside the ROI, and h is the convective heat transfer coefficient, computed as:

$$h(x, \theta) = \frac{\dot{q}_{conv}(x, \theta)}{[T_w(x, \theta) - T_c(x)]} \quad (5.15)$$

As discussed in Chapter 3, the bulk temperature (T_b) is supposed to be equal to the temperature at the center of the pipe (T_c). The error given by this assumption is taken into account inside the final uncertainty estimation. In Equation 5.15, the numerator is the wall heat flux transferred to the flow by forced convection ($\dot{q}_{conv}$); while the denominator is the difference between the local wall temperature (T_w) and the airflow temperature on the pipe axis (T_c) at the same x coordinate. The wall convective heat flux is evaluated by solving an energy balance for each finite volume (see Figure 5.14) of the test section (thermal conductivity $k_{TS}=0.228$ W/m² K). Inside the spherical cavity, the energy equation is given by:

$$\begin{aligned} \dot{q}_{conv}(x, \theta)A_c = & \dot{q}_{cond,\rho}(x, \theta)A_\rho + \partial \dot{q}_{cond,\varphi}(x, \theta)A_\varphi \\ & + \partial \dot{q}_{cond,\gamma}(x, \theta)A_\gamma \end{aligned} \quad (5.16)$$

while in the dimple-free region:

$$\begin{aligned}\dot{q}_{conv}(x, \theta)A_c &= \dot{q}_{cond,\rho}(x, \theta)A_\rho + \partial \dot{q}_{cond,x}(x, \theta)A_x \\ &+ \partial \dot{q}_{cond,\theta}(x, \theta)A_\theta\end{aligned}\quad (5.17)$$

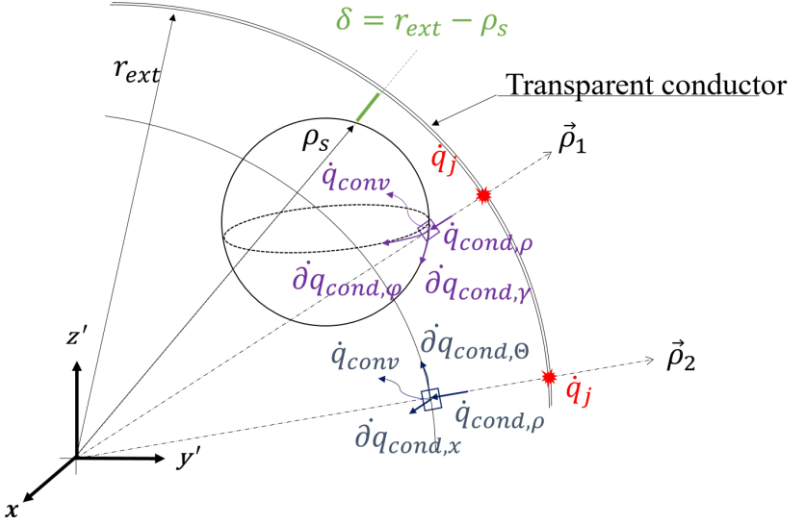


Figure 5.14 – Energy balances inside and outside one dimple

In Equations 5.16 and 5.17, the net heat fluxes (W/m²) are computed by using a numerical scheme based on central differences of the temperatures on the inner wall:

$$\partial \dot{q}_{cond,x} = \frac{\partial \dot{q}_{cond,x}}{\partial x} dx = \frac{\partial}{\partial x} \left(-k_{TS} \frac{\partial T_w(x, \theta)}{\partial x} \right) dx \quad (5.18)$$

$$\begin{aligned}\partial \dot{q}_{cond,\theta} &= \frac{1}{r_{int}} \frac{\partial \dot{q}_{cond,\theta}}{\partial \theta} r_{int} d\theta \\ &= \frac{1}{r_{int}} \frac{\partial}{\partial \theta} \left(-k_{TS} \frac{1}{r_{int}} \frac{\partial T_w(x, \theta)}{\partial \theta} \right) r_{int} d\theta\end{aligned}\quad (5.19)$$

$$\partial \dot{q}_{cond,\varphi} = \frac{1}{r_s} \frac{\partial \dot{q}_{cond,\varphi}}{\partial \varphi} r_s d\varphi = \frac{1}{r_s} \frac{\partial}{\partial \varphi} \left(-k_{TS} \frac{1}{r_s} \frac{\partial T_w(x, \theta)}{\partial \varphi} \right) r_s d\varphi \quad (5.20)$$

$$\begin{aligned} \partial \dot{q}_{cond,\gamma} &= \frac{1}{r_s \sin \varphi} \frac{\partial \dot{q}_\gamma}{\partial \gamma} r_s \sin \varphi d\gamma \\ &= \frac{1}{r_s \sin \varphi} \frac{\partial}{\partial \gamma} \left(-k_{TS} \frac{1}{r_s} \frac{\partial T_w(x, \theta)}{\partial \gamma} \right) r_s \sin \varphi d\gamma \end{aligned} \quad (5.21)$$

where (x, y', z') , (x, ρ, θ) , and (r_s, φ, γ) are the Cartesian, cylindrical, and spherical reference systems (Figure 5.15), respectively. The finite differences are computed on a Gaussian filtered temperature field, to smooth out the errors in the scattered data of the ROI. As in the work of Bozzoli et al. [39], the size of the kernel was chosen in a way that the filtered temperature field deviates from the original one with a difference, which is lower than the LCTG uncertainty. Furthermore, the radial conductive heat flux ($\dot{q}_{cond,\rho}$) is calculated as the difference between the external ($T_{w,ext}$) and internal (T_w) temperatures, divided by the thickness of the dimpled tube (δ):

$$\dot{q}_{cond,\rho} = -k_{TS} \frac{T_{w,ext}(x, \theta) - T_w(x, \theta)}{\delta(x, \theta)} \quad (5.22)$$

Owing to the presence of the dimples, the thickness of the pipe is not uniform, so it depends on the position, in terms of x and θ (Figure 5.15), inside the ROI. In Appendix H, the wall heat flux fields of the 3 conductive components are shown.

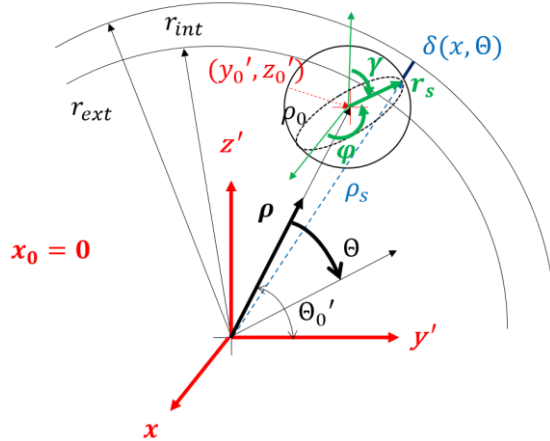


Figure 5.15 – Coordinate systems: Cartesian (x, y', z'), bold red; Cylindrical (x, ρ, θ), bold black; Spherical (r_s, ϕ, γ), bold green

5.4 Results and discussion

5.4.1 Effects of the dimpling technique on the pressure losses

As already mentioned, the enhancing techniques employed to augment the wall heat transfer of thermal systems have the drawback of introducing additional pressure losses to the flow. In Virgilio et al. [22], the use of continuous and discontinuous helicoidal turbulators provide a skin friction coefficient, respectively, 7 and 4 times higher than the one measured in smooth pipes.

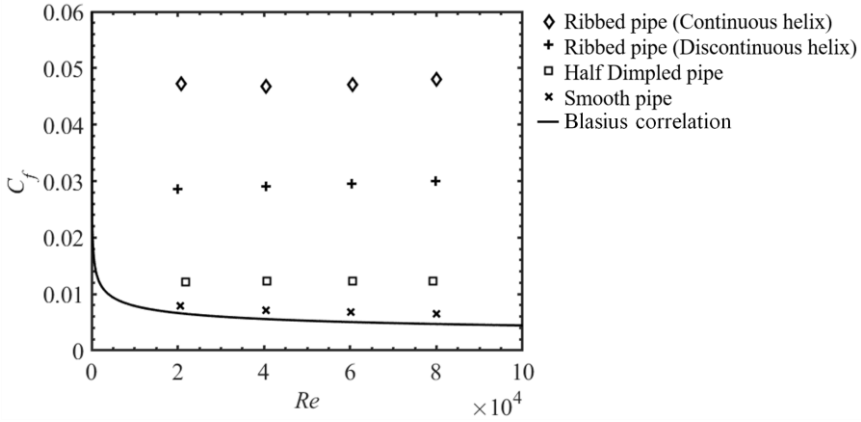


Figure 5.16 – Skin friction coefficient as a function of Reynolds number: solid line, Blasius correlation [36]; crosses, smooth channel [22]; squares, half dimpled pipe; plus signs, discontinuous helicoidal rib [22]; diamonds, continuous helicoidal rib [22]

Differently, the dimpling technique hereby proposed has a lower impact, only doubling the value measured in a smooth pipe ($C_{f,smooth}$). Figure 5.16 compares the skin friction values of the dimpled pipe with those of two differently helically ribbed pipes. For each type of turbulator, no significant influence of the Reynolds number is identified in the skin friction coefficients. The extended uncertainties on the measure of the C_f for the dimpled pipe result to be 46.4%, 13.4%, 6.2%, and 3.8% relatively for $Re=21600$, 43700, 61300, and 77700.

5.4.2 Turbulent structures influence on the dimple's wall heat transfer

Both the PIV and stereo-PIV measurements of the turbulent flow fields, respectively, the planes A-B and C-D, were performed for the dimple at the 9th row,

just adjacent to the dimple in the plane of symmetry (see Figure 5.12). This choice was made considering that the optical access, inside the dimple placed at the center of the five cavities at the 9th row, was not possible. The velocity field related to the stream-wise Plane A, which divides the spherical cavity into two halves, is shown in Figure 5.17 (a). A 3D separation occurs at the leading edge of the dimple, with the following flow reattachment at the dimple's rim, downstream the zero-velocity line (solid red line in Figure 5.17 (a)). Below the dividing line, a 'mean' recirculating bubble is captured by the time-averaged flow field, inside the dimple. The same flow structure is seen in the works of Dedeyne et al. [4] and Turnow et al. [14]. The *EF* profile at the wall (green symbols) has a maximum of 3.25 ± 0.25 in correspondence to the reattachment region, slightly upstream of the intersection between the wall and the zero-velocity line. This is explained considering that the flow reattachment at the wall reduces the thickness of the inner boundary layer [40] and, hence, leads the local convection to increase [40]. Another key point is related to the link between the Nusselt number and the unsteady mass exchange. Colder fluid particles are pushed down to the rim of the cavity's wall, and are mixed with the hotter flow bounded to the wall. This phenomenon is illustrated in Figure 5.17 (c), where positive radial velocity components are shown in the wall-bounded region inside the range $x = 4\text{--}12$ mm. In this process, the excess fluid heats up at the wall temperature and transports the heat outside the cavity when ejected in the core flow, beyond the shear layer.

