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Convective Heat Transfer Coefficients for Solar Chimney Power Plant Collectors

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1. Introduction

This chapter deals with internal heat transfer in Solar Chimney Power Plant Collectors (SCPP), a typical symmetric sink flow between two disks. In general, specific heat transfer coefficients for this kind of flow can not be found in the literature and, consequently, most of the works employs simplified models (e.g. infinite plates, flow in parallel plates, etc.) using classical correlations to calculate the heat flow in SCPP collectors.
The extent of the chapter is limited to the analysis of the steady, incompressible flow of air including forced and natural convection. The phenomena phase change, mass transfer, and chemical reactions have been neglected. To the author' expertise, the most precise and updated equations for the Nusselt number found in the literature are introduced for use in SCPP heat flow calculations.
SCPPs consist of a transparent collector which heats the air near the ground and guides it into the base of a tall chimney coupled with it, as shown in Fig. 1. The relatively lighter air rises in the chimney promoting a flow allowing electricity generation through turbines at the base of the chimney. The literature about SCPP is extensively referred by (Bernardes 2010) at that time and there is no means of doing it here.

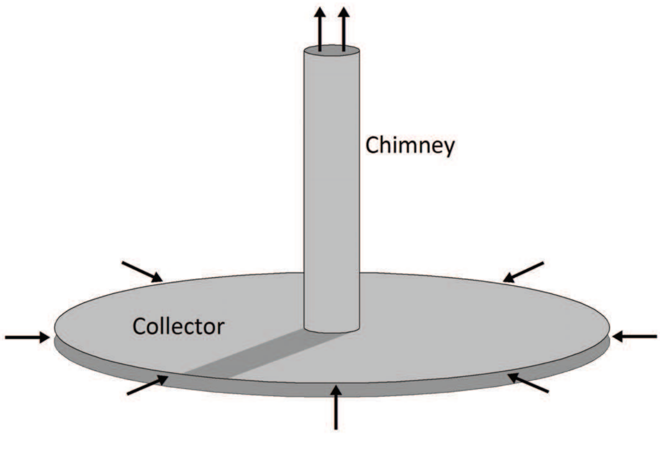

Fig. 1. Sketch of a SCPP.

The problem to be addressed here is the flow in the SCPP collector, i.e., the flow between two finite stationary disks concentrating on radially converging laminar and turbulent flow development and heat transfer. However, as a typical solar radiation dependent device, laminar, transient and turbulent and natural, mixed and forced convection, as well, may take place in the collector. Additionally, due to non uniform solar heating or ground roughness, the flow in collector should not converge axi-symmetrically. For that reason, the forced/natural convective flow in the SCPP collector can be treated as a flow:

1. between two independent flat plates in parallel flow or,
2. in a channel between parallel flat plates in parallel flow or
3. between two finite stationary disks in converging developing flow.

Moreover, collector convective heat transfers determine the rate at which thermal energy is transferred:

- between the roof and the ambient air,
- between the roof and the air inside the collector,
- between the absorber and the collector air.

It is necessary to remember that the literature for some typical heat transfer problems is extensive but scarce or even inexistent for some boundary conditions like constant heat flux, or for flow above rough surfaces like the collector ground.

1.1 Influence of the roof design in the heat transfer in collector

An important issue regarding the SCPP collector is its height as function of the radius. Some studies found in the literature (Bernardes 2004, Bernardes et al. 2003, Schlaich et al. 2005) make use of a constant height along the collector (Fig. 3). In this case, the air velocity increases continually due to the cross section decrease towards the chimney reducing the pressure in the collector, as shown in Fig. 2. Such pressure difference between the collector and surroundings allied with unavoidable slight gaps in the collector roof can result in fresh air infiltration reducing the air temperature. Furthermore, velocity variations in the collector denote different heat flows and, in this case, higher heat transfer coefficients and, consequently, a fresher collector close to the chimney. Besides, the relatively reduced collecting area in this region represents also lower heat gains harming the collector performance.

Fig. 2 also illustrates the air velocity for slight slanted roofs, evidencing a kind of 'bathtub effect'. Through this effect, the air velocity drops after the entrance region due to the cross area increasing and, especially for greater angles like 0.1° and 0.5°, remains minimal until achieves the chimney immediacy. In this region, the air velocity increases exponentially. Such air velocity profiles in collector represents lower heat transfer coefficients for a great collector area and, thus, lower heat transfer to the flowing air – predominance of natural convection – and higher losses to the ambient. Consequently, for this arrangement, the collector efficient would be inferior. (Bernardes et al. 1999) also disclose the presence of swirls when the flat collector roof is slanted.

