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Numerical Investigation of Natural Convection Heat Transfer Characteristics on Partially Heated Square Enclosure at Different Rayleigh Numbers

Farhana Isalm Muna, Mohammad Ilias Inam*
Department of Mechanical Engineering, Khulna University of Engineering & Technology, Khulna-9203, BANGLADESH

ABSTRACT
The paper presents a numerical analysis of natural convection inside a square cavity in which left and right wall were considered heated partially. Direct Numerical Simulation (DNS) was conducted by using Ansys FLUENT to observe the effect of partially heated section on the heat transfer characteristics at different Rayleigh Number (*Ra*) which was considered to 10^4 and 10^5. During these simulations, it was assumed that vertical walls (left one and right one) heated by 20%,40%,60%,80% and 100% with constant temperature. The obtained numerical results were illustrated and discussed with temperature contour, heat transfer co-efficient, local and average Nusselt Number, stream function in tabular and graphical form. These DNS results demonstrate that Nusselt Number (*Nu*) and average heat transfer coefficient, both, decrease along with increasing the percentage of heated portion, whereas increase with *Ra*.

Keywords: Partial heating, Nusselt Number, Heat transfer co-efficient, Rayleigh Number, Natural Convection

1. Introduction

Natural Convection in a closed enclosure has many engineering applications in different fields. Its applications are in solar energy systems, aerospace construction, nuclear reactor construction, development of room air, cooling of electronic devices, heat exchanger design and many others. Recently natural convection in a closed cavity has been studying experimentally and numerically. For these studies, many types of closed enclosure have been used. A brief review on natural convection in enclosure was given in a bibliography of Ostrach [1]. A numerical study of De Vahl Devis described the steady laminar two dimensional motion of a fluid in an enclosed rectangular [2] and square [3] cavity. A CFD analysis to study the free convection flow in a square cavity had been performed by Basak [4]. Effect of uniform and non-uniform heated hot wall on natural convection was studied here. Hinojosa [5] presented a study of steady-state and transient free convection in an isothermal open cubic enclosure, for different high Rayleigh numbers. Effect of the Prandtl and the Rayleigh numbers on flow characteristic in a square cavity was observed by Adnani [6]. A. Rincon-Casado [7] presented correlations to solve free convection heat transfer coefficients for building performance. N.C. Markatos [8] studied to obtain solutions for heat transfer in a square enclosure with partly heated vertical walls. Sajida [9] numerically studied to observe steady, laminar natural convection in nonrectangular enclosures with and without fin where vertical walls insulated, horizontal walls uniformly heated and the fin was placed on horizontal surface. G. Jilani [10] showed a CFD study of laminar steady state flow in a 2D, partial open enclosure with a heat source. Tatsuo Nishimura [11] described a computational analysis of laminar free convection heat transfer in a rectangular cavity. The cavity was horizontally divided into two zones fluid and porous. Yasin Varol [12] studied heat transfer numerically in an inclined trapezoidal enclosure. It was filled with a fluid-saturated porous medium. In his study, he observed the results with varying angles and Rayleigh number. Free convection in inclined square enclosure was numerically studied by Cianfrini [13].

The concept of partially heated walls came for the different behavior of natural convection in partially heated enclosure. Yücel, N. [14] studied free convection in partly heated cavities. For a given cooler size, natural convection characteristics were observed for increasing heater size. On the other hand, for a given heater size, natural convection characteristics were observed for increasing cooler size.

Different types of fluid have been used to analysis the difference. Nano-fluids are also included into these. Oztop[15] studied to examine the free convection in a partially heated rectangular enclosure filled with nanofluids like Cu, Al_2O_3 and TiO_2. Cheikh [16] inspected natural convection in a square cavity heated partially from below. A. S. Dogonchi [17] observed a CFD analysis of natural convection of copper–water-based nanofluid filling a triangular enclosure with semicircular bottom wall.

These studies show the importance of the size and location of the heated portion in natural convection.