The region of reverse flow, which goes from the reattachment to the trailing edge of the dimple, is characterized by a thickening of the boundary layer [40]. Therefore, the wall heat transfer by convection is lowered, namely linearly, from the *EF* maximum value to its local minimum. The latter equals to 0.80 ± 0.10 , and appears just downstream of the separation point of the flow in Plane A. The time-averaged flow field in the stream-wise plane closer to the dimple's edge (Plane B) is shown in Figure 5.17 (b, d).

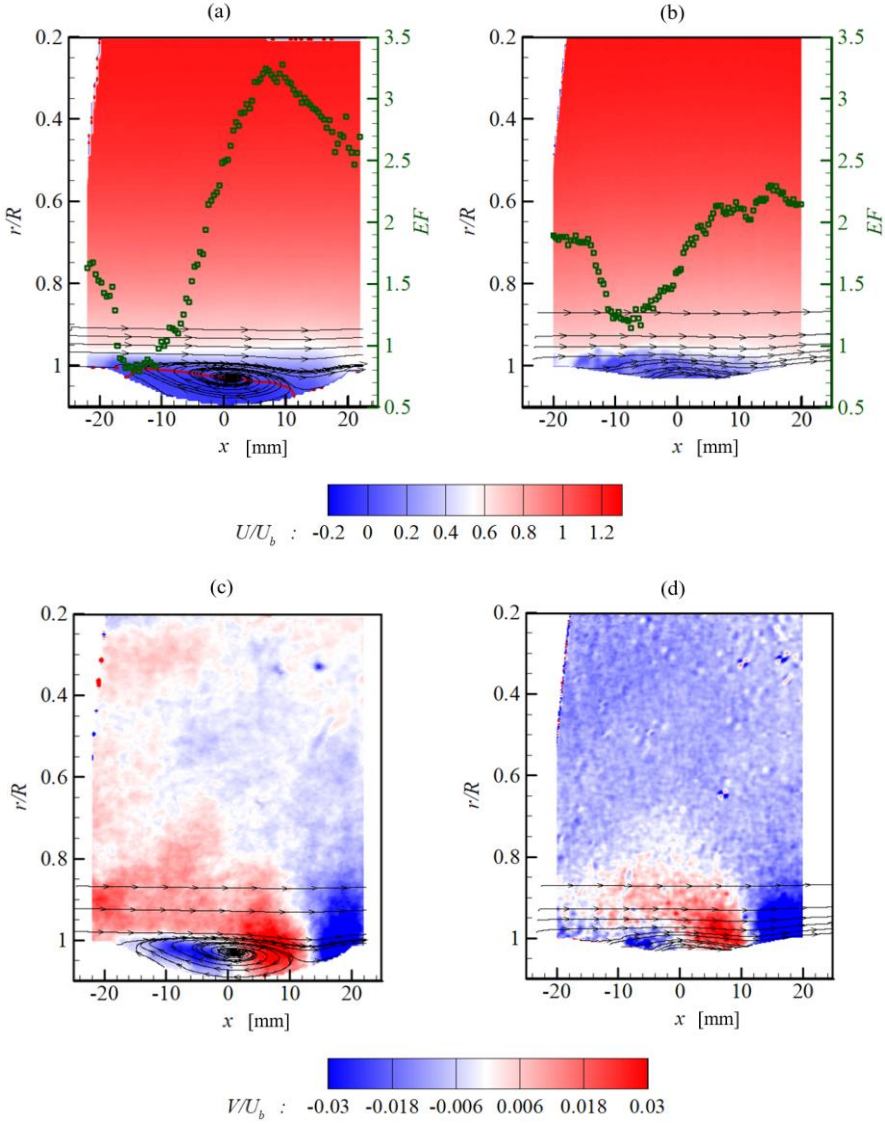


Figure 5.17 – Axial and radial velocities in the stream-wise planes: solid red line, zero-velocity line; solid black line, streamlines; green squares, local EF ; (a), axial velocity for plane A at $Re=24800$; (b), axial velocity for plane B at $Re=23500$; (c), radial velocity for plane A at $Re=24800$; (d), radial velocity for plane B at $Re=23500$

In this position, the mean flow does not show any zero-velocity line, i.e., there is no trace of a ‘mean’ recirculation zone. Despite this, the streamlines of the mean flow deviate further from the wall at $x = -10\text{mm}$, close to the leading edge of the dimple at $\Theta=13^\circ$. At this position, the local EF presents a local minimum of 1.30 ± 0.12 . Subsequently, in the stream-wise direction the EF augments to values equal to 2.20 ± 0.18 . Interesting is the correlation between, respectively, the local EF ’s minimum and maximum positions with the zones of negative and positive radial velocity components (Figure 5.17 (d)). As already mentioned before, volumes of fluid moving into the cavity participate in cooling the wall before being ejected outward. Both the maximum value of EF and its difference with the local minimum value of EF in Plane B are lower than those in Plane A. The author infers that this is due to the lack of a mean separation and reattachment, with reverse flow. This means that, the time-averaged flow field does not capture the influence of the swirling ejected turbulent structures. They indeed show recirculating zones in the flow field of a limited number of instantaneous PIV acquisitions. However, it is relevant to bear in mind that the streamlines of Figure 5.17 (b, d) are given by the time average of instantaneous fields characterized by intermittent vortices, as shown by Ligrani et al. [10] and Dedeyne et al. [4]. Those unsteady secondary flows are generated inside the cavity and then ejected towards the adjacent dimples, providing both a rise in the velocity fluctuations and the local wall heat transfer. Correspondingly, such an influence of the intermittent secondary flows, ejected through Plane B, on the heat removal by convection is linked with significant levels of vorticity at the wall region (Figure 5.18 (b)). Differently, on Plane A, high levels of negative vorticity are bounded inside the shear layer, which is separating the recirculation zones from the flow outside the spherical cavity (Figure 5.18 (a)). Besides, below the zero-velocity line, inside the recirculating bubble, mild levels of positive vorticity are given. Hence, agglomerates of fluid particles rotate counter-clockwise in the reverse flow region, participating in the heat removal from the wall.

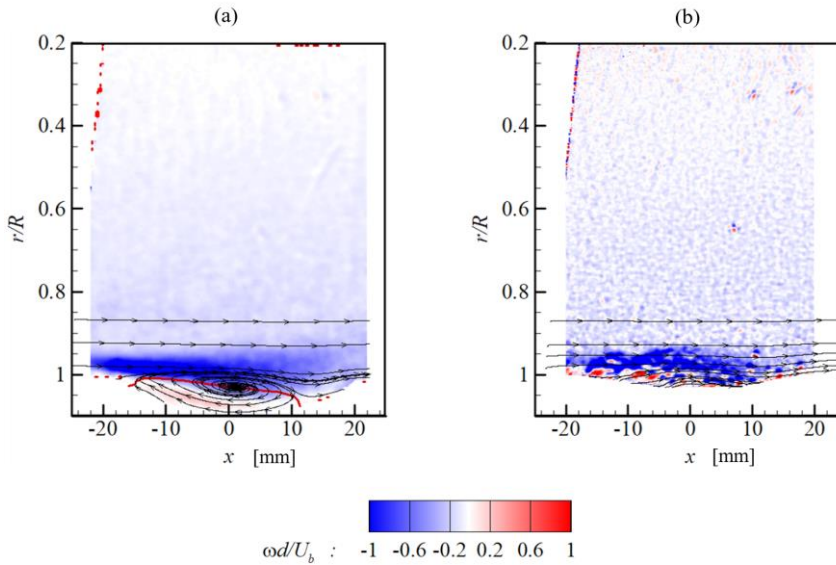


Figure 5.18 – Dimensionless vorticity fields inside the dimple: solid red line, zero-velocity line; solid black line, streamlines; (a), vorticity in plane A at $Re=24800$; (b), vorticity inside plane B at $Re=23500$

Figure 5.19 shows the axial velocity fields and flow patterns in both the spanwise plane of symmetry (Plane C) and the plane passing through the reattachment region at $x = 10\text{mm}$ (Plane D) parallel to the previous one. For the case of Plane C, the zero-velocity line (solid red line in Figure 5.19 (a)) separates the flow with a positive axial component from the reverse flow. Above this separating line, the time-averaged streamlines show the presence of a secondary flow, characterized by a focal point at $\Theta = -10^\circ$, and flow ejections to the adjacent dimples. In the average field, the mushroom-like vortex is skewed towards the left corner, influencing the symmetry of the EF field slightly. Moving from the EF local peak ($EF = 2.50 \pm 0.25$) at $\Theta = 0^\circ$ to the left dimple's edge, the heat transfer rate first decreases linearly till the presence of the focal node of the vortex, then suddenly rises to values of $EF = 2.00 \pm 0.12$. This does not occur at the right dimple's edge, where the EF reaches a value equal to 1.40 ± 0.13 . Differently, in Plane D the streamlines show the reattachment point at the

wall and corner separation close to the cavity’s edges. The *EF* has a parabola-like shape with a maximum value of 3.30 ± 0.23 in correspondence of the reattachment point at the wall (Figure 5.19 (b)). The Turbulent Kinetic Energy generated by the turbulent flow inside the dimple is shown in Figure 5.20 for, particularly, Planes A (Figure 5.20 (a)) and B (Figure 5.20 (b)).