The roof configuration for constant cross area – adopted by (Kröger & Blaine 1999, Pretorius & Kröger 2006) – leads, obviously, to constant air velocity in collector and, in terms of heat transfer, is the most appropriate for the collector. However, the roof height can achieve large values leading to higher material consumption (Fig. 3).

Lastly, the air velocity in the chimney should be taken in account. For a chimney diameter of 120 m, a collector diameter of 5000 m, an entrance collector height of 1 m and an entrance air velocity of 1 m/s, the air velocity in the chimney is 3 m/s approximately (continuity equation). Consequently, lower collector air velocities at the chimney entry are preferably and the roof configuration for constant cross area fits relatively well this condition.

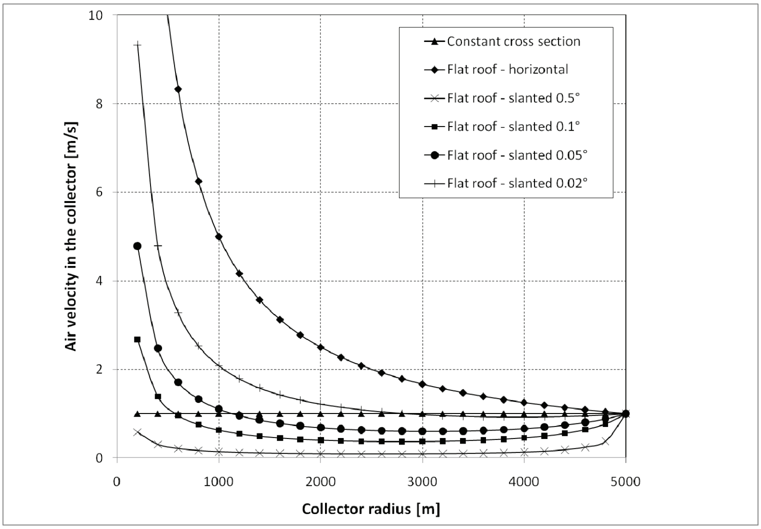

Fig. 2. Air velocity in collector for different roof arrangements.

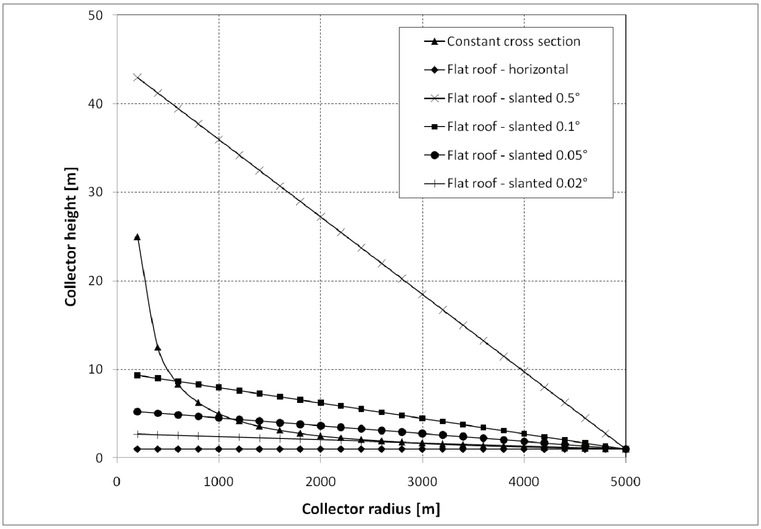

Fig. 3. Roof height for different roof arrangements.

2. Flow in collector as a flow between two independent flat plates

Due to a reasonable relative distance between the ground surface and the collector roof, the flow in a SCPP collector can be regarded as a flow involved by two independent plates, as employed be (Bernardes 2004). In this way, it is necessary to assume that the boundary layers develop indefinitely and separately.

2.1 Forced convection

The forced convection takes plate in the SCPP collector when the incident radiation is able to heat up the collector as much as necessary to promote a continuing air flow. In this condition, higher heat transfer coefficients are expected.
As represented in Fig. 4, the boundary layer flow over a flat plate regarding forced convection develops from a laminar boundary layer becoming unstable and turbulent after a certain plate length, when $Re_x = u_\infty \cdot x/\nu \approx 5 \times 10^5$. A more detailed description of this flow can be widely found in the literature, for instance, (Çengel 2007, Incropera 2007, Rohsenow et al. 1998), etc. In the following, the most important heat transfer coefficients and Nusselt numbers are introduced and their extent of application discussed.

