NATURAL CONVECTION AT ISOTHERMAL VERTICAL PLATE: NEIGHBOURHOOD INFLUENCE

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Abstract. *This work deals with heat transfer coefficient "h" of a isothermal vertical plate with $H = 0.15$ m. The neighbourhood surfaces innfluence in thar coefficient is aimed with simulation and standard experimentation. A novel technology to measure the heat flux, caling "Tangential Heat Fluxmeter" is applied and simulation with a CFD commercial code was performing. Five heat fluxmeters was glued on the vertical plate, heated $20\ ^0C$ over the air temperature. The neighbouhood and air temperature was maintened constants. The distance between the plate and base wall (floor) was changed as well as the distance between the plate and back side wall. Through simulation results will be compare with experimentals. The result expected is a increasing of heat tranfer coefficient, very usefully in heat exchange devices.*

Keywords: *Natural Convection, Fluxmeter, Vertical Plate, Simulation, Exprimentation*

1. INTRODUCTION

In numerous thermal applicatons the problems of control and measurement of heat transfer by natural or forced canvection are very importants. In most cases is impossible a theorical approach, them experimentation (numerical or standard exprimentation) is inevitable. This paper presents a methodology to determine the influence of neighbourhood on the convective heat transfer coefficient, basead on the use of heat transfer fluxmeters and numerical simulation method. Severals papers comment around vertical plate in many configurations but is strongly dificult to find the configuration shows in figure 1, with the neighbourhood influences considerated. The published studies on the literature, in general, deals eith the simplest geometries: a single isothermic vertical plate [Coutanceau, 1969] and [Mahajan and Gebhart, 1979] and [Bill and Gebhart, 1979] and [Tsuji and Nagano, 1988], wire heated reported by [Rohsenow et al., 1998], two plates uniform wall on a vertical channel of uniform cross-section [Elenbaas, 1942] and [Bar-Cohen and Rohsenow, 1984] and [Elembaas, 1942] and [Sparrow and Azevedo, 1985] and [Azevedo and Sparrow, 1985] as well as [Kim et al., 1990]. The device is easily assembled and very accurate tool for the measure of heat transfer in surfaces immersed, not only in air, but also in liquids and multiphase fluid flows. The validation of the numerical method was based on the Ostrach's results [Ostrach, 1952], his technical report provides an analytical correlation for Nusselt number which allows the easy determination of "h" $\left[\frac{W}{mK}\right]$.

2. EXPERIMENTAL APPARATUS AND PROCEDURES

2.1 Flat Pate Apparatus

A vertical flat plate is a cooper plate $(0.15 \times 0.15 \times 0.002\ m^3)$ heated by a skin heater in Constantan supplied by a constant tension source. Condition of constant temperature is obtained

