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Mixed convection around a tilted cuboid with an isothermal sidewall at moderate Reynolds numbers

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A B S T R A C T

Mixed convection around rectangular structures has great scientific value with various industrial applications. In this study, heat transfer in the mixed convection regime around a tilted rectangular cuboid is investigated experimentally and numerically. The cuboid has adiabatic wall boundaries, except for its frontal wall which is set isothermal and facing downwards when tilted. The effect of an upcoming horizontal flow on mixed convection around the cuboid was investigated focusing on flow characteristics for Reynolds numbers in the range of $10^4 < Re < 1.8 \times 10^5$ with low turbulence intensities of $TI < 5.5\%$ and Richardson numbers of $5 \times 10^{-3} < Ri < 2$, well within the mixed convection regime. A numerical simulation based on the three-dimensional SST $k - \omega$ turbulence model, validated against data from wind tunnel experiments, was used to accurately estimate the total heat transfer rate and the average Nusselt number as a function of cuboid inclination and wind direction at different *Ri* and *TI*. The results show that convective heat transfer is enhanced at a characteristic inclination of $\sim 80^\circ$ by $\sim 30\%$ in comparison to the case where the freestream flow is perpendicular to the isothermal wall. Flow behaviour in the vicinity of the cuboid demonstrated that laminar separation and turbulent reattachment were responsible for this enhancement, which is a behaviour not observed for isothermal flat plates but reported for flows around cylinders and spheres.

1. Introduction

An overwhelming amount of literature, since the comprehensive correlations review by Whitaker [1], has been published on convective heat transfer around heated objects under an incident fluid flow. In spite of a rectangular parallelepiped (hereafter called 'cuboid') being a fundamental geometry of importance in different fields, research on external flows has mainly focused on flat plates [2,3], cylinders [4,5], and spheres [6,7]. Investigations on external mixed convection around cuboids have been done, but only for a low Reynolds number regime [8,9] or wall-mounted cubes [10,11]. Interestingly, there are numerous fundamental studies on natural convection for rectangular cavities, i.e. enclosed buoyant flow [12–14].

Important applications of external mixed convection around cuboids include, but are not limited to, design of solar receivers for concentrating solar power [15], solar collectors [16], and characterisation of building envelope performance [17]. The motivation for the present study specifically arises from the study of receivers in concentrating solar thermal power applications [18,19]. In these systems, a large bank of tubes (up to 30 m tall) is mounted atop a tower (up to 250 m) and irradiated with concentrated sunlight from a heliostat field (up to 1.6 km distant from the tower). The receiver surface heats up to temperatures larger than 500 °C and is exposed to a broad range of natural and forced (wind) convection regimes during its required daylight hours of operation. An example of commercial application of flat tubular receivers with the geometry similar to that considered in this study is the design being developed by Vast Solar Pty Ltd [20,15].

Obtaining experimental data for the convective heat loss from these structures is challenging due to simultaneous solar irradiation, radiative losses, and internal heat transfer to a working fluid. Experimental set-ups are smaller scale and internally heated rather than irradiated, and often operated under natural convection only [21,22]. Cryogenic wind tunnels have been used to obtain forced convective heat loss with suppressed radiative phenomena [23,24]. Extraction of data from on-sun tests is very challenging, not least because of the variability of the solar irradiance [25]. In addition, a range of interesting concepts that could substantially reduce convective losses from solar receivers continue to motivate

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Nomenclature

A	surface area of isothermal boundary, m^2
$\mathbf{g}$	acceleration of gravity, m s^{-2}
g	magnitude of acceleration of gravity, m s^{-2}
Gr	Grashof number, –
h	enthalpy per unit mass, J kg^{-1}
h_{conv}	heat transfer coefficient, W m^{-2} K^{-1}
H	plate height, mm
k	turbulence kinetic energy, m^2 s^{-2}
K	kinematic energy per unit mass, J kg^{-1}
L	characteristic length, m
N	total number of cells, –
Nu	average Nusselt number, –
p	pressure, Pa
P_{heat}	heating power dissipated from heating elements, W
q	heat flux, W m^{-2}
$\dot{Q}_{cond}$	heat transfer rate (or heat loss) by conduction, W
$\dot{Q}_{conv}$	heat transfer rate (or heat loss) by convection, W
$\dot{Q}_{rad}$	heat transfer rate (or heat loss) by radiation, W
$\dot{Q}_{X,cond}$	heat conduction through insulation cover, W
R	specific gas constant, J kg^{-1} K^{-1}
Ra	Rayleigh number, –
Re	Reynolds number, –
Ri	Richardson number, –
t	time, s
T_w	wall temperature, K
T_∞	ambient temperature, K
TI	turbulence intensity, %
$\mathbf{u}$	velocity vector, m s^{-1}
U	Freestream wind speed, m s^{-1}
W	plate thickness, mm
W_X	thickness of insulation cover, mm
x, y, z	Cartesian coordinates, m
Y	coordinate from leading edge, m

