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Natural convection heat transfer around horizontal tube in vertical slot

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Abstract

An experimental and numerical study of natural convection around horizontal tube placed between two isothermal walls of different temperature is presented. Two experimental set-ups are built for visualizing the flow and to measure the temperature around the tube and walls confined in the close cavity. An FEM computer code has been applied for analysing the influence of various parameters on the flow structure and heat transfer. The fluid in slot is air ($Pr = 0.71$). The measurement are taken for Rayleigh number: for slot $Ra_s = 2 \times 10^7$–1×10^9; for tube $Ra_r = 6 \times 10^2$–9×10^2. On analysing the heat transfer from the tube, it has been found that the intensity decreased and increased unexpectedly. The explanation can be given after visualisation. It is observed that the hot air near the heated wall aspirated the layers of hot air from the horizontal tube. The uplift pressure and the tendency of the system to reach balance causes the air layers of the similar temperature and density to merge. However, this effect (similar to the movement of the hot air in a chimney) also found to be dependent on the intensity of the heat transfer.

1. Introduction

In the electronic equipment, solar thermal energy storage system and small refrigeration devices, we want good heat transfer without acoustic and electromagnetic noise. This is assured with natural convection heat transfer. The problem is more complicated, because usually warm elements remain in air, in semi-enclosed space. Very often we meet some elements between two parallel confining walls, usually with different temperatures. Thus, most of scientists [1,4,5] have discovered that the heat transfer from heated internal elements in the slot increases as a result of the 'chimney' effect. As a result of 'chimney' effect heat transfer from internal slot tube enhanced up to 50–60% more than free-space positioning. In case when the slot or vertical channel has been thinner and longer the convection heat transfer coefficient has increased up to 2.5 times than in free-space. It will be important to confirm the fact when and what kind geometrical configuration of the natural convection heat transfer is the best. The more important in this case is obtained results of influence boundaries and surfaces which are limited heat transfer area. Many of the scientists take trial of investigating different heat transfer system. Some of them had investigated almost the same configuration: tube between two confining walls as Farouk and Guceri [5], but in this case the walls were adiabatic and the results were obtained by numerical investigation. A few years later Farouk et al. [4] showed their experimental work on the same problem. The important fact of that was the influence walls to heat transfer around horizontal tube is negligible from aspect ratio (B/D — space between vertical walls to tube diameter) is higher than 12. When the ratio is equal to 8 we observed that the wall influence strongly decreases and eventually completely disappears. Naylor and Tarasuk [1] made full investigation about vertical heated plate

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Nomenclature

a	thermal diffusivity (m^2/s)
A	surface area (m^2)
B	slot width (m)
B_1, B_2	distance between tube and hot, cold plate respectively (m)
D	tube diameter (m)
g	gravitational acceleration (m^2/s)
H_1	height of the tube placed in the slot measured from slot inlet (m)
H_C	slot height (m)
H_s	height of the thermocouple probe placed in the slot measured from slot inlet (m)
L_C	slot depth (m)
L	tube length (m)
p	pressure (Pa)
T	temperature (K)
T_C	cold plate temperature (K)
T_E	environmental temperature (K)
T_H	hot plate temperature (K)
T_r	surface tube temperature (K)
$T_\in$	temperature inflow on tube (K)
ΔT	temperature difference (K)
t	time (s)
I	current (A)
U	voltage (V)
u, v	fluid velocity (m/s)
V	volume (m^3)
Θ	non-dimensional temperature
Θ_E	non-dimensional environmental temperature
$\Theta_\in$	non-dimensional inflow temperature
Pr	Prandtl number
Gr	Grashof number
Ra_r	Rayleigh number for tube
Ra_s	Rayleigh number for slot (walls)
Re	Reynolds number
Nu_r	Nusselt number for tube
Nu_s	Nusselt number for slot
q_C	heat flux on lateral tube surface (W/m^2)
q_s	heat flux on wall (W/m^2)
q_{conv}	convection heat flux on lateral tube surface (W/m^2)
q_{rad}	radiation heat flux on lateral tube surface (W/m^2)
q_{end}	heat flux lost by end of tube (W/m^2)
h_r	heat transfer coefficient for tube (W/m^2 K)
h_s	heat transfer coefficient for wall (W/m^2 K)
β	coefficient of thermal expansion (K^{-1})
ε_t	emissivity
λ	thermal conductivity (W/m K)
μ	dynamic viscosity (kg m/s)
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
σ	black body constant (W/m^2 K)
ψ	stream function
φ_E	environmental humidity (%)

placed between two isothermal vertical walls (vertical slot). In their investigation three heated elements (plate and walls) had equal temperature. They used two different methods to compare results: (1) numerical solution based on the FIDAP code and experimental measurements with visualisation method (Mach–Zehnder interferometer) and (2) thermocouple temperature measure. In this paper an investigation about heated tube inside vertical slot, but the temperature of walls and tube are different, is shown.