Partially heated enclosure is needed to study about natural convection characteristics for the necessity of applications. It has not been so much focused on research field where the effects are significant. So, this research is

* Corresponding author. Tel.: +88-01797197331
E-mail addresses: farhanaislam2098@gmail.com

done to observe the characteristics for different Rayleigh number and heated portion. This current study is for investigating natural convection characteristic for laminar steady flow in a square enclosure with partially heated vertical walls and adiabatic horizontal walls. The study is conducted for two different Rayleigh Number 10^4 and 10^5. Prandlt number is considered 0.71 for all cases. Flow is considered as Newtonian flow. Heated portions were assumed 20%, 40%, 60%, 80%, 100%. Mesh independency has been tested. DNS method is used for the numerical analysis. Temperature contour, Heat transfer co-efficient, local and average Nusselt Number, stream function are studied for the investigation.

2. Methodology

For these simulations, a two-dimensional rectangular enclosure with dimension (Height, H X Length, L) 100 mm X 100 mm, sketched at Fig. 1, was considered which is filled with air. The both horizontal walls are assumed to be adiabatic and both vertical walls heated partially. The heated portions, denoted by l_H, of the both vertical walls were considered to be symmetrical from the midpoint of the walls. For these simulations, heated portions, l_H, was assumed equal to 20%, 40%, 60%, 80% and 100% of the total length of the vertical wall. The remaining portion, except to the heated portion, of vertical walls were also considered adiabatic. The heated potion of the left and right vertical wall was considered to be heated with temperature, T_h =310K and T_c =300K, respectively. For these DNS simulations, it was assumed incompressible, steady flow and two dimensional. Also Boussinesq approximation was adopted.

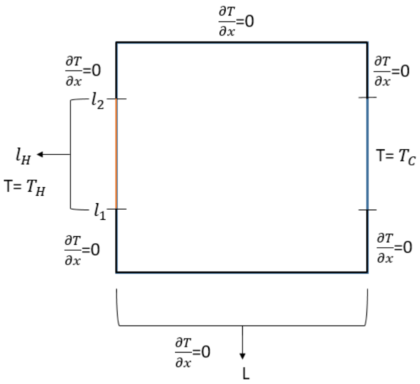

Fig:1: Domain with boundary conditions

The governing equations for this natural convection flow are continuity, momentum and energy equation are given below:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \quad (1)$$

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\vartheta\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \quad (2)$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho}\frac{\partial P}{\partial y}+\vartheta\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+g\beta(T_h-T_c) \quad (3)$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \quad (4)$$

where, u and v are the velocity components in the *X*-direction and *Y*-direction, respectively. Also, β, α and ϑ represent the coefficient of thermal expansion, thermal diffusivity and viscosity respectively.

The appropriate Boundary conditions are:

For lower wall:

$$u=v=0\ \&\ \frac{\partial T}{\partial x}=0 \text{ for } 0\le x\le l\ \&\ y=0 \quad (5)$$

For upper wall:

$$u=v=0\ \&\ \frac{\partial T}{\partial x}=0 \text{ for } 0\le x\le l\ \&\ y=H \quad (6)$$

Left vertical wall:

$$u=v=0\ \&\ T=T_H \text{ for } l_1\le y\le l_2\ \&\ x=0 \quad (7)$$

$$u=v=0\&\frac{\partial T}{\partial x}=0 \text{for} 0\le y\le l_1;$$
$$l_1\le y\le l_2\& x=0 \quad (8)$$

Right vertical wall:

$$u=v=0\ \&\ T=T_c \text{ for } l_1\le y\le l_2\&\ x=L \quad (9)$$

$$u=v=\frac{\partial T}{\partial x}=0 \text{ for} 0\le y\le l_1;$$
$$l_1\le y\le l_2\&\ x=L \quad (10)$$

Commercial software ANSYS Fluent 17 were used to carried out these Direct Numerical Simulations (DNS). In Fluent setup, pressure based solved and semi-implicit method for pressure & velocity linked equations (SIMPLE) was selected. Also, second order upwind scheme is chosen for the spatial discretization of the continuity, momentum and energy equations. The convergence criteria selected for all of the equations is 10^{-10}.