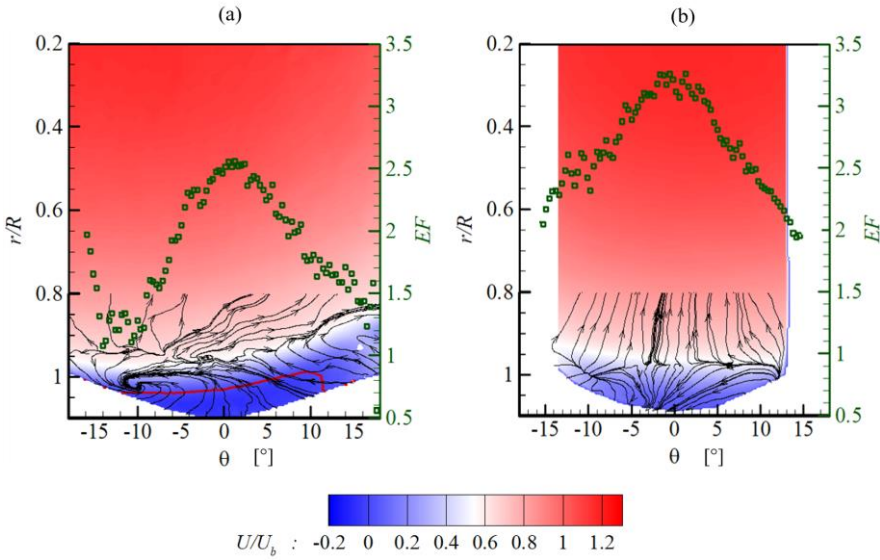


Figure 5.19 - Axial velocity fields in the spanwise planes: solid red line, zero-velocity line; solid line black, streamlines; green squares, local *EF*; (a), axial velocity for plane C at $Re=27900$; (b), axial velocity for plane D at $Re=30900$

In the streamwise plane of symmetry (Plane A), the TKE is generated by the intermittent shear layer, just above the spherical indentation. Differently, for Plane B, which is closer to the edge with the adjacent dimple, higher values of TKE (5% of the bulk kinetic energy) are bounded in the region close to the wall. Although in this plane there is a lack of separation and shear layer, flow fluctuations arise inside the indentation due to the ejection of intermittent flow structures [4, 14]. Generally speaking, flows having the peculiarity of high TKE values close to the wall, yield

higher convection rates, keeping the level of EF higher than unity [10]. The TKE of the flow field inside plane C and D is shown in Appendix I.

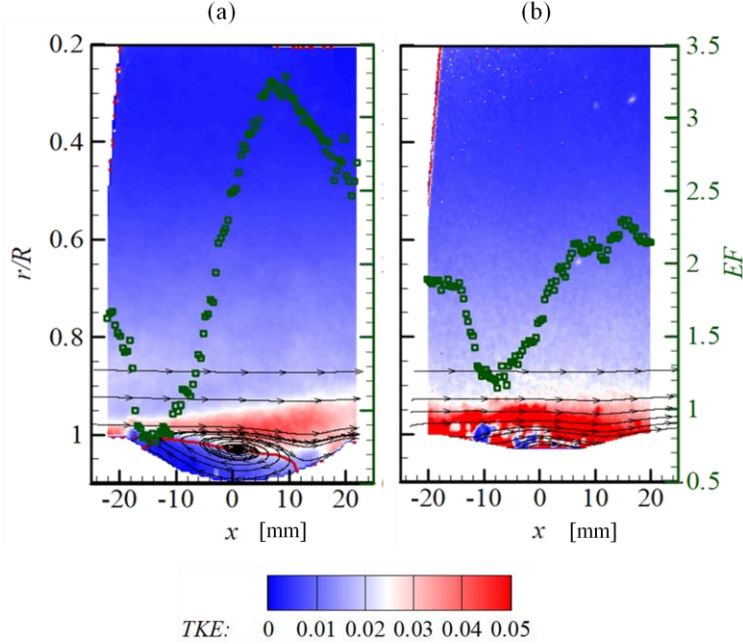


Figure 5.20 – Turbulent Kinetic Energy fields inside the dimple: solid red line, zero-velocity line; solid black line, streamlines; green squares, local EF ; (a), TKE for plane A at $Re=24800$; (b), TKE for plane B at $Re=23500$

5.4.3 Wall heat transfer field at $Re=20000$

At the flow regime of namely $Re=20000$, the EF field over the spherical cavity placed at the 9th row, adjacent to the dimple in the plane of symmetry (of the whole test section), is illustrated in Figure 5.21 (a). As a remark, the flow field described previously is the one generated inside this cavity of the dimples' array. Besides, in Figure 5.21 (b), the same EF distribution is shown for the dimple placed in the plane of symmetry for a comparison at the same flow regime. The relative uncertainties on

the EF measurements are given for each position of the two ROI's in Figure 5.21 (c, d). Both the dimples present the EF maximum value in the region of reattachment, between the center of the cavity and its trailing edge. Their minimum values are measured close to the separation points (dimples' leading edge). Behind the cavities, the effect of the wake is shown on the dimple-clear surface, where the thin turbulent boundary layer is redeveloping towards the following row of dimples. At this location, the EF equals values in the range of $2.00\text{--}2.60 \pm (0.15\text{--}0.20)$. A region characterized by an EF in the range $0.80\text{--}1.30$ takes place in the recirculation region, on the part of the spherical wall close to the leading edge. This zone appears to be smaller and slightly more symmetric (with respect to Plane A), for the dimple in the center of the 9th row. In addition, higher values of EF are measured just downstream of the trailing edge of the same dimple (Figure 5.21 (b)). The only significant difference in the measurement uncertainty of EF over the two dimples is shown in the region just upstream of the trailing edge (Figure 5.21 (c, d)).

A more detailed comparison is illustrated in Figure 5.22 for different spanwise and stream-wise positions inside the ROI's of the two dimples at $Re=20000$. A high level of similarity is shown in the stream-wise planes $\Theta=7^\circ$ (Figure 5.22 (b)) and $\Theta=14^\circ$ (Figure 5.22 (d)) and in the span-wise planes $x=10$ mm (Figure 5.22 (g)) and $x=25$ mm (Figure 5.22 (i)). On the contrary, in the planes of symmetry $\Theta=0^\circ$ (Figure 5.22 (a)) and $x=0$ mm (Figure 5.22 (f)), particularly, and in the other planes $\Theta=-7^\circ$ (Figure 5.22 (c)), $\Theta=-14^\circ$ (Figure 5.22 (e)), $x=-10$ mm (Figure 5.22 (h)) and $x=-25$ mm (Figure 5.22 (j)) dissimilarities are presented. This is given by the fact that at $Re=20000$, there is an influence of the different local dimples' pressure boundaries (since the pipe is not fully dimpled) on the flow structure and relative effect on the wall heat transfer.

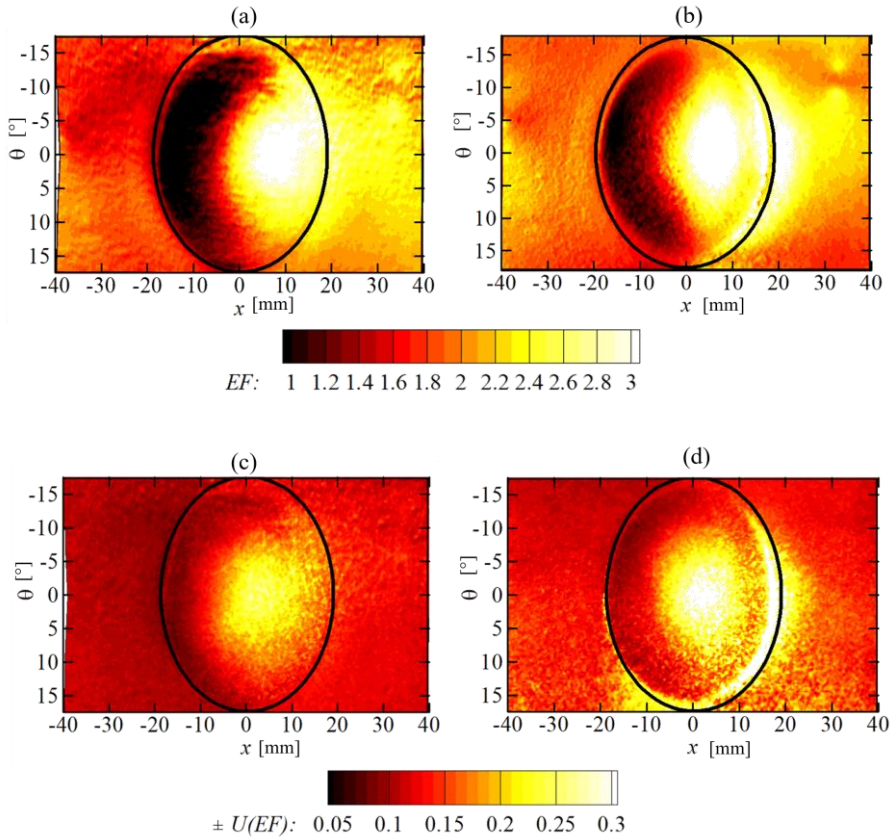


Figure 5.21 – Enhancement Factor field inside the dimples at $Re=21600$: (a), dimple adjacent to the cavity placed at the plane of symmetry; (b), dimple at the plane of symmetry; (c), uncertainty of the EF for the dimple adjacent to the cavity placed at the plane of symmetry; (d), uncertainty of the EF for the dimple placed at the plane of symmetry

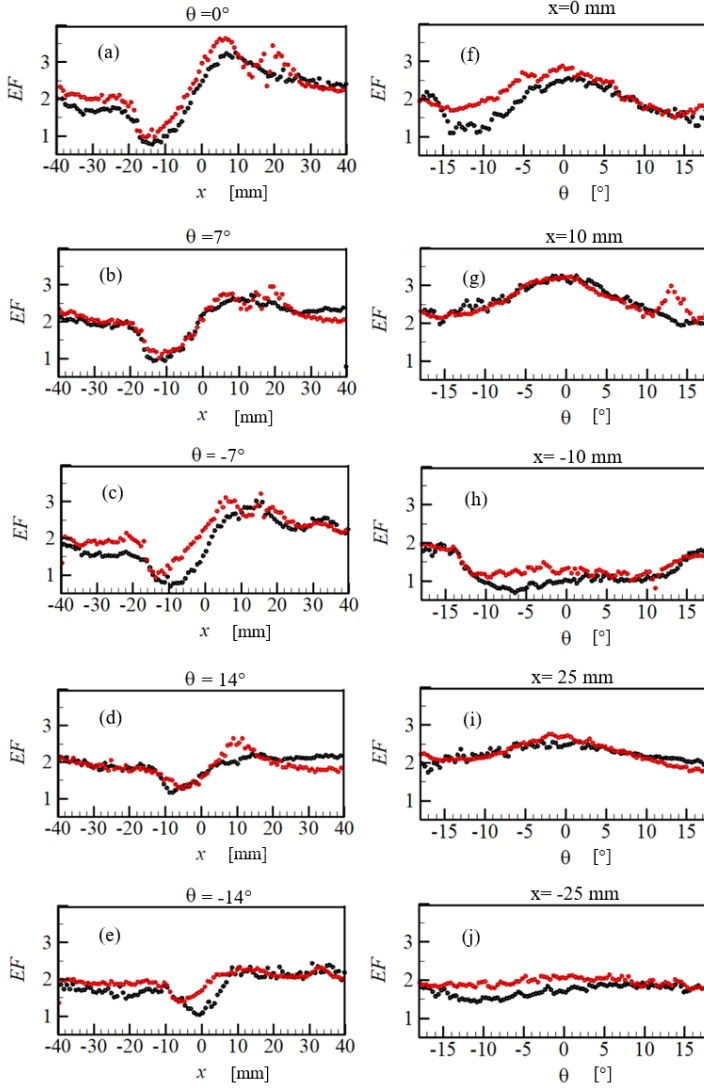


Figure 5.22 – Heat transfer comparison between two circumferentially adjacent dimples at $Re=21600$: red circles, dimple placed at the plane of symmetry; black circles, adjacent dimple; (a), EF at $\Theta=0^\circ$; (b), EF at $\Theta=7^\circ$; (c), EF at $\Theta=-7^\circ$; (d), EF at $\Theta=14^\circ$; (e), EF at $\Theta=-14^\circ$; (f), EF at $x=0$ mm; (g), EF at $x=10$ mm; (h), EF at $x=-10$ mm; (i), EF at $x=25$ mm; (j), EF at $x=-25$ mm