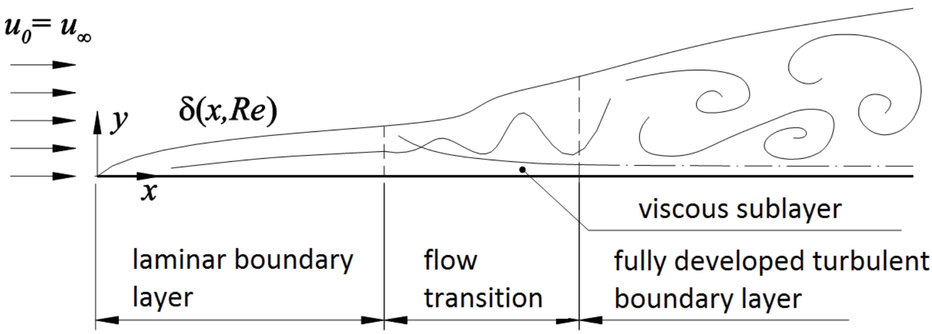

Fig. 4. Flow development in a flat plate in parallel flow.

Fundamentally, solution methodologies for the governing equations are based on the nondimensional groups and analytical means can be used to solve only a limited number of cases. Otherwise, experimental or numerical solution procedures must be employed.

Laminar flow – prescribed temperature

For two-dimensional cases, where the flow is laminar up to the point of transition to turbulent flow or flow separation, established analytical solutions can be extensively found in literature.
For the case of the laminar flow over a flat plate at uniform temperature and Pr ≈ 1 (like air, for instance), the similarity equations approach returns the local Nusselt number showed by equation (1) and the average Nusselt number by equation (2).

$$Nu_x = 0.332 \mathrm{Re}_x^{1/2} \mathrm{Pr}^{1/3} \quad (1)$$

$$\overline{Nu} = 0.664 \mathrm{Re}_L^{1/2} \mathrm{Pr}^{1/3} \quad (2)$$

$$Nu_x = \frac{\sqrt{\mathrm{Re}}\sqrt{\mathrm{Pr}}}{\sqrt{\pi}\left(1+1.7\,\mathrm{Pr}^{1/4}+21.36\,\mathrm{Pr}\right)^{1/6}} \qquad 0.25 \le \mathrm{Pr} \le \infty \tag{3}$$

(Baehr & Stephan 1996)

Laminar flow – Uniform wall heat flux

For a flat plate subjected to uniform heat flux instead of uniform temperature, the local and average Nusselt number are given by equations (4) and (5) respectively.

$$Nu_x = 0.453\,\mathrm{Re}_x^{1/2}\,\mathrm{Pr}^{1/3} \qquad \mathrm{Pr} > 0.6 \tag{4}$$

(Çengel 2007)

$$\overline{Nu} = 0.6795\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} \tag{5}$$

(Lienhard IV & Lienhard V 2008)

$$Nu_x = \frac{\sqrt{\pi}\sqrt{\mathrm{Re}}\sqrt{\mathrm{Pr}}}{2\left(1+2.09\,\mathrm{Pr}^{1/4}+48.74\,\mathrm{Pr}\right)^{1/6}} \qquad 0.25 \le \mathrm{Pr} \le \infty \tag{6}$$

(Baehr & Stephan 1996)

Turbulent flow – prescribed temperature

For the case of turbulent flows, approximate analytical solutions based on phenomenological laws of turbulence kinetics are established for local and average Nusselt numbers as introduced by equations (7), (8) and (9).

$$Nu_x = 0.032\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{0.43} \qquad 2\times10^5 < \mathrm{Re}_x < 5\times10^6 \tag{7}$$

(Žukauskas & Šlanciauskas 1999)

$$Nu_x = 0.0296\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{1/3} \qquad \begin{array}{c} 5\times10^5 \le \mathrm{Re}_x \le 10^7 \\ 0.6 \le \mathrm{Pr} \le 60 \end{array} \tag{8}$$