1.1. Experimental apparatus

At the beginning of our experiment and numerical study the horizontal tube could have regular different temperature, occurring predominantly around horizontal tube located in a vertical slot, heated from one side and cooled from the other.

For full investigation we had to perform all experimental and numerical techniques. Experimental studies were performed on two set-ups where one of them used flow visualisation method, the second one used temperature measurement method with thermocouple. All the results obtained using experimental methods were compared successfully to numerical results for temperature field performed by FEM commercial code NISA 386 available from EMRC.

In this model the fluid which filled the vertical slot was air (Fig. 1). All investigations were carried out for Prandtl number, $Pr = 0.71$. After the period of completing tools for measurements as well as the test measurements, we decided to do research of only those cases in which the temperature of horizontal tube was equal or almost equal to temperature of the hot wall. Because the geometry of the model was complicated (in case of natural convection), we decided to refer Rayleigh number for slot (Ra_s) and for tube (Ra_r) respectively. All results presented were obtained for Rayleigh numbers: for tube $Ra_r = 6 \times 10^2$–9×10^2 and for slot $Ra_s = 2 \times 10^7$–1×10^9. Numbers were counted according to Eqs. (1)–(5). The cross section of the model is shown in Fig. 2. When the experiment lasted the temperature of elements was kept in such a way that the heat transfer on the depth of the slot was neg-

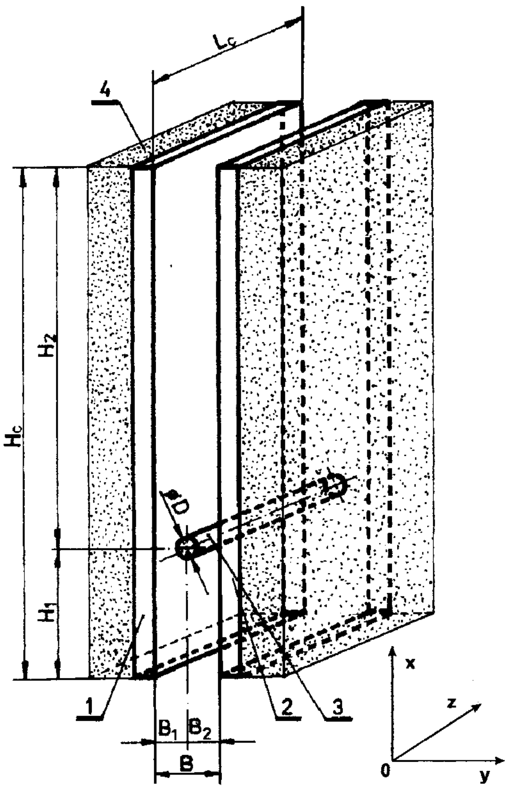

Fig. 1. Schematic diagram of the horizontal tube between two vertical walls, 1, 2 — vertical isothermal plates, 3 — horizontal heated tube, 4 — isolation.

ligible. Becuase of this we could solve the problem in two dimensions [6].

- equations for slot and tube

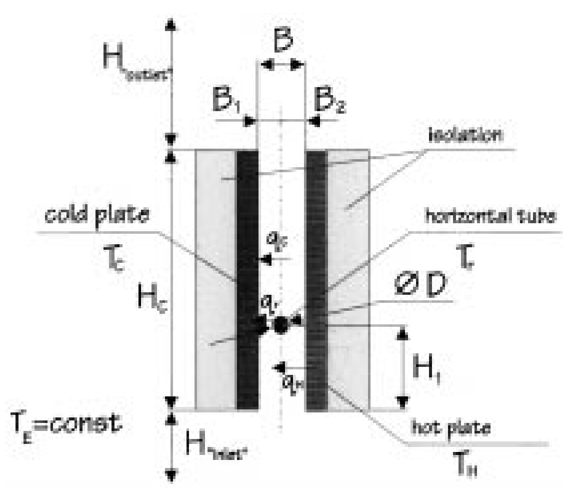

Fig. 2. Two-dimensional model.