5.4.4 Effects of Re on the local heat transfer

Concerning the plane $\Theta=0^\circ$, the asymmetry in the EF profiles are shown in Figure 5.23(a) and (e), at $Re=20000$. A little deviation is shown for the EF profiles at $\Theta=7^\circ$ and $\Theta=-7^\circ$ upstream of the cavity and in the region having a positive derivative ($\partial EF / \partial x$), downstream of the separation point. At the same locations, a slightly higher difference is shown for the EF profiles placed further from $\Theta=0^\circ$, i.e., $\Theta=14^\circ$ and $\Theta=-14^\circ$. At higher Reynolds numbers, till $Re=80000$, the differences stated before get minimized (Figure 5.23(b, c, d, f, g, h)). Besides, the rise in the Re also influences the maximum value of the EF slightly at $\Theta=0^\circ$.

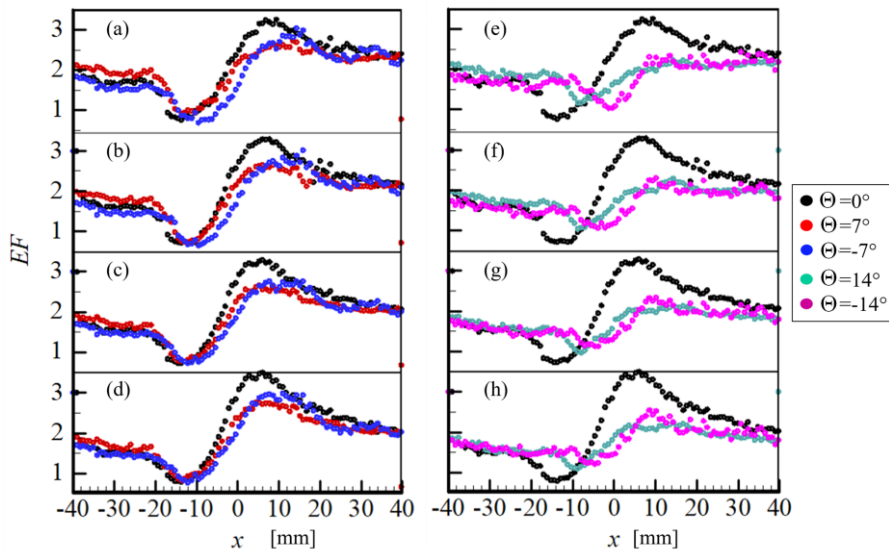


Figure 5.23 – EF profiles in different streamwise directions: black circles, EF at $\Theta=0^\circ$; red circles, EF at $\Theta=7^\circ$; blue circles, EF at $\Theta=-7^\circ$; green circles, EF at $\Theta=14^\circ$; magenta circles, EF at $\Theta=-14^\circ$; (a, e), $Re = 21600$; (b, f), $Re = 43700$; (c, g), $Re = 61300$; (d, h), $Re = 77700$

Differently, in the planes perpendicular to the pipe axis, the influence of the flow regimes on the wall heat transfer has a more significant impact (Figure 5.24). With a rise in the bulk velocity of the flow, the EF values at $x=0$ mm augment towards the heat transfer levels distributed over the wall at $x=10$ mm, in the reattachment zone. Also, the EF profile at $x=-10$ mm moderately undergoes a reduction of the derivative $\partial EF / \partial \theta$ in the region $\Theta=0^\circ-15^\circ$ (which is the edge adjacent to the dimple in the center of the 9th row). Other than that, no significant changes are seen for the EF profiles at $x=\pm 25$ mm at the different Reynolds numbers.

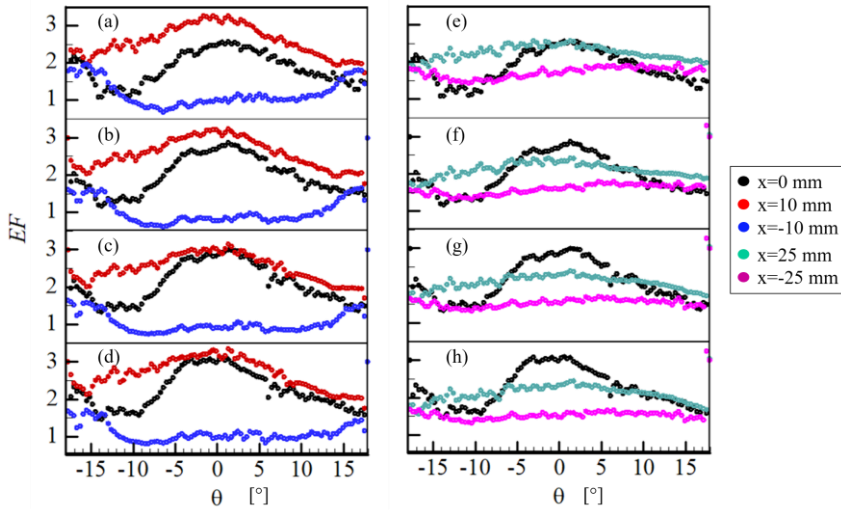


Figure 5.24 - EF profiles in different span-wise directions: black circles, EF at $x=0$ mm; red circles, EF at $x=10$ mm; blue circles, EF at $x=-10$ mm; green circles, EF at $x=25$ mm; magenta circles, EF at $x=-25$ mm; (a, e), $Re=21600$; (b, f), $Re=43700$; (c, g), $Re=61300$; (d, h), $Re=77700$

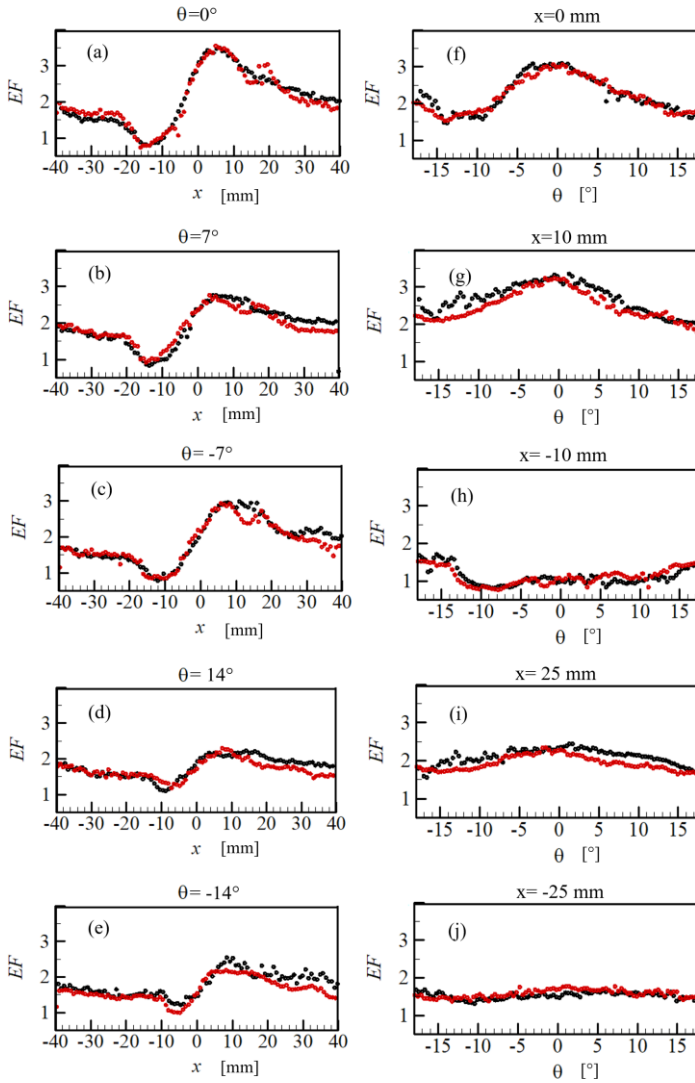


Figure 5.25 - Heat transfer comparison between two circumferentially adjacent dimples at $Re=77700$: red circles, dimple placed at the plane of symmetry; black circles, adjacent dimple; (a), EF at $\Theta=0^\circ$; (b), EF at $\Theta=7^\circ$; (c), EF at $\Theta=-7^\circ$; (d), EF at $\Theta=14^\circ$; (e), EF at $\Theta=-14^\circ$; (f), EF at $x=0$ mm; (g), EF at $x=10$ mm; (h), EF at $x=-10$ mm; (i), EF at $x=25$ mm; (j), EF at $x=-25$ mm

In Figure 5.25, the effect of the flow regime on the EF profiles at different positions inside the ROI's of the two dimples is compared. For the sake of simplicity, the results are shown only at the highest Reynolds number investigated, which is close to $Re=80000$. As opposed to the results at $Re=20000$ (Figure 5.22), and particularly to the two dimple's planes of symmetry, i.e., $\Theta=0^\circ$ (Figure 5.25 (a)) and $x=0$ mm (Figure 5.25 (f)), the EF profiles are quite identical. The differences seen previously at a lower Re (Figure 5.22 (c, e, h, j)), get significantly reduced at $Re=80000$ (Figure 5.25 (c, e, h, j)). At higher flow regimes, the flow structures inside the two dimples have the same behavior influencing the wall heat transfer in the same manner.