Greek symbols

α_{eff}	effective thermal diffusivity, kg m^{-1} s^{-1}
β	thermal expansion coefficient, K^{-1}
δ	mesh size normal to hot boundary surface, μm
ΔT	temperature difference, K
Δ_7	absolute difference with finest mesh, %
θ	pitch angle, degrees
λ	thermal conductivity, W m^{-1} K^{-1}
μ_{eff}	effective viscosity, m^2 s^{-1}
ν	kinematic viscosity, m^2 s^{-1}
ρ	density, kg m^{-3}
σ	thermal emissivity, –
ϕ	yaw angle, degrees
ω	turbulence specific rate of dissipation, s^{-1}

Subscripts

∞	ambient condition
c	critical
$cond$	conduction
$conv$	convection
exp	experimental
f	film
max	maximum
num	numerical
rad	radiation
X	insulation cover
Y	local value from the leading edge

work on the characterisation of convection heat transfer from these structures [26].

Correlations for external heat transfer from cuboids are scarce and often approximated to a flat surface with sharp edges [27], which usually give a wrong estimate due to boundary layer separation commonly found in the flows surrounding finite bodies, even at low Reynolds numbers [28,29]. Furthermore, the Prandtl number Pr has an influence on the transition threshold from laminar to turbulent flow in natural convection [30,31]. Air ($Pr = 0.7$) has been the main focus as a working fluid due to its broad applications, and hence it is also the working fluid chosen in this study.

Combined forced and natural convection, or mixed convection, occurs when an externally-imposed flow (such as wind) and buoyancy forces act together to transfer heat. The effect of mixed convection can be estimated by the Richardson number, which is a dimensionless number that quantifies the ratio of the buoyancy effect in natural convection to the momentum effect in forced convection. The Richardson number Ri is defined as

$$Ri = \frac{Gr}{Re^2}, \tag{1}$$

where Gr is the Grashof number and Re is the Reynolds number, which are dimensionless numbers defined as

$$Re = \frac{UL}{\nu}, \tag{2a}$$

$$Gr = \frac{g\beta\Delta TL^3}{\nu^2}, \tag{2b}$$

where U is the magnitude of the flow velocity (or freestream wind speed, $U = U_\infty$), L is the characteristic length of the object which for the cuboid in this study is defined as the side of a square isothermal surface,[1] ν is the kinematic viscosity, g is gravity, β is the thermal expansion coefficient, and $\Delta T = T_w - T_\infty$ is the temperature difference between the isothermal surface T_w and the freestream flow T_∞. From Eqs. (1) and (2), the Richardson number then becomes

$$Ri = \frac{g\beta\Delta TL}{U^2}, \tag{3}$$

which interestingly does not depend on the viscosity.

Iwatsu et al. [32] studied the effect of mixed convection in a lid-driven cavity. They showed that for that geometrical configuration and $Ri \ll 1$ the buoyancy effect is out-weighted by forced convection, whereas for $Ri \gg 1$ prominent buoyancy-driven convective features are discernible. The Richardson number threshold for the transition from forced to mixed convection depends on the geometry and flow characteristics [33]. In another study on mixed convection heat transfer from a small vertical cylinder placed in a cross flow [34], it was shown that, for a cylinder of equal length and diameter, Richardson numbers of $Ri < 1.4 \times 10^{-3}$ contain both predominantly forced and fairly mixed convection regimes. The mixed convection regime corresponded to $Ri > 6.7 \times 10^{-4}$ which provides an indication of the convection regimes around other geometries, such as the cuboid considered in this study.

Despite its shape being closely related to numerous applications, there are no studies on mixed convection from heated

[1] For an isothermal system consisting of a cuboid submerged in a fluid, a characteristic length reasonably could be chosen to be the longest side of the cuboid. For a non-isothermal system, however, the characteristic length in the Reynolds and Grashof numbers should coincide to avoid singularities of the local Richardson number.