$$Pr = \frac{\nu}{a}; \quad \bar{H} = \frac{H_1}{H_C}; \quad \bar{B} = \frac{B_1}{D}; \quad \Theta = \frac{T - T_C}{T_H - T_C}; \quad (1)$$

- equations for slot (walls):

$$Gr_s = \frac{g\beta\Delta T_s H_C^3}{\nu^2}; \quad \Delta T_s = T_H - T_C;$$

$$Nu_s = \frac{h_s H_C}{\lambda}; \quad (2)$$

$$Ra_s = Gr_s \times Pr\left(\frac{H_C}{B}\right); \quad (3)$$

- equations for tube:

$$Gr_r = \frac{g\beta\Delta T_r D^3}{\nu^2}; \quad Ra_r = Gr_r \times Pr\left(\frac{B_1}{D}\right); \quad (4)$$

$$Nu_r = \frac{h_r D}{\lambda}; \quad (5)$$

One part of our experimental study was made on the set-up with thermocouples for temperature measure. The main elements of this set-up were two vertical parallel plates ($1200 \times 560 \times 23$ mm) which were adopted from evaporators from cooling system on ships made of aluminium (compact type). Some of them were heated electrically, some heated or cooled by water from ultra-thermosystem. The fields of temperature for both walls were registered during all measurements. Thermocouples type T were used to perform this. The horizontal heated tube was made of copper pipe painted black and had electrical heater inside (scheme of tube-cross section — see Fig. 3).

Investigation was carried out with different tubes of diameters: 6, 8, 10, 12, 16 and 20 mm. All set-ups were in isothermal cold room (main dimensions are shown in Fig. 2). In order to assess temperature field of air in vertical slot special thermocouple probe type T was constructed (Fig. 4). This probe could move in 'x', 'y', 'z' directions.

The leaders were made of drawing boards' leaders. Probe movement and placement was remote controlled outside the cold room. It was important to investigate how radiation from the probe and walls effect the results of measured temperature. The probe was without coat (no coated) and therefore the following assumption seems to be plausible: the sum of convection heat transfer from the probe is equal to the sum of radiation heat transfer which can be as follows:

$$h(T_{air} - T) = \varepsilon_t \times \sigma\left(T^4 - T_H^4\right) + \varepsilon_t \times \sigma\left(T^4 - T_C^4\right) \quad (6)$$

where h is the heat transfer coefficient over thermocouple welding, ε_t is the emissivity thermocouple weld-

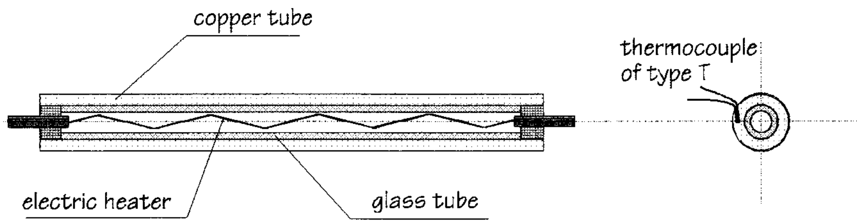

Fig. 3. Horizontal tube.

ing ($\varepsilon_t = 0.04$–0.06 for thermocouple welding according Ref. [3]), σ is the black body radiation constant; ($\sigma = 5.67\text{e}-8$ W/m^2 K), T is the temperature of thermocouple, and T_{air} is the temperature of inlet air to the thermocouple welding.

Because the measured temperature differences were small, we may take approximately:

$$\Delta T_{\text{r}} = (T - T_{\text{air}}) \approx -\frac{\varepsilon_t \sigma}{\alpha} T^3 \left[(T - T_{\text{H}}) + (T - T_{\text{C}}) \right] \quad (7)$$

where ΔT_{r} is the error in measured temperature.

Therefore, maximum error was approximately 0.5°C, which can be neglected. Since the rest of measuring equipment, which for example consist digital voltmeter, has the same error and the sum of errors from all equipment were added.