(Çengel 2007)

$$\overline{Nu} = 0.037\,\mathrm{Re}_L^{0.8}\,\mathrm{Pr}^{1/3} \qquad \begin{array}{c} 5\times10^5 \le \mathrm{Re}_L \le 10^7 \\ 0.6 \le \mathrm{Pr} \le 60 \end{array} \tag{9}$$

(Çengel 2007)

Entire plate

A relation suitable to calculate the average heat transfer coefficient over the entire plate including laminar and turbulent is given by equation (10).

$$\overline{Nu} = \left(0.037\,\mathrm{Re}_L^{0.8} - 871\right)\mathrm{Pr}^{1/3} \qquad \begin{array}{c} 5\times10^5 \le \mathrm{Re}_L \le 10^7 \\ 0.6 \le \mathrm{Pr} \le 60 \end{array} \tag{10}$$

(Çengel 2007)

Turbulent flow – uniform wall heat flux

When the turbulent flow over a flat plate is subjected to uniform heat flux, the local and average Nusselt numbers are given by equations (11) (12) and (13).

$$Nu_x = 0.0308\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{1/3} \tag{11}$$

(Çengel 2007)

$$\overline{Nu} = 0.037\,\mathrm{Re}_L^{0.8}\,\mathrm{Pr}^{0.43} \qquad 2\times10^5 \le \mathrm{Re}_L \le 3\times10^7 \tag{12}$$

(Lienhard IV & Lienhard V 2008)

$$Nu = \frac{0.037\,\mathrm{Re}^{0.8}\,\mathrm{Pr}}{1+2.443\,\mathrm{Re}^{-0.1}\left(\mathrm{Pr}^{2/3}-1\right)} \quad 5\times10^5 < \mathrm{Re} < 10^7 \quad 0.6 < \mathrm{Pr} < 2000 \tag{13}$$

(Petukhov & Popov 1963)

Mixed forced convection

$$Nu = \sqrt{Nu_{lam}^2 + Nu_{turb}^2} \qquad 10 < \mathrm{Re} < 10^7 \tag{14}$$

(Baehr & Stephan 1996)

2.2 Natural convection

Natural convection at horizontal isothermal plates of various planforms with unrestricted inflow at the edges are related with the correlations presented by equations (15), (16) and (17).

$$Ra = \frac{g\beta\,\overline{\Delta T}\left(L^*\right)^3}{\nu\alpha} \tag{15}$$

$$Nu = \frac{qL^*}{A\,\overline{\Delta T}\,k} \tag{16}$$

$$L^* = A / p \tag{17}$$

Uniform Heat Flux Parallel Plates

Classical correlations:

(Lloyd & Moran 1974) presented correlations for natural convection at horizontal isothermal plates taking into account both hot side up or cold side down and hot side down or cold side up, as shown by equations (18), (19) and (20).

$$Nu = 0.54\,Ra^{1/4} \qquad 10^4 \le Ra \le 10^7 \tag{18}$$

hot side up or cold side down, (Lloyd & Moran 1974)

$$Nu = 0.15Ra^{1/3} \qquad 10^7 < Ra \le 10^{10} \tag{19}$$

hot side up or cold side down, (Lloyd & Moran 1974)

$$Nu = 0.27Ra^{1/3} \qquad 10^5 < Ra \le 10^{10} \tag{20}$$

hot side down or cold side up, (Lloyd & Moran 1974)

(Rohsenow et al. 1998) introduced correlations for heated upward-facing plates with uniform temperature or heat flux (1 < Ra < 10^{10}), namely, equations (21), (22), (23) and (24).

$$Nu = 0.835\bar{C}_l Ra^{1/4} \qquad 1 < Ra < 10^{10} \quad \bar{C}_l = 0.515 \text{ for air} \tag{21}$$

$$Nu_{lam} = \frac{1.4}{\ln\left(1 + 1.4 / Nu\right)} \tag{22}$$

$$Nu_{turb} = C_t^U Ra^{1/3} \qquad C_t^U = 0.14 \text{ for air} \tag{23}$$

$$Nu_t = \left(Nu_{lam}^m + Nu_{turb}^m\right)^{1/m} \qquad m = 10 \tag{24}$$

On the other hand, for horizontal isothermal heated downward-facing plates, equations (25) and (26) are suggested by (Tetsu et al. 1973) for 10^3 < Ra < 10^{10}.