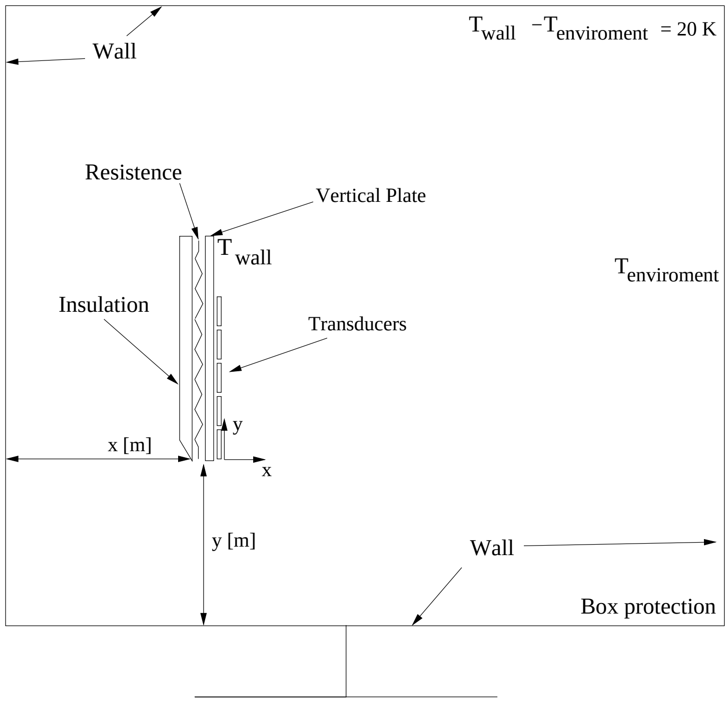


Figure 1. General Overview: Experimental apparatus

in a half-hour *(steady state)*. Temperature difference along the plate is smaller than $0.1\ ^0C$. The backside surface is isolated by $0.02\ m$ of expanded polystyrene with an epoxy sheet, this insulation is very important to warranty that the air behind board will not be heated by this board. Distance of base and the behind wall can be adjusted to permit study the influence in the entrance region on the natural convection. The system is protected of the surroundings perturbations by a aluminium box $\left(1.0 \times 1.0 \times 0.6\, m^3\right)$. The box must be ensuring a good exchange of heat produced in the flat polished plate and an external fan can be provided to increase the global external heat transfer coefficient. Surroundings are maintained at constant temperature $\left(\sim\ 25\ ^0C\right)$.

2.2 Heat Fluxmeters

The sensors, developed by [Thery et al., 1980] and [Güths, 1994], called "Tangential Gradient Heat Fluxmeters" have the major advantage of being very thin $(300\,\mu m)$ and are very sensitive $\left(30\ \frac{\mu V}{W/m^2}\right)$, for a $\left(0.05 \times 0.01\ m^2\right)$ sensor area. A *constantan* strip is placed deposited on a *kapton* support with *cooper* deposits on it. Periodic volumetric gaps etched into the *cooper* plate allows an asymmetrical constriction of any heat flow reaching the sensor surface.

Therefore, the *cooper/constantan* plated thermoeletrical circuit is subject to tangential thermal gradients. So, the heat or cooling rate can be directly measure out as a function of voltage. When these thermoeletric elements are placed in series, an *e.m.f.* proportional to the thermal flux is generated. Many articles ([Leclercq and Thery, 1983] and [Lassue et al., 1993]) have been published showing the negligible disturbance of these sensors during the measurement in numerous applications. Sensors are glued side by side over an aluminiun sheet $(100\,\mu m\ in\ thickness)$. A litle space between the sensors is filled by epoxy glue. This space is necessary to reduce the lateral heat transfer, wich is source of error at the calibration and local "h" coefficient. The space is fill to not disturb the fluid flow profile.

Figure 2 exemplify these remarks.