5.4.5 Overall aerothermal performance

The EF values, surface-averaged over the Region of Interest of the dimple, are compared, in Figure 5.26 (a), with those of two differently ribbed pipes [41]. The thermal performances are also provided with data concerning the rise in the friction coefficients (Figure 5.26 (b)). Taking into account the range of uncertainties, the Re has practically no influence on the averaged values of EF . Only for the case of the dimpled pipe at $Re=20000$, the EF values are higher than those at the flow regime $Re=40000$ - 80000 . From this comparison, it results that the dimples are providing the same level of thermal efficiency as the one given by continuous helicoidal turbulators. In the heat transfer analysis among the investigated geometries, the helicoidal turbulator, having the discontinuous layout, provides the lowest thermal performance. Nevertheless, the use of such geometry helps in lowering the friction losses given by ribs, as shown in Figure 5.26 (b). In this figure, the skin friction coefficient (C_f) is dimensionalized with the measured value for a smooth pipe ($C_{f,smooth}$). Despite so, the most compelling evidence, from the illustrated comparison, is that the dimpling technique provides C_f augmentations even lower than those given by discontinuous inserts. In detail, the array of dimples provides a rise in the pressure losses which is 0.33 times those in tubular heat exchangers equipped with discontinuous helices.

Another way to compare the improvement in the heat transfer with the rise in skin friction is shown in Figure 5.27. As already mentioned before, the half-dimpled pipes result to be the optimum among the three geometries investigated in this work. For the sake of clarity, it is important to note that its low rise in C_f is mainly given by the fact that the pipe is only half dimpled. In a real scenario, heat exchangers or tubular reactors would be manufactured fully dimpled, all over the circumference of the pipe. For this reason, Figure 5.27 also shows the results for a hypothetical fully dimpled pipe (magenta symbols) at different flow regimes. This consideration is based on the hypothesis of having a skin friction which is twice as much the one provided by the half-dimpled tube. This assumption does not take into account eventual changes in the enhancement factor.

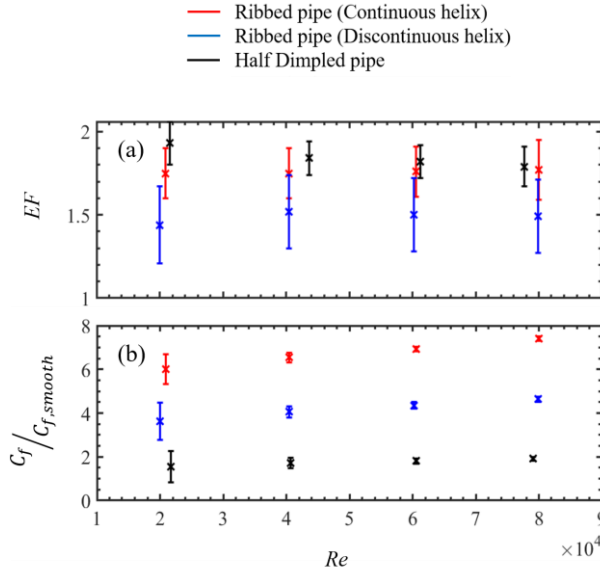


Figure 5.26 – Comparison between the aero-thermal efficiencies of both the dimpling and the ribbing techniques [22, 41]: black crosses, half dimpled tube; red crosses, tube ribbed with a continuous helix; blue crosses, tube ribbed with a discontinuous helix; (a), EF as function of Re ; (b), $C_f/C_{f,smooth}$ as function of Re

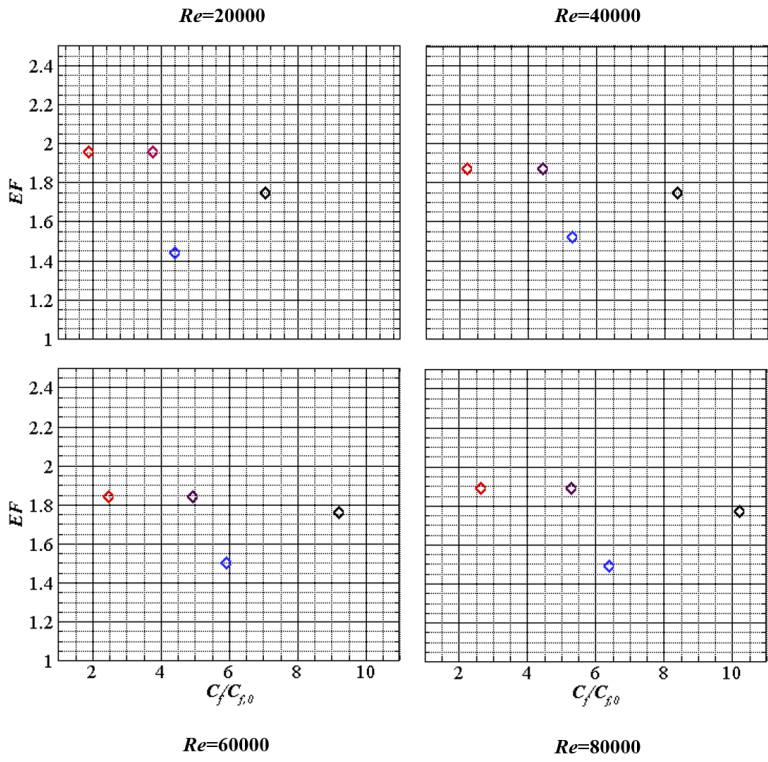


Figure 5.27 –Pareto chart showing the aero-thermal efficiencies of both the dimpling and the ribbing techniques [22, 41]: black diamonds, continuous helix; blue diamonds, discontinuous helix with a continuous helix; red diamonds, half dimpled pipe; magenta diamonds, fully dimpled pipe

5.5 Conclusions

The use of dimples as a heat transfer enhancing technique inside tubular heat exchangers has been investigated experimentally. This study involved the use of both planar and stereo Particle Image Velocimetry to characterize the turbulent flow field

inside the spherical cavities. Distributions of the wall heat transfer at different flow regimes, in the range $Re=20000-80000$, were provided using Liquid Crystals Thermography. The impact of the measurements uncertainty on the final results is provided for every single position inside the cavity under investigation. A novel dataset is made available for validation of numerical simulations of the turbulent flow inside tubular reactors, augmented with the technology proposed. The most important can be summarized as follows:

1. The increase in the skin friction coefficient is a factor twice higher than in a smooth pipe. It is, however, only half of the one seen in tubes with discontinuous helices.
2. The spherical cavity that functions as flow obstacle, generates a massive flow separation/reattachment and augments the vorticity level and promotes secondary flows. This increases the wall heat transfer by convection if compared to smooth pipes.
3. Surface fields of the Enhancement Factor along the wall of two adjacent dimples are measured. In both the cavities, the maximum and minimum values of $EF=3.25$ and 0.80 have been determined, respectively, at the reattachment region and close to the obstacle's leading edge.
4. Surface-averaged EF values of $1.85-1.95$ were measured in the range $Re=20000-80000$. This thermal performance of the dimpling technique is at the same levels as those obtained with the use of helicoidal inserts, meant as ribbing methodologies.
5. The flow regime (Re) also affects the EF distribution, depending on axial and angular positions. The rise in the mass flow rate, influences the spanwise and streamwise EF profiles. This occurs in a way that, at $Re=80000$, the array of dimples shows an azimuthal periodicity, which was missing at $Re=20000$.

It can thus be concluded that the use of clustered dimples in tubular heat exchangers, chemical reactors, or in turbines' internal cooling channels, is demonstrated to be an alternative solution for energy-saving purposes if compared with helicoidal inserts.

5.6 References

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6

Conclusions and perspectives

6.1. Conclusions

The objective of this investigation was to provide detailed experimental results and analysis on the effects of different heat transfer enhancing techniques in tubular heat exchangers. More precisely, in addition to a detailed analysis of the aero-thermal characteristics of the flow field, the goal was to validate the output of LES codes, developed in the Laboratory for Chemical Technology (LCT) of Ghent University (UGent). These computations simulate the fluid dynamics and the wall heat transfer in reactors for steam cracking of heavy feedstocks. Since LES codes present a high degree of complexity in solving separated flows [1], experimental datasets are definitely required. High-fidelity simulations are the basis of optimization routines, used to find the most performant enhancing technique. The shape for tubular reactors has to provide the maximum rise in the heat transfer coefficient with the minimum increase in the friction losses. The effects on the flow field and heat transfer of both a continuous and a discontinuous helicoidal turbulators have been compared. Both the

ribs generate a rise in the skin friction (in the range $Re = 20000\text{--}80000$) and a separated flow with following reattachment in the inter-rib region. Concerning the discontinuous geometry, its rise in the pressure losses is 4 times higher than the case of a smooth pipe, while the continuous geometry provides a rise 7 times higher, at the same flow regimes. Despite this difference, the helicoidal discontinuity lowers the swirling velocity magnitudes, and the wall-bounded TKE distribution without affecting the flow vorticity. This is translated in a lower surface-averaged heat transfer enhancement provided by the discontinuous helix, yielding an enhancement factor equal to 1.50, rather than 1.80 achieved in pipes equipped with continuous helices.

This thesis also investigated the use of dimples in tubular heat exchangers to understand the phenomena developing inside these cavities. With this type of corrugations, the increase in skin friction is only twice as high as the one in smooth pipes at the same turbulent flow regime. The flow separation generates mainly a recirculation bubble and secondary flows, which influence the wall convective heat transfer inside the cavity. This influence provides a surface-averaged enhancement factor equal to 1.90 in the range $Re = 20000\text{--}80000$. Therefore, tubular heat exchangers with half dimpled walls have a thermal performance which is similar to continuous helicoidal inserts without significant rise in the friction losses.

The most compelling evidence is related to the fact that each of the enhancing techniques presented in this work provides an energy-saving. In gas turbines blades, the ribbed cooling channels lead the TIT to increase with a correspondingly rise in the turbomachine efficiency [2]. In a steam cracking reactor, the enhancement of the tubular reactors should lead to a coke deposition at a lower speed and to decreased tube wall temperatures to reach similar severities [3].