$$Nu = \frac{0.527}{\left(1 + \left(1.9 / \mathrm{Pr}\right)^{9/10}\right)^{2/9}} Ra^{1/5} \qquad 10^3 < Ra < 10^{10} \tag{25}$$

$$Nu_{lam} = \frac{2.5}{\ln\left(1 + 2.5 / Nu\right)} \tag{26}$$

Mixed natural convection

$$Nu^4 = Nu_{lam}^4 + Nu_{turb}^4 \tag{27}$$

(Baehr & Stephan 1996)

Free and forced convection including radiative heat flux

The work by (Burger 2004) introduced correlations, which took into account significant natural convection mechanisms by evaluating convective and radiative heat fluxes onto or from a smooth horizontal flat plate exposed to the natural environment. As shown in equation (28), T_m is the mean temperature between the collector roof and ambient air, g is the gravitational constant and ΔT is the difference between the roof and ambient air temperature. The variables ρ, μ, c_p and k symbolize the density, dynamic viscosity, specific heat capacity and thermal conductivity of the air respectively, all of which are evaluated at the mean temperature T_m. If the collector roof temperature only marginally exceeds the ambient temperature equation (29) can be employed. Equation (30) was derived by (Kröger 2004) using Gnielinski's equation for fully developed turbulent flow, by approximating the flow in the collector as flow between variably spaced plates.

$$h = \frac{0.2106 + 0.0026\, v\left(\dfrac{\rho T_m}{\mu g \Delta T}\right)^{1/3}}{\left(\dfrac{\mu T_m}{g\,\Delta T\, c_p k^2 \rho^2}\right)^{1/3}} \tag{28}$$

(Burger 2004), [1]

$$h = 3.87 + 0.0022\left(\frac{v\rho c_p}{\Pr^{2/3}}\right) \tag{29}$$

(Burger 2004), [1]

$$h = \frac{(f/8)(\mathrm{Re}-1000)\Pr}{1+12.7(f/8)^{1/2}\left(\Pr^{2/3}-1\right)}\left(\frac{k}{D_h}\right) \tag{30}$$

(Burger 2004)

3. Flow in collector as a flow in a channel between infinite parallel plates

The development of the hydrodynamic and thermal boundary layers can produce four types of laminar flows in ducts, namely, fully developed, hydrodynamically developing, thermally developing (hydrodynamically developed and thermally developing), and simultaneously developing (hydrodynamically and thermally developing), as sketched by Fig. 5. In this case particularly, the velocity profile and dimensionless temperature profile do not change along the flow direction. On the other hand, hydrodynamically developing flow is isothermal fluid flow in which the velocity profile does not remains constant in the flow direction.
The hydrodynamic entrance length is defined as the distance over which the velocity distribution changes and the hydrodynamic boundary layer develops. (Rohsenow et al. 1998) synthesize those phenomena, as shown in Table 1.

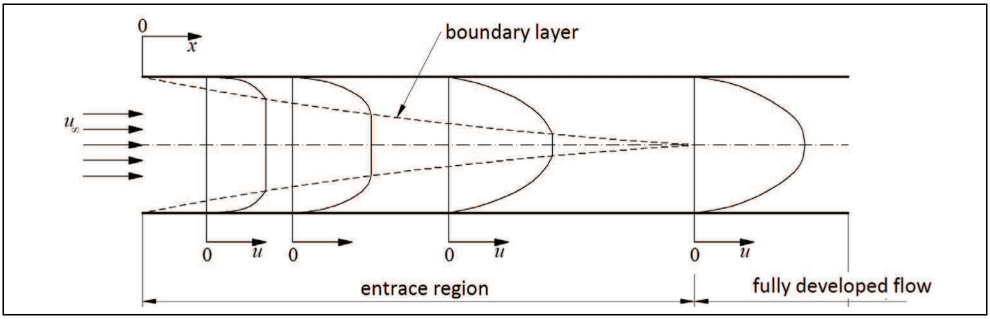

Fig. 5. Flow development in parallel plates channel.

[1] $v = v_w$ for heat transfer between roof and ambient.

Flow type	Hydrodynamic boundary layer	Velocity distribution in the flow direction	Thermal boundary layer	Dimensionless temperature distribution in the flow	Nussel number
Fully developed Flow	Developed	Invariant	Developed	Invariant	Constant
Hydrodynamically Developing flow	Developing	Variant	-	-	-
Thermally Developing flow	Developed	Invariant	Developing	Variant	Variant
Simultaneously Developing flow	Developing	Variant	Developing	Variant	Variant

Table 1. Flow types in parallel plates channel.