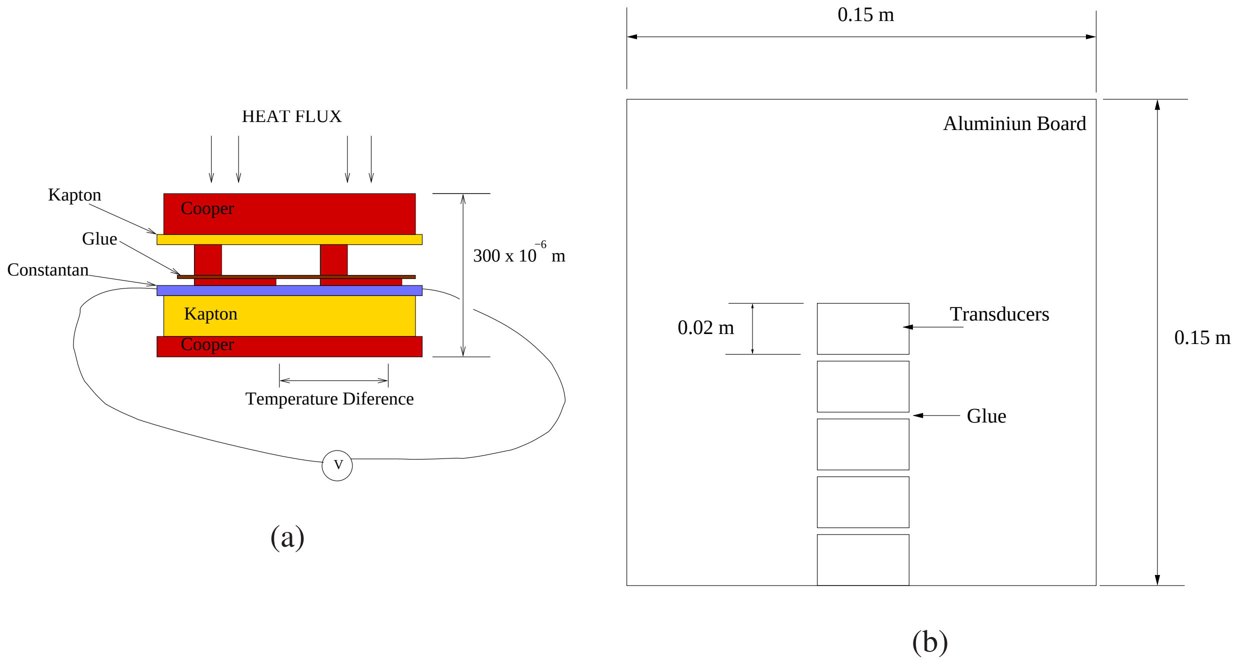


Figure 2. Tranducers: (a) Fluxmeter, (b) Fluxmeter plus Board assembly

2.3 Calibration of Fluxmeters

The calibration technique presented can be make "in situ", and used as a part of the experiment. The output signal is an *e.m.f.* $[V]$ proportional to the heat flux $\left(q^{"}\right)$,

$$q^{"} = cV \left[\frac{W}{m^2}\right] \tag{1}$$

where c is the calibration constant. Calibration procedures consists to submit the sensor to a known heat flux. The flux is generated by a skin heater with the same surface area of the sensor $\left(0.052 \times 0.099\ m^2\right)$ supplied by a constant voltage source. the constant c can be determined as:

$$c = \frac{RI^2}{SV}\left[\frac{W}{\mu V m^2}\right] \tag{2}$$

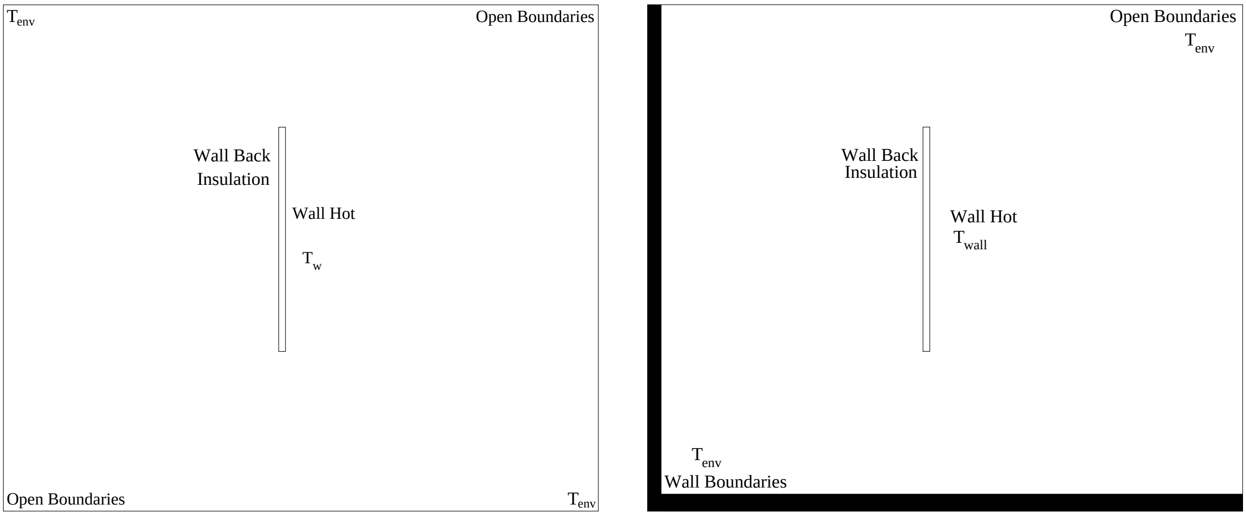
where I is the direct current measured by multimeter in $[A]$, R in $[\Omega]$ is the ohmic resitence of heater, $S\left[m^2\right]$ is the area of sensor and V is voltage of each sensor. The heat flux lost by the opposite surface is measured by another heat fluxmeter previously calibrated.

The calibrations was executed using the classical method above detailed, however each these transducers presented systematic errors that need of relative corrections based on Ostrach's results. Thus, the quality of experimentals values will be analysed, shown the tendency of coefficient in function of heightt $y\ [m]$.