These DNS simulations were carried out for different Rayleigh number, *Ra*, and fixed Prandlt number, *Pr*, which is

$$Ra=\frac{g\beta(T_h-T_c)l^3}{\vartheta^2} \quad (11)$$

$$Pr=\frac{C_p\mu}{K}; \quad (12)$$

Also local Nusselt number, *Nu*, and average Nusselt number, Nu_{ave}, were calculated with the following equations:

$$Nu=\frac{hl}{k} \quad (13)$$

$$Nu_{ave}=\frac{1}{l_H}\int_{l_2}^{l_1} Nu\, dy \quad (14)$$

where, h is convection Heat transfer coefficient and k is thermal conductivity.

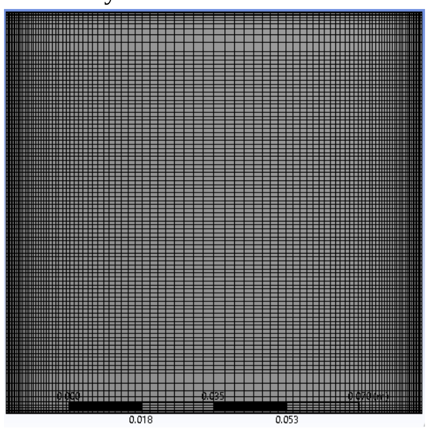

Fig:2: Grid Distribution

Mesh generation is very important aspect for CFD simulation. Rectangular mesh were created by ANSYS meshing software by dividing horizontal and vertical wall. Different types of meshes (coarse, fine), shown at Table 1, were created by changing the number of division. Higher mesh resolution is required near to wall. As a result, uniform growth rate were applied from each wall to center of the cavity. Table 1 demonstrate mesh independency test results for $Ra = 10^4$ with 100% heating. It clearly visible from the table that Nusselt number and heat transfer coefficient change significantly for coarse meshes with element number 100 and 2500, however both properties remain unchanged for fine meshes with element number 10000 and 40000. This indicates meshes with 10000 and 40000 element can produce accurate results. For that reason mesh with 10000 element were consider for the further simulations.

Table:1: Properties for $Ra = 10^4$,100% heated portion on vertical walls:

Elements	Nusselt number	Heat transfer co-efficient
100	2.174	0.522
2500	2.213	0.531
10000	2.244	0.543
40000	2.244	0.543

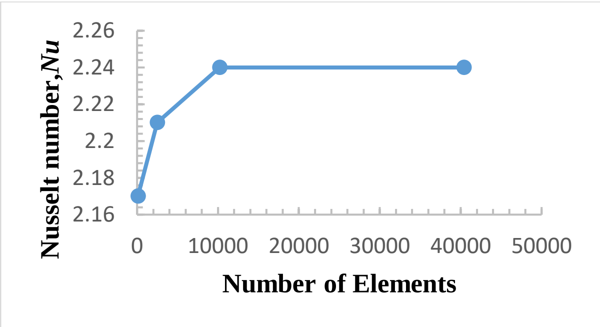

Fig:3: Nusselt Number Vs Number of elements

This numerical results were also validate by comparing with the results obtain by N. C. MARKATOS* and K. A. PERICLEOUS (1983) [8] for natural convection in a square cavity heated uniformly vertical walls and adiabatic horizontal wall for $Ra = 10^4$ & $Ra = 10^5$, shown at Table 2. Results clearly demonstrate that at Nusselt number varies by 1.9% and 2.2% for $Ra = 10^4$ & $Ra = 10^5$, respectively.

Table:2: Comparisons between two solutions

Rayleigh number	Nusselt number (present work)	Nusselt number (N. C. MARKATOS* and K. A. PERICLEOUS (1983))	Δ%
10^4	2.244	2.201	1.9%
10^5	4.529	4.430	2.2%

3. Result and Discussion

Figure 4 depicts the effect of percentage of heated portion on Average Nusselt Number (Nu) at $Ra = 10^4$; $Ra = 10^5$. It is clearly observed from figure that Nu decrease almost linearly with increasing heated portion for both Ra, however Nu have higher value for each heated portion at higher Ra.