The experimental investigation made use of the Liquid Crystals Thermography for the heat transfer measurements and the stereoscopic PIV for flow field velocity measurements. This decision was taken considering the high potential of the two

techniques in providing 2D measurements with a high resolution. To reduce the measurement uncertainties associated with these 2 techniques, two different facilities were designed and mounted in the “Heat and Mass Transfer Laboratory” of the von Karman Institute for Fluid Dynamics. Several issues, concerning the experimental aerothermal investigation, were found in the literature. The difficulties were linked with the low accuracy in the TLCs calibration for 2D Nu measurements (see Ref. [4]) and with the stereo-PIV, particularly, in the reconstruction of the third component (see Ref. [5]) of the velocity vectors. The design of the facilities was driven towards the reduction of the measurement errors through a parametric analysis, based on the equations of the uncertainty propagation. For the stereo-PIV, particular attention was given to maximize the resolution for a spatially resolved turbulent field and to lower the image distortions. In fact, the choice of water as the working fluid, the dimensions of the test section, the use of a surrounding water tank and water prisms, and the calibration system are in line with what is stated before. Similarly, the design of the low-speed wind tunnel, employed for the heat transfer measurements, aimed at minimizing the significant uncertainties given by the use of TLCs. With this purpose, up-to-date and new algorithms were used for an in-situ calibration of the TLCs, in line with the work of Kodzwa et al. [6]. This was achieved by mainly calibrating the crystals scattering with the signal from an IR camera and by the use of a structured multi-calibration algorithm [4]. Also the choice of the two different heating and illuminating systems proposed in this work proved to be significant. Both Inconel and transparent conductor foils provide uniform heat flux as thermal boundary conditions. The use of one heating system rather than the other depends on the type of enhancing technique adopted for the tubular heat exchanger. Besides, the LED system for the illumination of the test section allowed the optical access inside the ROI, filtering out the light reflections generally generated by other types of illumination systems.

Large Eddy Simulations of the aerothermal effects on tubular heat exchangers of the two helicoidal turbulators (presented in this work) are shown and discussed in the doctoral thesis of Dedeyne [7]. For a significant comparison, the experimental datasets

presented in this PhD thesis are provided with a detailed quantitative estimation of the measurement uncertainty. Statistical tools (e.g., the bootstrap method) were used for the quantification of the uncertainty when measuring the turbulent fluctuations and the Nusselt number distributions.

6.2. Perspectives

The LCT of UGent is coming up with new geometries aiming at optimizing the performance of tubular steam cracking reactors. The optimization routines are generating new geometries with dimples and grooves presenting more complex shapes; the pipes are deformed spirally in different manners [8, 9, 10]. According to the simulations, those geometries are considered to provide energetic performances higher than those obtained from the geometries presented in this thesis. This is particularly achieved by manipulating the way the obstacles generate the recirculation zones, the fluctuating components of the velocity vectors, and the intensity of the swirling flow structures. As seen in the experimental results described in the core of this work, the 3D separation and the flow reattachment have a significant influence on the local wall heat transfer. PIV measurements for highly corrugated pipes are affected by images with strong optical deformations. In such a way, the calibration of the cameras with dotted grids are characterized by images corrections with substantial errors. The calibration obtained with a polynomial function, as in Soloff [11], provides a corrected grid which has a significant deviation from the real dotted grid. Standard deviations, much higher than 2 pixels, are found in PIV calibrations through systems with limited optical access. A way to solve this issue is related to the use of fluids with the same refractive index of the test sections materials. In this way, the light rays of the particle images reach the camera's sensor without any refractions, keeping the optical deformations almost insignificant [12]. As a matter of fact, the fluids utilized for the index-matching are expensive and, since they have a high viscosity, substantial modifications to the PIV facility need to be considered. A cheaper solution is the use

of ray-tracing techniques, which are physical-based rendering algorithms. This means that this type of software uses the physical equations of the light propagation through media, to follow and trace the path of each light ray scattered by an image. In this fashion, a ray-tracing based PIV approach is able to match the position of each point of a real object with each pixel position of the camera sensors. Therefore, the calibration process gets rid of the 3D polynomial function, providing image reconstructions with a standard deviation lower than 1 pixel [13]. This is considered to be helpful in stereo-PIV calibrations through highly distorted images, where the reconstruction of the third velocity components is significantly influenced by the measurement errors. Martins et al. [13] validated the use of ray-tracing for improvements in PIV measurements by applying this rendering algorithm for a jet flow seen through one, two and three glass spheres placed in front of the photographic equipment. Therefore, ray-tracing can be considered as a high potential solution for PIV-based velocimetry in complex geometries.

The uncertainty level on the 2D measurements of the Nusselt number through the use of TLCs can be improved. The two TLCs' calibration techniques presented in this thesis yield extended uncertainties in the range 15-20 % and 10-15 %, respectively, for the line-by-line and pixel-by-pixel methods. This is mainly caused by the significant derivatives $\partial T / \partial Hue$ in the green region of the TLCs calibration curves. This issue can be fixed by either mixing wide-band TLCs paints with different red-starts and color plays or applying several layers of narrow-band TLCs, stack on top of each other, along the surface. According to the author, the first methodology is supposed to increase the variation of the Hue (∂Hue) for small temperature variations within the green scattering zone, playing with different red-starts of the TLCs paints. The second methodology is known to reduce the uncertainty on the temperature measurements, but is expensive, and gives a higher complexity in both the paints application and the calibration process. Another way to improve the measurements can be the use of SLR cameras, whose sensors have a higher resolution in the

digitalization of the signals. This will help in reducing significantly the measurement uncertainties mentioned above.

Advanced measurement technologies should focus on the experimental characterization of reacting flows, to validate codes simulating the effects of enhanced reactors taking into account the chemistry. With Planar Laser-Induced Fluorescence (PLIF) or with Laser-Induced Phosphorescence (LIP), the measurement of the intermediate species concentrations (ethyl-radical, propyl-radical, etc.) and of the reacting flow temperature profiles inside the turbulent fields can be provided. In this way, it is possible to investigate the degree of uniformity of the temperature profiles, which are influencing the kinetics of the chemical reactions. Moreover, advanced numerical codes, which simulate the coke formation kinetics with a higher level of complexity, have found difficulties in being validated. This is given by the fact that, in the literature, there is a lack of this type of experimental investigations without flow assumptions. For this reason, future investments must be foreseen for the measurement of the local rate of coke formation in differently enhanced tubular steam cracking reactors.

6.3. References

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Appendix A

Resolution in turbulence

Van Doorne et al. [1] used the stereo-Particle Image Velocimetry to measure the laminar, transitional, and turbulent state in pipe flows. Particularly for the case of a turbulent flow, the authors chose the diameter of the tube ($D = 40\text{mm}$), the stereo PIV setup, and the interrogation windows size ($32 \times 32 \text{ pix}^2$) in order to have a spatial resolution of 15 Kolmogorov scales. Their experimental results in terms of mean flow and turbulence statistics were validated by means of DNS data [2].

Following the Kolmogorov hypothesis [3], the smallest turbulent scale is proportional to a characteristic length with the inverse of the Reynolds number:

$$\frac{\eta}{l_0} \sim Re^{-3/4}$$

In ribbed or dimpled pipe, the characteristic length is not the pipe radius, as in Van Doorne et al. [1], but it is the obstacle's height or indentation's depth. Considering that those lengths are smaller than the pipe diameter, in channels with a low block-

ratio, the resolution (in terms of Kolmogorov scales) of the measurements has to be increased.

One possible solution is to use smaller interrogation windows, and thinner laser planes. Despite so, for stereo-PIV, the thickness of the laser cannot be decreased to values lower than 1 mm, to have a good calibration in the direction perpendicular to the laser plane. Hence, to increase the spatial resolution of the system, the pipe diameter of the test section was chosen 4 times higher than the one in Van Doorne et al. [1]. In such a way, also the obstacle's height increases with the diameter, proportionally to the blockage ratio of the turbulator. In any case, the scaling factor should have been chosen equal to 14, in order to reproduce the same quality as in the stereo-PIV campaign made by the authors [1]. Nevertheless, the choice of a diameter higher than 150mm would have shortened the length of the piping line, due to the limited space of the laboratory. In light of what stated before, the author of the dissertation considers the choice of having a pipe diameter of 150mm a good compromise between spatial resolution and technical limitations.

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Appendix B

Correlation maps and Reconstruction errors

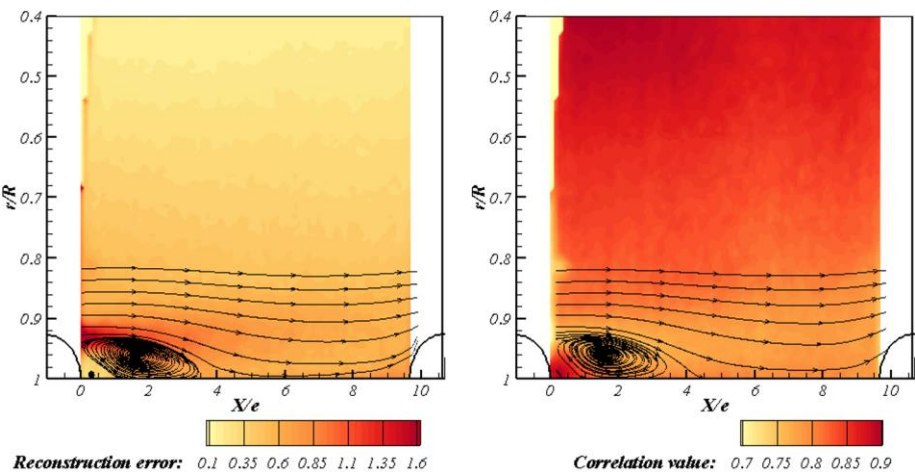


Figure B.1 – Quality of the stereo-PIV cross-correlation algorithm for RIB-1 at $Re=20000$: left, Reconstruction error field; right, Correlation map

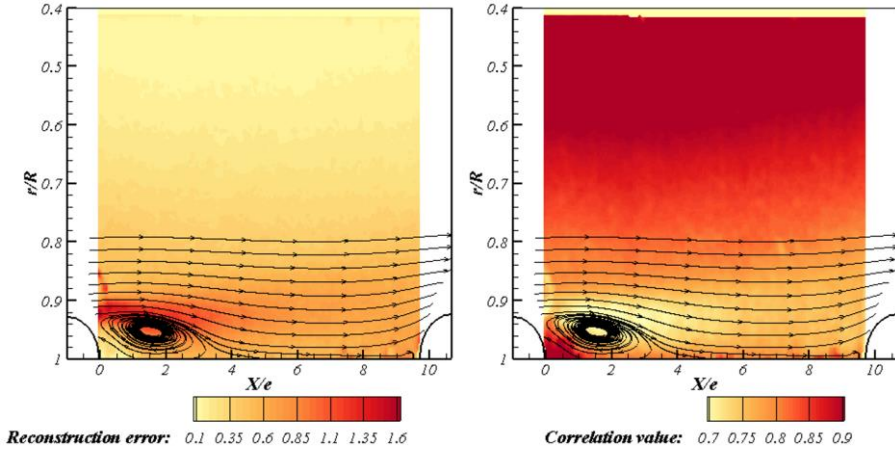


Figure B.2 – Quality of the stereo-PIV cross-correlation algorithm for RIB-2 at $Re=20000$: left, Reconstruction error field; right, Correlation map