In general, the flow in a SCPP collector can be assumed as simultaneously developing flow. Thus, the following equations are recommended for a typical flow in a channel between infinite parallel plates.

Laminar Flow – Fully Developed Flow

For a parallel plate duct with hydraulic diameter D_h = 4t (*t* represents the half-distance between the plates). According to (Shah & London 1978), the following equations for the corresponding boundary conditions.
Uniform temperature, the other wall at the uniform entering fluid temperature: eq. (31)

$$Nu_1 = Nu_2 = 4 \tag{31}$$

Uniform wall heat flux at one wall; the other wall insulated: eq. (32)

$$Nu_1 = 0 \quad Nu_2 = 5.385 \tag{32}$$

Uniform temperature (different from the entering fluid temperature) at one wall; the other wall insulated: eq. (33)

$$Nu_1 = 0 \quad Nu_2 = 4.861 \tag{33}$$

Uniform wall heat flux at one wall; the other wall maintained at the entering fluid temperature: eq. (34)

$$Nu_1 = Nu_2 = 4 \tag{34}$$

Laminar Flow –Uniform temperature at each wall: eq. (35)

$$Nu_1 = Nu_2 = 7.541 \tag{35}$$

(Shah & London 1978)

Laminar Flow – Uniform temperature at one wall and uniform heat flux at the other: eq. (36)

$$Nu_1 = 4.8608 \quad Nu_2 = 0 \tag{36}$$

(Shah & London 1978)

Laminar Flow – Developing Flow – Equal and uniform temperatures at both walls: eqs. (37) and (38)

$$Nu_x = 7.55 + \frac{0.024x^{*-1.14}\left[0.0179\,\mathrm{Pr}^{0.17}\,x^{*-0.64} - 0.14\right]}{\left[1 + 0.0358\,\mathrm{Pr}^{0.17}\,x^{*-0.64}\right]^2} \tag{37}$$

(Shah & Bhatti 1987), (Hwang & Fan 1964), (Stephan 1959)

$$\overline{Nu} = 7.55 + \frac{0.024x^{*-1.14}}{1 + 0.0358\,\mathrm{Pr}^{0.17}\,x^{*-0.64}} \tag{38}$$

(Shah & Bhatti 1987), (Hwang & Fan 1964), (Stephan 1959)

Turbulent Flow

Transition Flow

The mean Nusselt number in the thermal entrance region of a parallel plate duct with uniform wall temperature at both walls in the range of 2300 < Re < 6000 is given by (Hausen 1943) as follows:

$$\overline{Nu} = 0.116\left(\mathrm{Re}^{2/3} - 160\right)\mathrm{Pr}^{1/3}\left[1 + \left(\frac{x}{D_h}\right)^{-2/3}\right] \tag{39}$$

(Hausen 1943)

Fully developed flow: eq. (40).

$$Nu_H = \frac{Nu}{1 - \gamma\theta^*} \tag{40}$$

(Hausen 1943)

$\gamma = 0$: one wall is heated and the other is insulated;
$\gamma = 1$: uniform heat fluxes of equal magnitudes are applied to both walls;
$\gamma = -1$: heat transfer into one wall and out of the other wall, while the absolute values of the heat fluxes at both walls are the same.

Nusselt Numbers and Influence Coefficients for Fully Developed Turbulent Flow - Parallel Plates Duct With Uniform Heat Flux at One Wall and the Other Wall Insulated, Pr = 0.7, (Kays & Leung 1963)

Re = 10^4		3×10^4		10^5		3×10^5		10^6	
Nu	θ*	Nu	θ*	Nu	θ*	Nu	θ*	Nu	θ*
27.8	0.220	61.2	0.192	155.0	0.170	378.0	0.156	1030.0	0.142

Natural Convection within Enclosures

The natural convection in the collector can regarded as natural convection in horizontal rectangular enclosures. Nusselt number relations for this are introduced in the following, namely, equations (41) and (42).

$$Nu = 0.195Ra^{1/4} \qquad 10^4 < Ra < 4\times10^5 \tag{41}$$

(Jakob 1949)

$$Nu = 0.068Ra^{1/3} \qquad 4\times10^5 < Ra < 10^7 \tag{42}$$

(Jakob 1949)