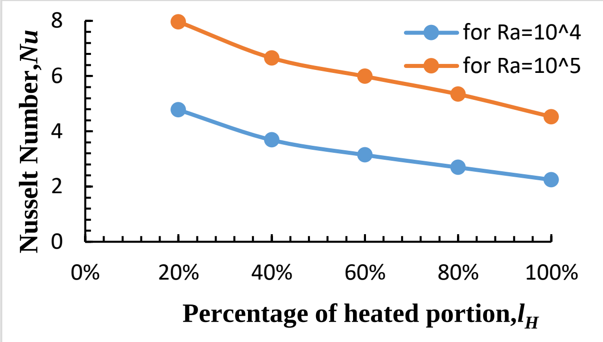

Fig:4: Local Nusselt number variation along the Heated(left) wall

From the Fig. 5, heat transfer co-efficient vs percentage of the heated portion, it is seen that heat transfer co-efficient decreases when the percentage of heated portion increases. Heat transfer co-efficient increases when Rayleigh number increases. Temperature variation along the horizontal center line has been plotted for 40% of heated wall in Figure 6 for both Rayleigh number. It shows that at higher Ra temperaure drop and rise sharply near to heated and coled wall, respectively, and remains constant along the centre. Due to this sharp drop Nusselt number is higer at higher Ra.

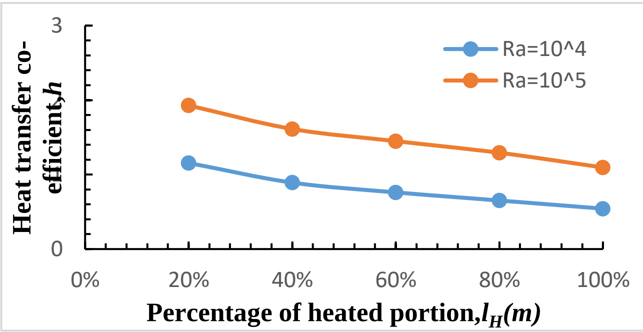

Fig:5: Local Heat transfer co-efficient variation along the heated (left) wall

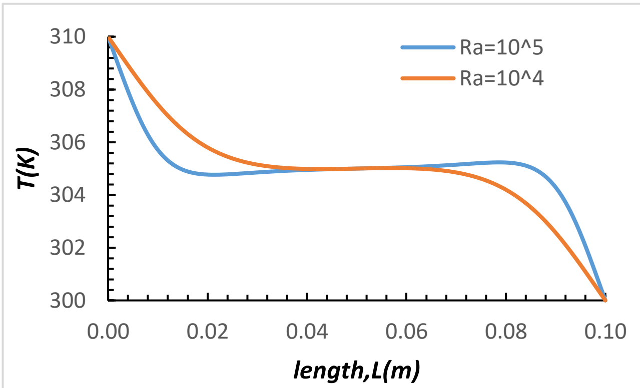

Fig:6: Temperature variation horizontally along the Length for l_H=40% heated wall of different Rayleigh number

Figure:7 demonstrates variation of temperature along the horizontal center line for $Ra = 10^4$ for different portion of heating It is observed from figure that temperature gradient varies a little with the change of heated portion for same Rayleigh number. That means effect of Rayleigh number is much noticeable in temperature gradient than the change in percentage of heated portion. From fig:8: it has been observed for $Ra = 10^5$. The graph is quite different from fig:7 but the change for different percentage of heated portion is so little. Velocity and stream functions have been illustrated for each case in fig:9 & fig:11 respectively. It has been observed that flow pattern does not change much for varying the heated portion. The fluid circulation in the cavity is almost unchanged for five cases. Flow pattern has been changed for Rayleigh number. In fig:9, for $Ra = 10^4$, five cells of oval shape appear. At Ra = 10^5, the center cell near of the enclosure tends to approach diagonally and form a large cell. In fig:11, for $Ra = 10^4$, one cell almost symmetrical with respect to the cavity axes plane (X, Y) appear. At Ra = 10^5, the flow near the middle region of the enclosure becomes more important. That one cell located on the center tend to proceed diagonally and form two cells. Because for higher Rayleigh Number the circular flow separates in the collision point and creates two cells. Isotherms have been illustrated in figure:10. Isotherms have been affected for the variation of heated portion of vertical walls. But little bit change has been noticed for the different Rayleigh number. Because Nusselt number depends on the length of heated portion.