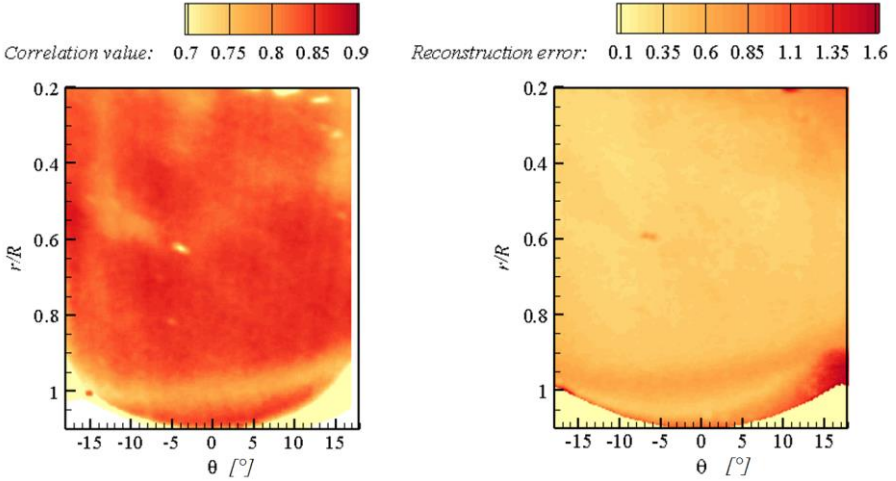


Figure B.3 – Quality of the stereo-PIV cross-correlation algorithm for the half-dimpled pipe at $Re=20000$: left, Reconstruction error field; right, Correlation map

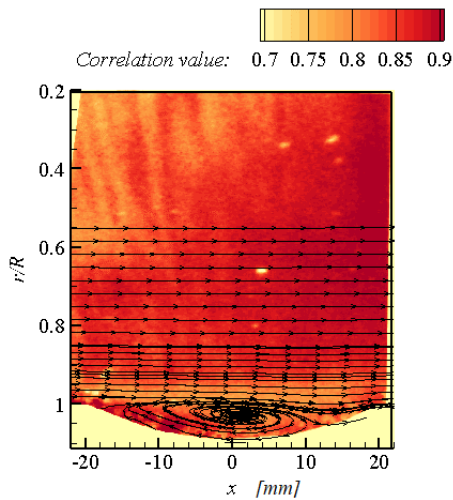


Figure B.4 – Quality of the PIV cross-correlation algorithm for the half-dimpled pipe at $Re=20000$: Correlation map

Appendix C

ΔP measurements

The difference between static pressure amid inlet and outlet of the ribbed pipe provides data for the evaluation of C_f . As a matter of fact, in the absence of pipe curvatures, the static pressure equals the total pressure at the wall position (since the flow velocity is null), for the case of incompressible flows. The loss of the total pressure is linked to the skin friction coefficient through the computation of a surface-averaged value of τ_{wall} .

The experimental campaign expects the use of 2 pressure taps. The pressure tap used to measure the static pressure downstream of the test section was placed 7D far from the outlet section. This distance was chosen to avoid local effects of the swirling recirculating bubbles on the measured static pressure. Nevertheless, the assumption of a recovered swirl-less boundary layer at a distance of 7D was assumed. Therefore, this assumption expects an error in the measurements. With the hypothesis of having the flow still swirling at 7D far from the test section's outlet, the dynamic pressure would be 0.2 Pa and 2 Pa, at respectively $Re=20000$ and $Re=80000$. The computation considers the tangential flow with its maximum intensity (25% the bulk velocity). Those errors correspond to 8% and 6% of the

corresponding static pressure differences across the ribbed test section (case with RIB-1). The evaluation of the static pressure jumps (Figure C.1) across the ribbed test section follows the equation:

$$\Delta p_{static} = \Delta P_{RIB}^* + \Delta P_{smooth}^*$$

with

$$\Delta P_{RIB}^* = \Delta P_{meas} - \Delta P_{smooth}$$

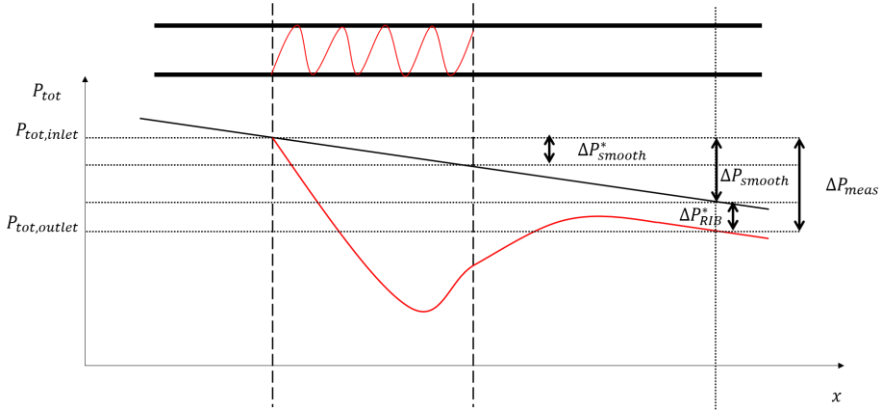


Figure C.1 – Sketch representing the total pressure jumps inside the pipe due to the presence of a helicoidal turbulator

Appendix D

$\langle uv \rangle$ uncertainty

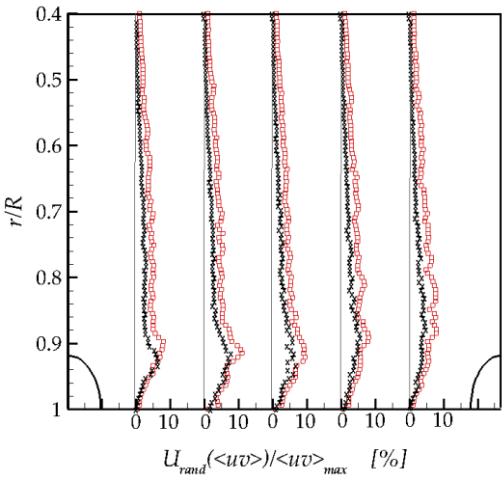


Figure D.1 - Normalized profiles of the random uncertainty in the measurements of the shear stress $\langle uv \rangle$ at five different axial positions between the 7th and the 8th obstacles: red squares, RIB-1; black crosses, RIB-2

Figure D.1 illustrates the bootstrap results made to evaluate the relative random uncertainties in measuring the shear stress $\langle uv \rangle$. For clarity, this physical quantity

represents the covariance between the axial and the radial fluctuations of the turbulent field. Higher radial gradients of the axial velocity, generated by the wall obstacles, let the shear stresses increase. In comparison with RIB-2, the uncertainty in measuring such turbulent quantities is shown to be higher for the case of RIB-1. For the latter, peak values of 10% are found in the wake zone behind the 7th obstacle.

Appendix E

Fully developed flow

Velocity profile measurements at 15D were already made by the authors Mayo et al. [1] at the same flow regime. The velocity profile of the mean flow has been compared with that one given by the DNS results in the following journal article: “Direct numerical simulation of a 30R long turbulent pipe flow at $R^+ = 685$: large- and very large-scale motions” of X. Wu, J. R. Baltzer and R. J. Adrian (2012) in Journal of Fluid Mechanics. The measurements of Mayo et al. are at $Re=24000$ while the DNS results of Wu et al. are at $Re=24580$. The comparison is shown in the following Figure:

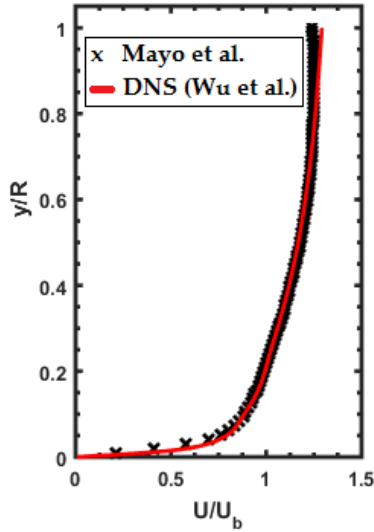


Figure E.1 - Normalized radial profile of the mean axial velocity: black crosses, s-PIV measurements at 15D; red solid line, DNS results

References

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Appendix F

IR camera's calibration

The TLCs employed for the heat transfer measurements, in both the ribbed and dimpled test sections, are calibrated with the use of an infrared camera. The latter was calibrated using the smooth test section as a target, covered with the same multi-layers described in Chapters 4 and 5. The Inconel foil heats the inner surface temperature, where a layer of thermochromic liquid crystals is painted on. A further coat of black paint is applied to have heat reflections as low as possible. The Intensity Units (IU) reordered by the IR camera are calibrated with the temperature values of 3 different surface thermocouples, glued at three angular positions (see Figure F.1). At each angular position, the spatial average of the IU is matched with the time-averaged value of the local thermocouple. The results of the three different calibration curves are shown in Figure F.2. The angular effects on the calibration are practically significant only for temperature higher than 46°C. All those deviations are taken into account in the final IR camera uncertainty. Therefore, the latter turns to be a function of the temperature level of the measurement. The same procedure was used to calibrate the IR camera adopted to measure the temperature of the external surface of the test section, made in Plexiglas (and not black matt).