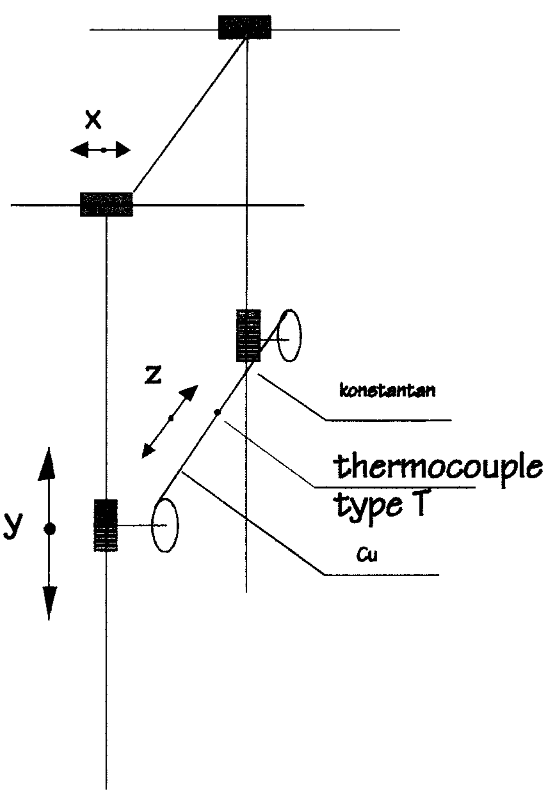

Fig. 4. Thermocouple probe.

The experimental study was completed by flow visualisation in which Schlieren method knitted with computer system picture transformation. The experimental apparatus was 'Schlieren-Aufnahmegerat 80' produced by the Jena, Germany which is based on Toepler method. It is well known that the Schlieren method does not give very good results of measurement in cases where the change of gradient of density in fluid was too small. For this purpose, we tried to use certain specific of picture transformation from bit map format to vector format. 'Aldus Photo Styler 2.0SE' and function trace contour was used to achieve this aim. These experiments required observation of the flow structure. The pictures were registered by computer with a 'frame grabber' card produced by the FAST, Germany.

2. Experimental results

The natural convection experiments performed on horizontal tube placed in the slot between two isothermal vertical, parallel walls indicate that decrease in intensity of heat transfer, approximating the value of $B_1/D = 2.5$ coefficient, where maximum heat transfer coefficient h (Fig. 5) for all investigated cases was observed. The increase in heat transfer intensity for B_1/D coefficient below minimum heat transfer coefficient h results from the increasing influence of diffusion stream over the convection stream. The most effective transfer takes place at coefficient value approximately 3.5. The shape of curve (Fig. 6.) obtained in this experiment remains in accordance with the shape of the curve presented by Naylor and Tarasuk [1] for a vertical plate placed in a vertical slot. Additionally, qualitative investigation of this phenomena confirm the fact of change in the character of the stream between the plates for such small width of the slot.