2.4 Procedures

The standard experimental procedure was conducted on the steady state and laminar regime, typically attained after a half-hour. The polished flat plate, to avoid radiation exchange, was heated by a power supply with direct current kept constant. A data acquisition system is used to measure the voltage.

Experiments are conducted in a closed-box with zero air velocity field (the air movement will be buoyancy force cause only, near of vertical plate). Figure 1 shows the box scheme used. The



(a) (b)

Figure 3. Vertical plate modelling: (a) Open boundaries, (b) Wall boundaries

temperature of vertical plate was maintened constant $T_0 \approx 45\ ^0C$. Air temperature is approximately $T_\infty \approx 25\ ^0C$ resulting $\triangle T \approx 20\ ^0C$. Primarily this vertical plate was assembled, encased over insulation and heated until $\triangle T \approx 20\ ^0C$, matchable the Ostrach's case plate [Ostrach, 1952]. So, the voltage generated by $[\mu V]$ of sensors was measured by *HP Acquisition system* and the mean value was calculated. The results of heat transfer coefficient "h", in [Ostrach, 1952] was comparated with this experimentals values (mean values).

After this, the vertical plate was assembled far way any boundary, the classical mode (vertical free plate was called), without neighbourhood around. This assembling show values sligthly different of Ostrach's case.

That classical mode was modified approaching a base wall (like a floor). The values of voltage was measured by aquisition system and the mean values was calculated to after comparasion. At classical mode was approached the back wall, making channel on the posterior side of heated face.

The channel generated by approximation (back wall only or base wall and back wall) modify the air flow around vertical plate, increasing the heat transfer coefficient.

3. NUMERICAL APPROACH AND MATHEMATICAL FORMULATION

3.1 Numerical Modelling

The problem was solved trough the finite volume method on a two-dimensional structured grid built using ANSYS BUILD grid and mesh generator software. This grid was optimized to allow up to 30 volumes in the boundary layer (hydrodynamic or thermal). Boundaries conditions were assumed "openings boundaries" with imposed constant pressure and temperature ($P_{rel} = 0\ [atm]$) and ($T_\infty = 298\ [K]$), isothermal neighbour surfaces ($T = 298\ [K]$) and insulation $\left(\frac{\partial T}{\partial \eta} = 0\right)$ on back part of vertical plate. The governing equations, was solved using fluid properties available from ANSYS CFX Library. Numerous cases were simulated for different configurations layouts, where the distance from vertical plate until the neighbour surfaces was modified, for the same values of temperatures and pressures conditions. It was made for the analysis of influence these neighbourhood surfaces on the "h" coefficient.

Mesh discretisation was generated under Windows architeture PC with a usefull CAD software. The ANSYS CFX aided to created the geometry and mesh. A Beowulf Cluster with 8 nodes was used for simulations. Around 250000 volumes was created for the domain discretisation. How shown in figure 3 the domain was design in finite dimensions but the boundaries are imposed to simulate the infinite region. The mesh details is visualised in the figure 4. To compare the mesh dicretization the are in the same figure a surface colored by red indicating velocity boundary layer scale and surface green the temperature boundary layer scale. The board thickness have 5 *mm*, velocity boundary around 3 *mm* and thermal boundary layer have 2.5 *mm*. These were scales founded.

A deprecated poor mesh discretisation presents bad results for velocity, wall heat flux and all variables in question. This way the mesh choosed for to eliminate this variations. The mesh spacing around 0.0002 *m* up to 0.025 *m* was selected in a hexahedral volumes. How were compared in the figure 4.

The boundaries conditions is show in table 1.

Table 1. Boundaries Details

Boundary	Location	Value
Wall	Neighbourhood	Temp. 298 *K*
Opening	Entrance, Output surf.	298 *K* at 0 *Pa* (*relative pressure*)
Wall Back	Back side of Vertical Plate	Insulation
Wall Hot	Isothermal	Temp. 318 *K*

[Zamora and Hernández, 2001] say that the influence of variables like viscosity, density and others variables presents no considerable variation because the velocity variation is very low, the temperature is low been estimated in mean temperature ($\sim$ 308 K). The buoyancy fluid flow presents laminar regime charateristic, because the range of Rayleigh number is less than 1.0×10^9. The incompressible assumption was adopted and Boussinesq model for density too. Approach equations by [Bejan, 1995] for modell used are shown in equations 3, 4, 5, 6.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{3}$$