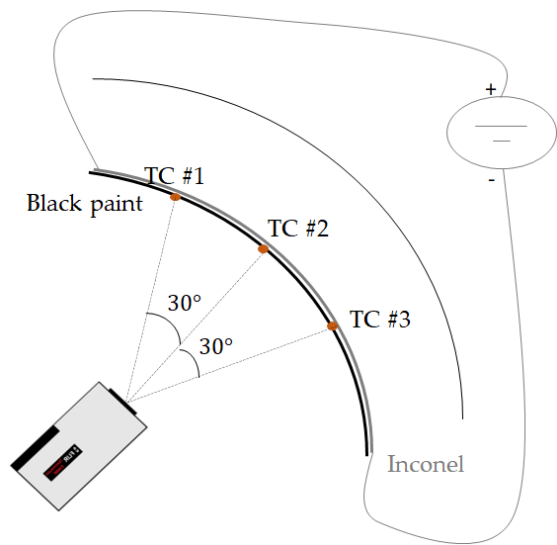


Figure F.1 – Set-up for the IR camera calibration at several angular positions

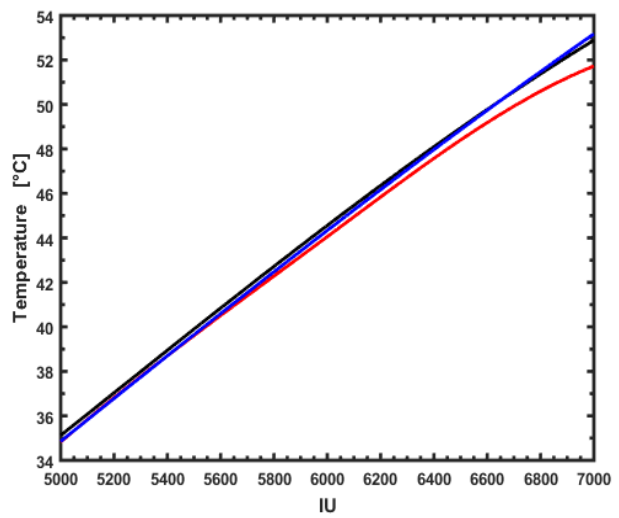


Figure F.2 – Calibration curves of the IR camera at 3 angular positions: black solid line, calibration at TC#1; blue solid line, calibration at TC#2; red solid line, calibration at TC#3

Appendix G

Conduction losses in RIB-2

As shown in Chapter 4, the conduction losses in RIB-2 have a significant impact on the EF field, particularly for $Re=20000$. Figure G.1 illustrates clearly the remarkable importance of the conductive heat flux compared to convection at $Re=20000$, rather than higher flow regimes. This occurs mainly at the leading edge of the obstacles and in the rib-free zone, where the conduction represents 50% of the heat flux. The latter (q_{tot}) is computed as the sum of the convective (q_{conv}) and the conductive (q_{cond}) heat fluxes. Therefore, at low bulk velocities, the influence of the flow structures on the wall heat transfer is reduced by significant conduction in the radial, tangential, and axial directions. The latter ones are shown as a percentage of the total conductive heat flux in Figure G.2 for RIB-2, at $Re=20000$. Notably, the lateral heat fluxes are due to the existence of both tangential and axial gradients of temperature over the inter-rib zone.

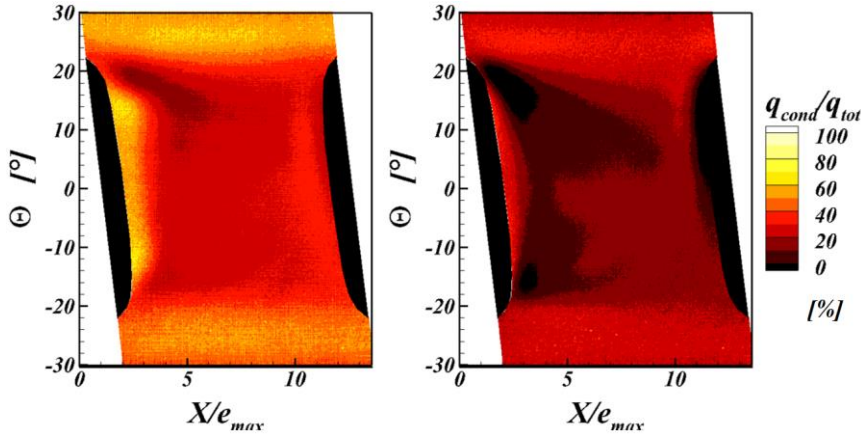


Figure G.1 – Percentage of heat lost by conduction over the inter-rib surface for the case of RIB-2: left, $Re=21160$; right, $Re=78850$

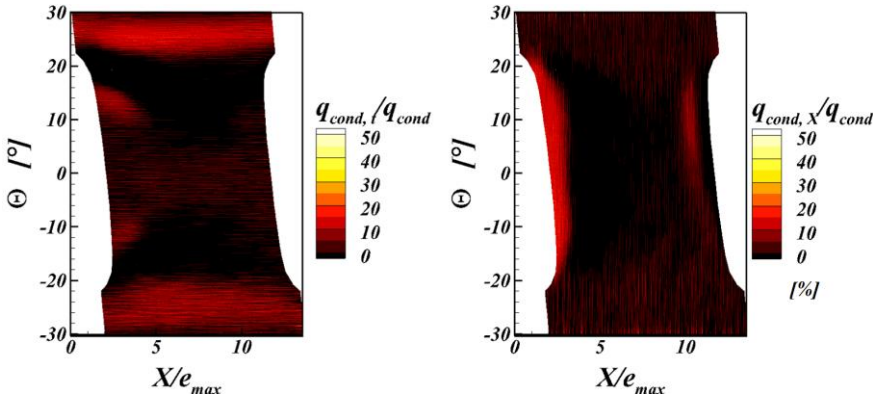


Figure G.2 – Percentage of the lateral heat flux by conduction for the case of RIB-2 at $Re=20000$: left, tangential heat flux; right, axial heat flux

Appendix H

Wall conduction in dimples

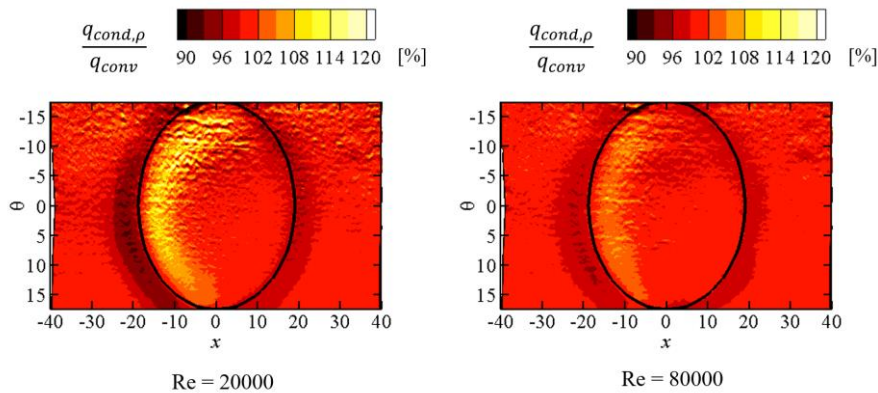


Figure H.1 – Ratio between radial conductive and convective heat flux: left, $Re=20000$; right, $Re=80000$

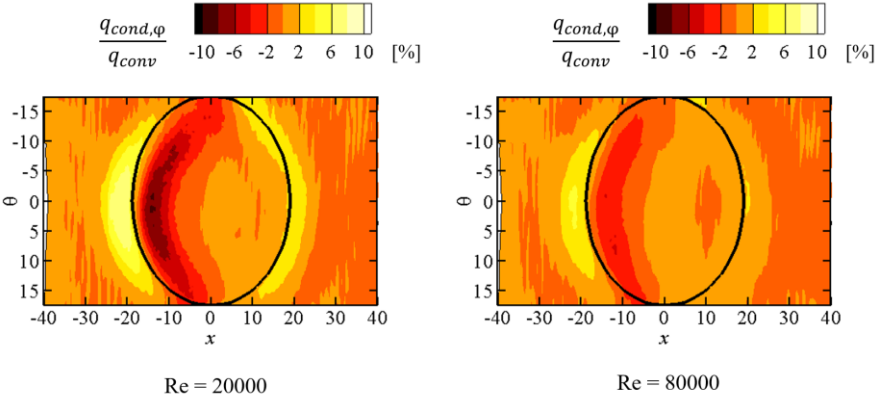


Figure H.2 – Ratio between azimuthal conductive and convective heat flux: left, $Re=20000$; right, $Re=80000$

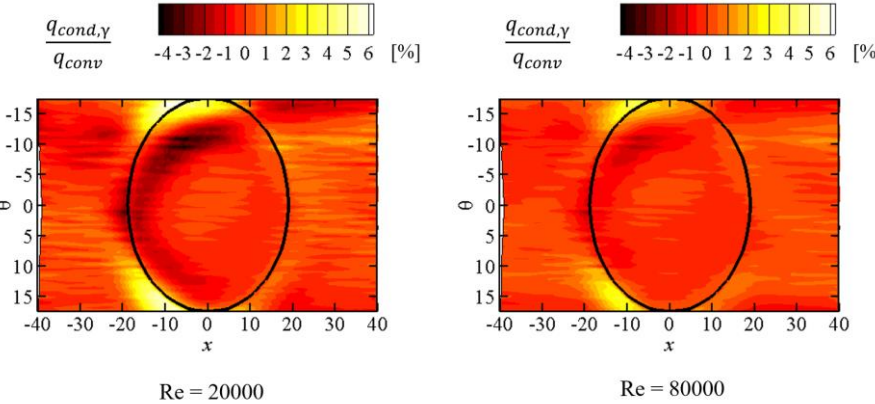


Figure H.3 – Ratio between zenithal conductive and convective heat flux: left, $Re=20000$; right, $Re=80000$

Appendix I

TKE inside dimples

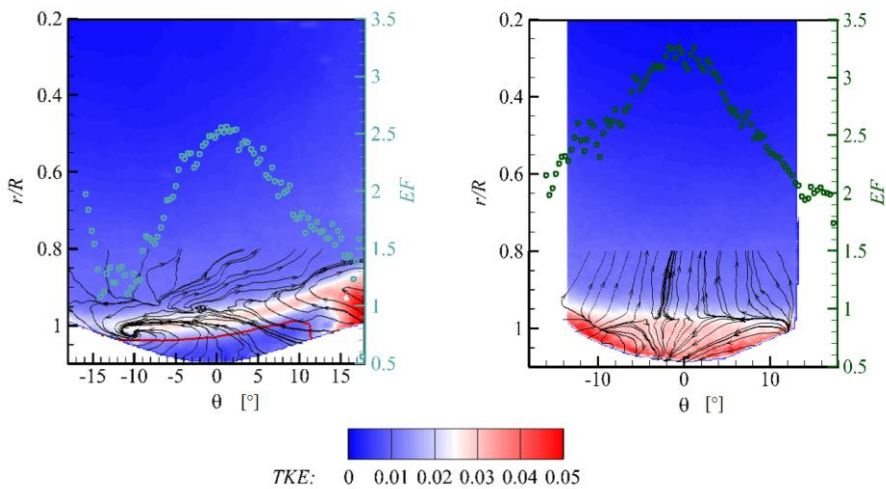


Figure I.1 – Turbulent Kinetic Energy fields inside the dimple: solid red line, zero-velocity line; solid black line, streamlines; green squares, local EF ; left, TKE for plane C at $Re=27900$; right, TKE for plane D at $Re=30900$