From the Nu Vs l_H graph, a relation has been obtained between them. The relation is non-linear.

For $Ra = 10^4$; $Nu = 2.3827(l_H)^{0.451}$
For $Ra = 10^5$; $Nu = 4.8289(l_H)^{0.329}$

From the h Vs l_H graph, a relation has been obtained between them. The relation is also non-linear.

For $Ra = 10^4$; $h = 0.5768\ (l_H)^{0.0.451}$

For $Ra = 10^5$; $h = 1.1686\ (l_H)^{0.33}$

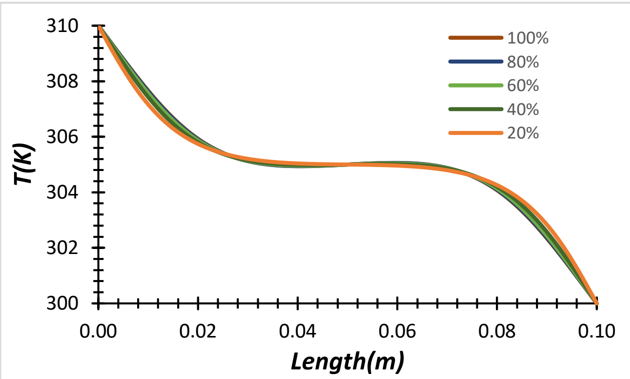

Fig:7: Temperature variation horizontally along the Length of different portion heated wall for Ra=10^4

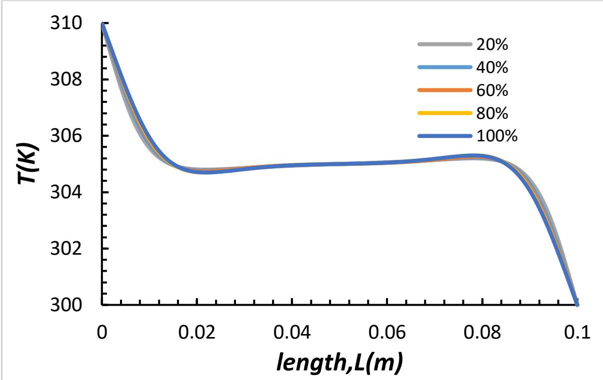

Fig:8: Temperature variation horizontally along the Length of different portion heated wall for Ra=10^5

Table:3: Average Nusselt number and Average Heat transfer co-efficient for different Rayleigh number and different heated portion on vertical walls

Rayleigh Number	Percentage of Heated Portion	Avg. Nusselt Number	Avg. Heat Transfer Co-Efficient
Ra=10^4	20%	4.776	1.156
	40%	3.685	0.892
	60%	3.143	0.761
	80%	2.692	0.652
	100%	2.244	0.543
Ra=10^5	20%	7.966	1.928
	40%	6.656	1.612
	60%	5.989	1.449
	80%	5.343	1.293
	100%	4.529	1.096

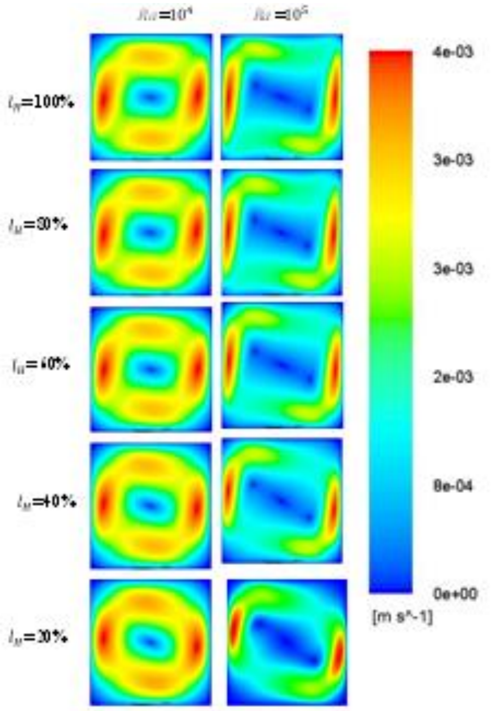

Fig:9: Velocity Contours for different heated portion of different Rayleigh Number

