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Heat transfer in production and decay regions of grid-generated turbulence

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ABSTRACT

Heat transfer measurements around the centreline circumference of a cylinder in crossflow are performed in a wind tunnel. The cylinder is placed at several stations downstream of three turbulence-generating grids with different geometries and different blockage ratios σ_g: a regular grid (RG60) with $\sigma_g = 32\%$, a fractal-square grid (FSG17) with $\sigma_g = 25\%$ and a single-square grid (SSG) with $\sigma_g = 20\%$. Measurements are performed at 20 stations for 3 nominal Reynolds numbers (based on the diameter D of the cylinder) $Re_\infty = 11,100,\ 24,500,\ 37,900$. Hot-wire measurements are performed along the centreline, without the cylinder in place, to characterise the flow downstream of the grids. The extent of the turbulence production region, where the turbulence intensity Tu increases with the streamwise distance x from the grid, is higher for SSG and more so for FSG17 than for RG60. The angular profiles of the Nusselt number Nu are measured in the production regions of these two grids and are compared to those obtained in the decay regions, where Tu decreases with x. This comparison is made at locations with approximately same Tu. It is found that, for SSG, $Nu/Re^{0.5}$ on the front of the cylinder (boundary layer region) is lower in the production region than in the decay region. This is explained by the presence of clear and intense vortex shedding in the production region of SSG which reduces the turbulent fluctuations which are "effective" in enhancing the heat transfer across a laminar boundary layer. For higher Re_∞, the values of $Nu/Re^{0.5}$ on the front of the cylinder are higher in the production region of FSG17 than in that of SSG, despite Tu being higher for SSG. This is consistent with a lower intermittency of the flow for FSG17 caused by the presence of the fractal geometrical iterations. The recovery of Nu on the back of the cylinder (wake region) is appreciably higher in the production region than in the decay region for both FSG17 and for SSG. This can be due to the lower integral length scale ratio L_u/D in the production region and suggests, for the same Re_∞, a reduction of the vortex formation length downstream of the cylinder, possibly promoted by the interaction between the wakes of the bars of the grid and the wake of the cylinder. At a large distance from the grids, the heat transfer enhancement is higher and it is more efficient for FSG17 and for SSG than for RG60. For high values of x in the turbulence decay region of the grids, the values of $\overline{Nu}$ (circumferential average of Nu) are similar for FSG17 and for SSG and they are both appreciably higher than for RG60. This happens despite both FSG17 and SSG having a lower blockage ratio than RG60. The use of FSG17 has the practical advantage of combining high heat transfer rates on the cylinder with a weak vortex shedding from the grid.

1. Introduction

The effects of turbulent flows on the heat transfer from cylinders are of interest in different engineering applications. Several industrial devices use cylindrical geometries to exchange heat between a wall and a fluid in a turbulent regime. Examples of such devices are shell-and-tube heat exchangers, pressurised water reactors or water to air radiators [22] to name a few. Understanding how to increase the heat transfer by tuning the turbulence properties of the flow is highly desirable.

Different investigations have focused on the effects of some turbulent flow parameters on the heat transfer values for a cylinder in crossflow. Smith and Kuethe [44] developed a theoretical model by assuming that in the proximity of the front stagnation point the Reynolds stresses are proportional to the turbulence intensity Tu in the free-stream (flow approaching the cylinder) and to the distance from the wall. Their model, supported by experimental

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Nomenclature

a_1	strain rate parameter
C_ϵ	dissipation coefficient, $C_\epsilon = \epsilon L_u/u'^3$
D	diameter of the cylinder
E_{sh}	energy of the vortex shedding from the largest bars of the grids
E_u	power spectral density of u
f	frequency
f_λ	"Effective" frequency
f_{sh}	frequency of the vortex shedding from the largest bars of the grids
h	heat transfer coefficient, $h = \frac{q_{conv}}{T_w - T_\infty}$
H	height (width) of the wind tunnel's working section (length of the cylinder)
I	electric current
L_0	distance between the largest bars of the grid
L_e	dissipation length scale, $L_e = 1.5u'^3/\epsilon$
L_h	length of the heated section of the cylinder
L_u	integral length scale of u, $L_u = U\Theta_u$
Nu	local Nusselt number, $Nu = \frac{hD}{\lambda_{film}}$
$\overline{Nu}$	circumferentially averaged Nusselt number, $\overline{Nu} = \frac{1}{180}\int_0^{180} Nu(\theta)\mathrm{d}\theta$
q_{cond}	conductive heat flux
q_{conv}	convective heat flux
q_{gen}	input heat flux, $q_{gen} = V_h I/(\pi D L_h)$
q_{rad}	radiative heat flux, $q_{rad} = \epsilon_h \sigma\left(T_w^4 - T_\infty^4\right)$
Re	local Reynolds number, $Re = \frac{UD}{\nu_{film}}$
Re_∞	inlet Reynolds number, $Re_\infty = \frac{U_\infty D}{\nu_\infty}$
t_0	thickness of the largest bars of the grid in a plane parallel to the grid
t_h	thickness of the heating foil
T	temperature
T_{film}	film temperature, $T_{film} = (T_w + T_\infty)/2$
Tp	turbulence parameter, $Tp = TuRe^{1/3}(L_u/D)^{-1/3}$
Tu	turbulence intensity, $Tu = u'/U$
Tu_{eff}	"Effective" turbulence intensity, $Tu_{eff} = u'_{eff}/U$
Tu_{peak}	maximum value of turbulence intensity on the centreline
u	streamwise fluctuating velocity component
u'	RMS value of u
u'^2_{eff}	"Effective" turbulent kinetic energy
U	local mean streamwise velocity
U_∞	inlet velocity
V_h	voltage drop across the length of the heating foil
x	streamwise distance from the grid
x^*	wake-interaction length scale, $x^* = L_0^2/t_0$
x_{peak}	centreline streamwise location of the maximum of turbulence intensity
y	spanwise direction (origin of the reference system on the centre of the grid)