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial P}{\partial x} + \mu\nabla^2 u \tag{4}$$

$$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial P}{\partial x} + \mu\nabla^2 v - \rho g \tag{5}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\nabla^2 T \tag{6}$$

4. RESULTS AND DISCUSSION

Mesh discretisation is very important for a solution calculations. Convergence criterium and results depend of how discretisation was used. For this solutions in the present work around 250000 was generated, this means that less volume $\left[m^3\right]$ of control volumes is $0.2 \times 0.2 \times 1\ mm^3$ and more than 30 volumes was indicated for boundary layer discretisation. The mesh was created with exponential variation length node, more volumes was necessary in the boundary layer than so far way region.

The fluxmeters used presents a slightly sensibility at flow perturbation, but the results show that expected variation of coefficient, mainly wall Down approximation.

All simulated results was easily reached and maximums residuals were less than 1.0×10^{-6} shown excelent convergence. Mass, Heat Exchange and Pressure Imbalances on domain was less than

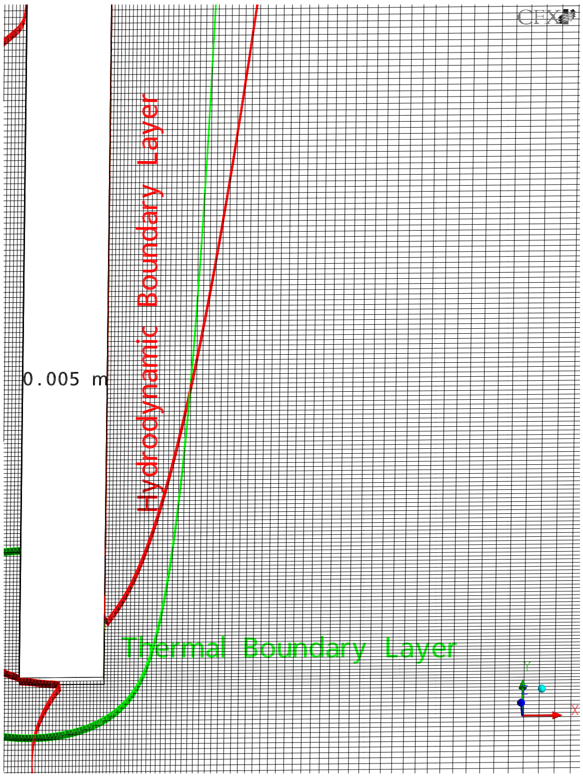


Figure 4. Mesh Model

0.00001 % indicanting the steady state. The experimentals values (mean values) of "h" coefficient profile in *y* heightt have very good aggreement with Ostrach's case and simulations values too. So, the comparation of experimentals results were checked in figure 5. The base wall approximation increasing the "h" coefficient until 22 %, shown in figure 6. Back wall approximation is too efficient, figure 7 do show this remark. At the approximation on the wall (neighbourhood) the convective flux do increasing the "h" coefficient. Like shown in figures 6, 7, 8, the channel generated b approximation difficult the air flux and to mass conservation is warranted ths velocity increase, increasing the Nusselt number on the heated face of board.

Figure 8 presents results for the cases which layout the heated board is approximated to Wall Down and Wall Left at the same time. The values of "h" depends of arrange layout choosed, those values increasing until a maximmun value to decrease until value shown in that figure.

The increasing caused by back side influence is around 23 % while the wall Down is 30 % indicating that wall Down approximation is slightly more efficient.

5. CONCLUSION

A numerical and standard experimental study of the interaction natural convection in a isothermal vertical plate finite-sized, with a neighbourhood vertical (back side) and horizontal (down placed) surface has been carried out. This configuration is general conception model of various eletronic equipments. The full governig elliptical equations was solved using ANSYS CFX 5.7.1 solver with a finite volume method and results obtained show the increasing of exchange heat transfer when wall down (horizontal surface) is aproximated of vertical plate more than the channel created from aproximation of back side wall (vertical neighbourhood).

Severals interesting and importants results are obtained, for low Rayleigh number $Ra \leq 10^6$. Such scales is encountered in pratical systems that employ thermal sources. Standard flow (Free Board) is modified by horizontal wall and the same way by by vertical back wall.