Equation (43) is suggested by (Hollands et al. 1976). Their study was based on experiments with air correlation for horizontal enclosures. The notation $[\ ]^{+}$ indicates that if the quantity in the bracket should be set equal to zero if it is negative.

$$Nu = 1 + 1.44\left[1 - \frac{1708}{Ra}\right]^{+} + \left[\frac{Ra^{1/3}}{18} - 1\right]^{+} \qquad Ra < 10^8 \tag{43}$$

4. Flow in collector as a flow between two finite stationary disks with converging flow developing

The flow between two finite stationary disks with converging flow developing and constant distance between the plates ($2t$) was investigated by (Bernardes 2003). Thermal and hydrodynamic boundary layers, friction factor, pressure development and Nusselt number were derived. For lager values of the dimensionless radii (>> 1), the velocity profile becomes parabolic and invariant and the friction factor approaches the classic value obtained for fully developed flow between infinite plates. At radii less than one a typical external boundary layer evolves close to the wall with an approximately uniform core region, the boundary layer thickness decreases from one-half the disk spacing to values proportional to the local radii as the flow accelerates. The local Nusselt number decreases with the radius and the eq.(44) is recommended for thermally developing flow. The Nusselt number for thermally fully developed flow is a straight line: $Nu \approx (r/R)$.

$$Nu = 230\left(\frac{r}{R}\right)^{0.650}\left(1 - \frac{r}{R}\right)^{-0.386} \tag{44}$$

7. Nomenclature

A	area	m²
c_p	heat capacity	J/(kg·K)
d_h	hydraulic diameter	m
k	thermal conductivity	W/(m·K)
f	friction factor	-
g	gravitational acceleration	m²/s
h	convective heat transfer coefficient	W/(m²·K)
L^*	plate length	m
L	plate length	m
Nu	Nusselt number	-
Nu_x	local Nusselt number	-
Nu_{lam}	Nusselt number for laminar flow	-

Nu_{turb}	Nusselt number for turbulent flow	-
$\overline{Nu}$	average Nusselt number	-
p	perimeter	m
Pe	Peclet number	-
Pr	Prandtl number	-
q	heat transfer rate	W
r	cylindrical coordinate	m
R	disc radius	m
Ra	Rayleigh number	-
Re	Reynolds number	-
Re_x	local Reynolds number	-
Re_L	Reynolds number for a plate with length L	-
t	half-distance between the plates	m
T	temperature	K
T_m	average temperature	K
v	velocity	m/s
x^*	dimensionless axial coordinate for the thermal entrance region, = x/D_h Pe	
α	fluid thermal diffusivity	m^2/s
β	coefficient of thermal expansion	K^{-1}
γ	ratio of heat fluxes at two walls of a parallel plate duct	-
ν	fluid kinematic viscosity	m^2/s
μ	fluid dynamic	Pa·s
ρ	density	kg/m^3

subscripts

1,2	plate 1, 2, etc.

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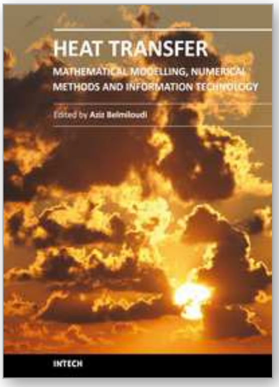

Heat Transfer - Mathematical Modelling, Numerical Methods and Information Technology

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Over the past few decades there has been a prolific increase in research and development in area of heat transfer, heat exchangers and their associated technologies. This book is a collection of current research in the above mentioned areas and describes modelling, numerical methods, simulation and information technology with modern ideas and methods to analyse and enhance heat transfer for single and multiphase systems. The topics considered include various basic concepts of heat transfer, the fundamental modes of heat transfer (namely conduction, convection and radiation), thermophysical properties, computational methodologies, control, stabilization and optimization problems, condensation, boiling and freezing, with many real-world problems and important modern applications. The book is divided in four sections : "Inverse, Stabilization and Optimization Problems", "Numerical Methods and Calculations", "Heat Transfer in Mini/Micro Systems", "Energy Transfer and Solid Materials", and each section discusses various issues, methods and applications in accordance with the subjects. The combination of fundamental approach with many important practical applications of current interest will make this book of interest to researchers, scientists, engineers and graduate students in many disciplines, who make use of mathematical modelling, inverse problems, implementation of recently developed numerical methods in this multidisciplinary field as well as to experimental and theoretical researchers in the field of heat and mass transfer.

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