Greek symbols

Δf	width of the frequency integration interval for the computation of E_{sh} and of u'^2_{eff}
δ	boundary layer thickness
λ	thermal conductivity of air
λ_h	thermal conductivity of the heating foil
ν	kinematic viscosity of air
σ	Stefan-Boltzmann constant
σ_c	blockage ratio of the cylinder
σ_g	blockage ratio of the grid
θ	angular position measured from the front stagnation point
Θ_u	integral time scale of u
ϵ	turbulent kinetic energy dissipation rate per unit mass
ϵ_h	surface emissivity of the heating foil

Subscripts

∞	inlet values (upstream of the grids)
film	evaluated at T_{film}
FSP	front stagnation point, $\theta = 0^\circ$
RSP	rear stagnation point, $\theta = 180^\circ$
w	wall values

results, showed that the Frossling number at the front stagnation point $Nu_{FSP}/Re^{0.5}$, which is invariant with Re for laminar free-stream conditions [15], was directly proportional to the turbulence parameter $TuRe^{0.5}$, where Re is the Reynolds number based on the diameter D of the cylinder and on the mean streamwise velocity U. Kestin and Wood [25] and Lowery and Vachon [31] measured respectively the mass transfer and the heat transfer from a cylinder in a turbulent crossflow generated by grids in a wind tunnel. Both studies correlated the values of $Nu_{FSP}/Re^{0.5}$ with a second-degree polynomial function of $TuRe^{0.5}$. The properties of the turbulent flow approaching the cylinder affect the entire angular heat or mass transfer profile [5,41] and so the values of the angle-averaged Nusselt number $\overline{Nu}$. Similarly to Nu_{FSP}, $\overline{Nu}$ was correlated with empirical fits as a function of both Re and Tu (e.g. [13,43,31,33,26]). The main conclusion from these investigations was that both Nu_{FSP} and $\overline{Nu}$ are increased by larger values of Tu and that the enhancement is more evident at higher Re.

Several experiments have shown that the role of the integral length scale L_u of the incoming flow can also be important [40]. Van Der Hegge Zijnen [47] reported that for the same Re and Tu, $\overline{Nu}$ increased with the ratio L_u/D for $0 < L_u/D < 1.6$ whereas it decreased for $L_u/D > 1.6$. Zukauskas et al. [56] also found the presence of an optimal value of L_u/D for which $\overline{Nu}$ is maximum, but in this case it occurred for $L_u/D = 10/Re^{0.5}$ in the range $10^4 < Re < 10^6$ and for $Tu = 0.5\%$. To predict the effect of L_u on the front stagnation point heat transfer, different relations have been developed in the form $Nu_{FSP}/Re^{0.5} \propto Tu^\alpha Re^\beta (L_u/D)^\gamma$ with α, $\beta > 0$ and $\gamma < 0$ (see e.g.[3,49,2,41,16]), thus showing that Nu_{FSP} is usually anti-correlated with L_u/D. On the contrary, the experimental results on the effect of L_u/D on the Nusselt number at the rear stagnation point, Nu_{RSP}, appear somewhat contradictory. Torii and Yang [45] found that higher values of L_u/D caused a noticeable decrease of Nu_{RSP} whilst Yardi and Sukhatme [51] and Sanitjai and Goldstein [41] did not find an appreciable effect of the free-stream turbulence on the values of Nu_{RSP}.

The turbulent flow approaching the cylinder has been usually generated with perturbing grids placed upstream of the heat transfer model in a wind tunnel. Examples of these turbulence generators are regular square mesh grids, arrays of parallel wires, damping screens or perforated plates. Quintino [39] measured $\overline{Nu}$ for an electrically heated cylinder (made in copper) in crossflow which was mounted horizontally in a wind tunnel and placed at different distances from grids made of two vertical strips; $\overline{Nu}$ was obtained from the electric power delivered to the cylinder and from the wall temperature which was uniform given the heating technique used in that experiment. Global heat transfer mea-