In order to establish the convection heat transfer

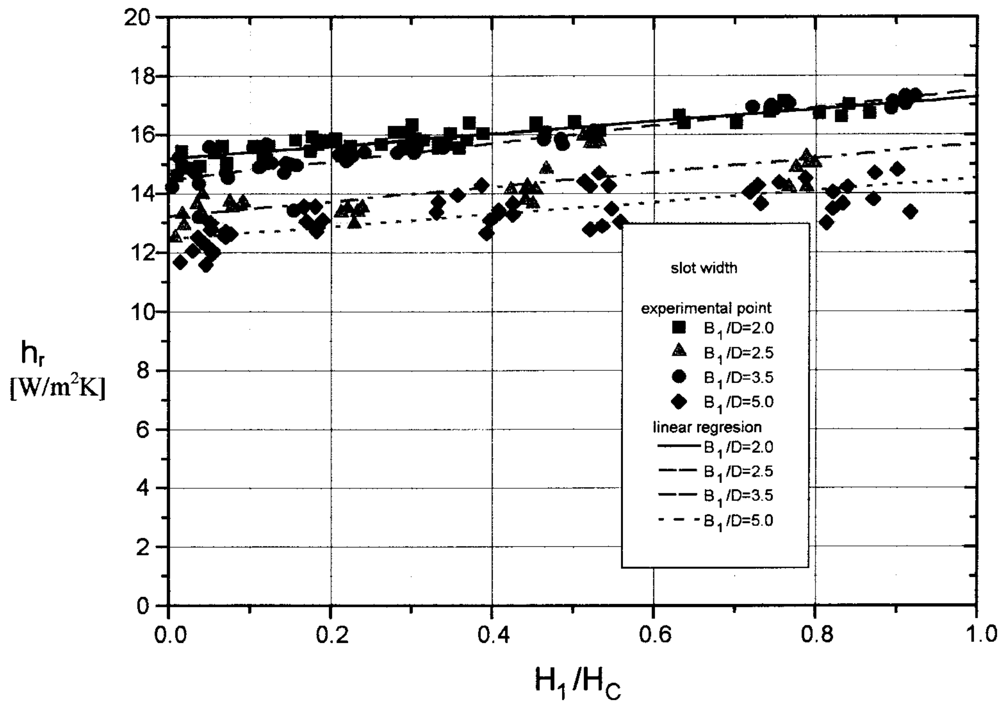

Fig. 5. Dependence of heat transfer coefficient and slot width and height of tube in slot.

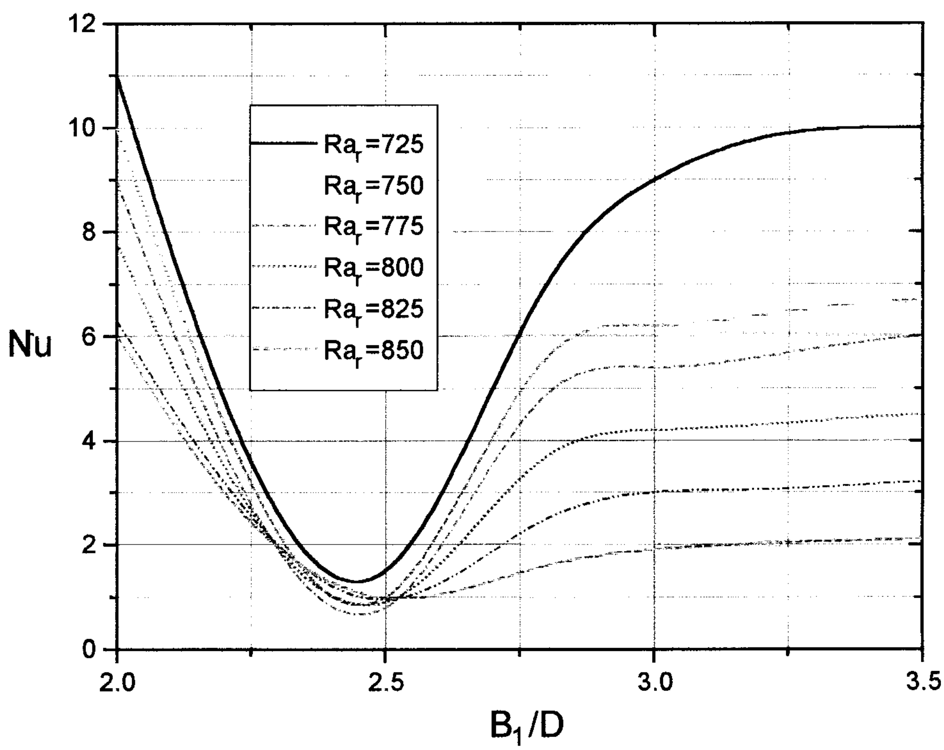

Fig. 6. Change of Nusselt number for tube as width of slot function.

coefficient h_r from the tube the following equation was used:

$$q_c = \frac{U \times I}{A}; \quad q_c = q_{conv} + q_{rad} + q_{end}; \quad h_r = \frac{q_{conv}}{\Delta T_r} \tag{8}$$

where $\Delta T_r = T_r - T_\epsilon$. After a series of measurements, the effect of the slot width, its temperature, as well as the height, on which the tube was placed on heat transfer coefficient, from tube was detected. Because of a great number of parameters affecting heat transfer in this case, the following results for the slot $B_1/D = 2.0$; 2.5; 3.0; 5.0 are presented, where measurements were carried out for $T_H = T_r$, and for steady temperature in the cold room Θ_E. In all measurements B_1 is equal to $B/2$. For narrow slots $B_1/D = 2.0$; 2.5, at the inlet of the slot the temperature oscillated below and above the value of the temperature in the cold room environment. Cold wall definitely has a substantial effect on the lowering of the inlet temperature.