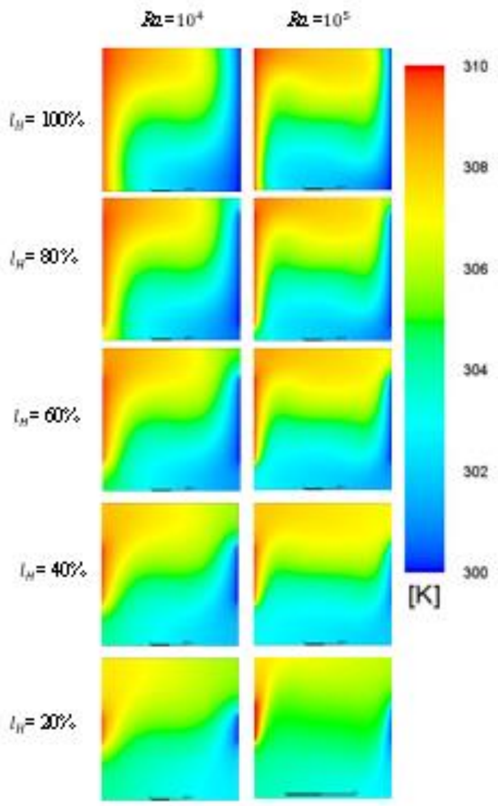

Fig:10: Temperature contours for different heated portion of different Rayleigh number

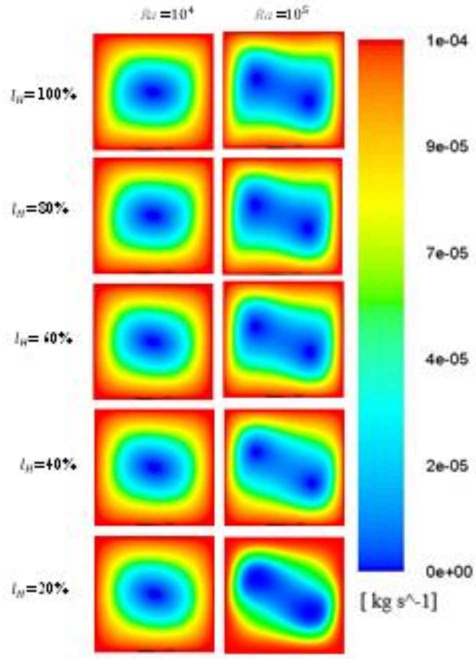

Fig:11: Stream function contours for different heated portion of different Rayleigh number

4.0 Conclusion:

A numerical analysis on a partially heated square enclosure has been presented. DNS method had been used to solve the whole simulations. The flow structure and thermal field are strongly affected by increasing the heated portion and the Rayleigh number. The increasing value of Rayleigh number enhances the heat transfer. It is also observed that Nusselt Number has a proportional relation with Rayleigh number. The isotherms have been shown great influence by different heated portions, however not affected for different Rayleigh number. Besides, velocity and stream function have great impact for varying Rayleigh number. From this study relation between heat transfer co-efficient and heated portion of a specific Rayleigh number has been obtained. Relation between Nusselt number and heated portion has been also obtained for specific Rayleigh number. By observing the contours, natural convection characteristics has been discussed.

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NOMENCLATURE

g	: Gravitational acceleration, m/s^2
k	: Thermal conductivity, W/m.K
L	: Length of the cavity, m
Nu	: Local Nusselt number
Nu_{ave}	: Average Nusselt number
P	: Pressure, N/m^2
α	: Thermal diffusivity, m^2/s
β	: Thermal expansion coefficient,1/K
ϑ	: Kinematic viscosity, m^2/s
Pr	: Prandtl number
Ra	: Rayleigh number
T	: Absolute temperature, K
T_h	: Temperature of the hot wall, K
T_c	: Temperature of the cold wall, K
ρ	: Density,kg/m^3
μ	: Dynamic viscosity, kg.m/s^s
C_p	: Specific heat capacity, kJ/kg.K
h	: Convection heat transfer coefficient, W/m^{2}K
l_H	: Heated portion, m