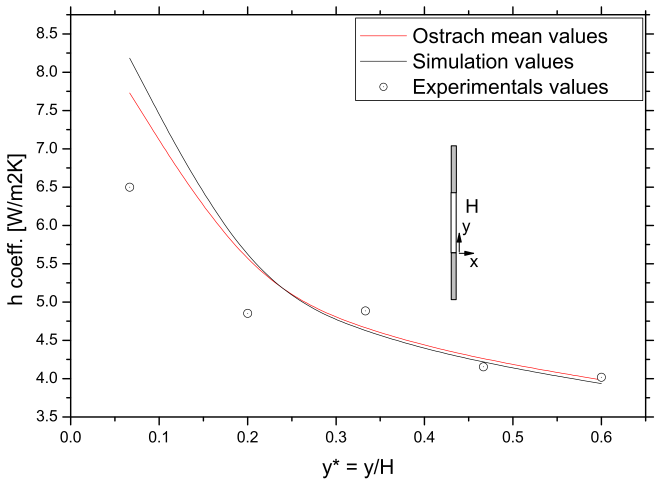


Figure 5. Comparation of Simulation, Experimentation and Analitical values

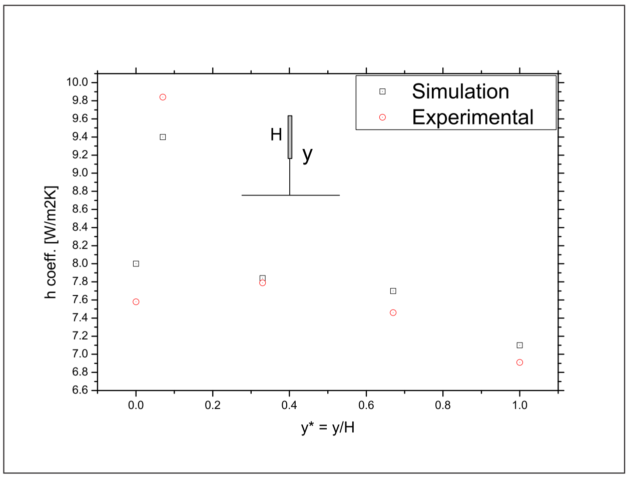


Figure 6. Base Wall Approximating

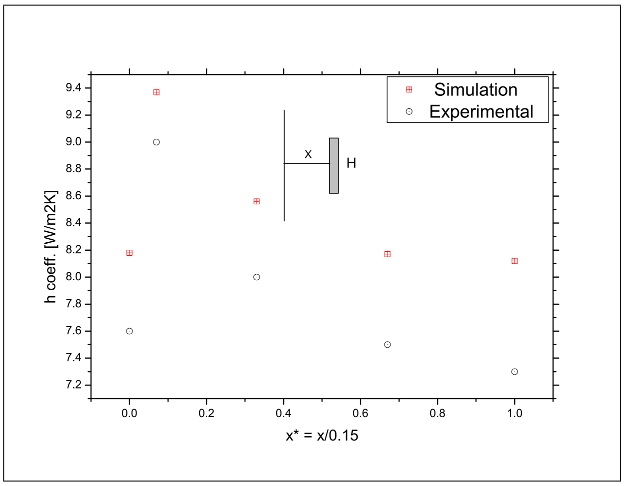


Figure 7. Back Wall Approximation

The experimentals values are obtained for various layouts with transducer aided, these fluxmeters are very slighty and thermal inertia is low and the accurate values are shown.

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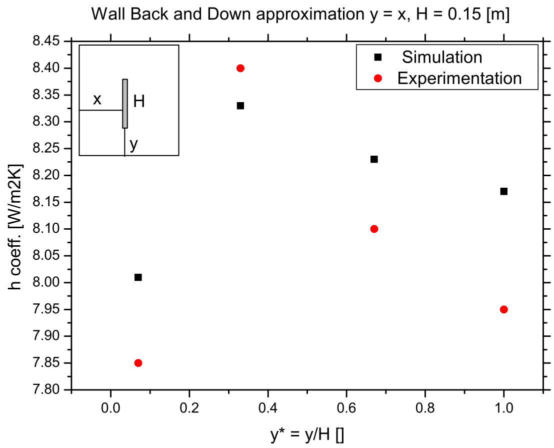

Figure 8. Comparation of Simulation and Experimentation

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