The increase of air temperature in the slot in the entry section exerts a substantial influence on inflow temperature on the tube placed inside. By inflow temperature we underset the temperature measured below the tube at a distance equal to five dimensions. This distance has been estimated experimentally, which guaranteed no influence of tube temperature on the measured air temperature. The analysis of the results used to state that the intensity of heat transfer primarily depends on the inflow temperature on the tube, and to a smaller extent, on the mutual effect of the boundary layers on one another. The width of the slot has the greatest effect on the inflow temperature, except for the inlet fragment in which the inlet air temperature of the hot tube and the hot plate were the same, 'suction' of the stream rising from the tube by the stream from the hot plate took place. It is well visible on further presented Schlieren pictures.

The optimal conditions of heat transfer for a single tube in the investigated space remains within range: $3.0 < B_1/D < 5.0$. In case of wider slots $B_1/D > 8.0$ the heat transfer round the tube proceeds in the same way as in unlimited space. For narrow slots, diffusion stream dominates in heat transfer over undeveloped free convection. The increase of the slot width results in decrease of the diffusion stream and free convection development. Further widening of the slot causes the increase of the local air speeds. This increase in the air speed results in the effect of 'suction' described above, and the increase in the intensity of heat transfer.

The most important phenomenon observed during the experiments is the structure of the air stream around the tube placed inside the slot as it can be seen in the figures, hot air layers near hot elements are 'glued' to each other. Thus, we can suppose that this effect of joining of the layers will result in local increase of the air stream speed. The effect of 'joining' of air layers is most visible for the width of the slot for which the increase in heat transfer was stated after quantitative measurements. Fig. 7 presents a tube and two horizontal tubes (one above the other), placed in horizontal slot. The stream of heat given by the tubes were identical.

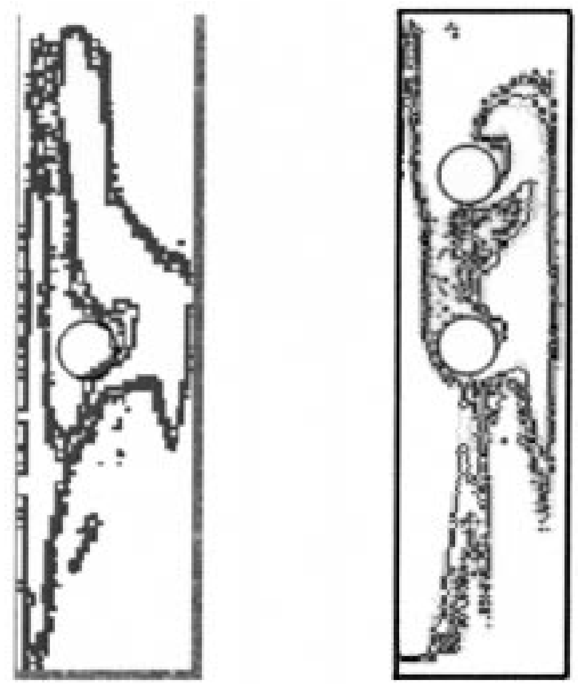

Fig. 7. One and two tubes in the slot. $D = 6$ mm; $B = 35$ mm; $Ra_r = 9 \times 10^2$; $Ra_s = 1 \times 10^9$; (hot plate on left side at the Figure, cold on the right) picture after computer's transformation (tracing).

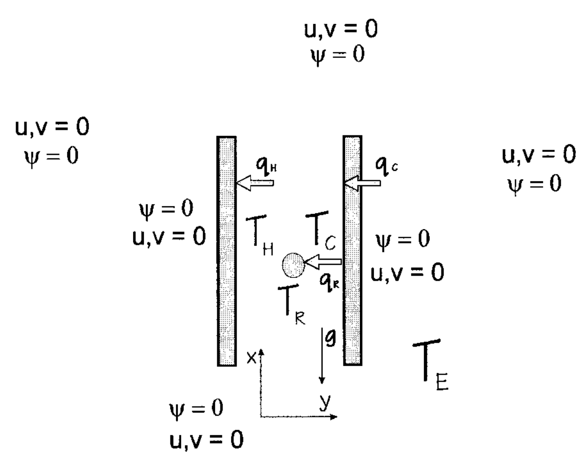

Fig. 8. Boundary condition for numerical model.

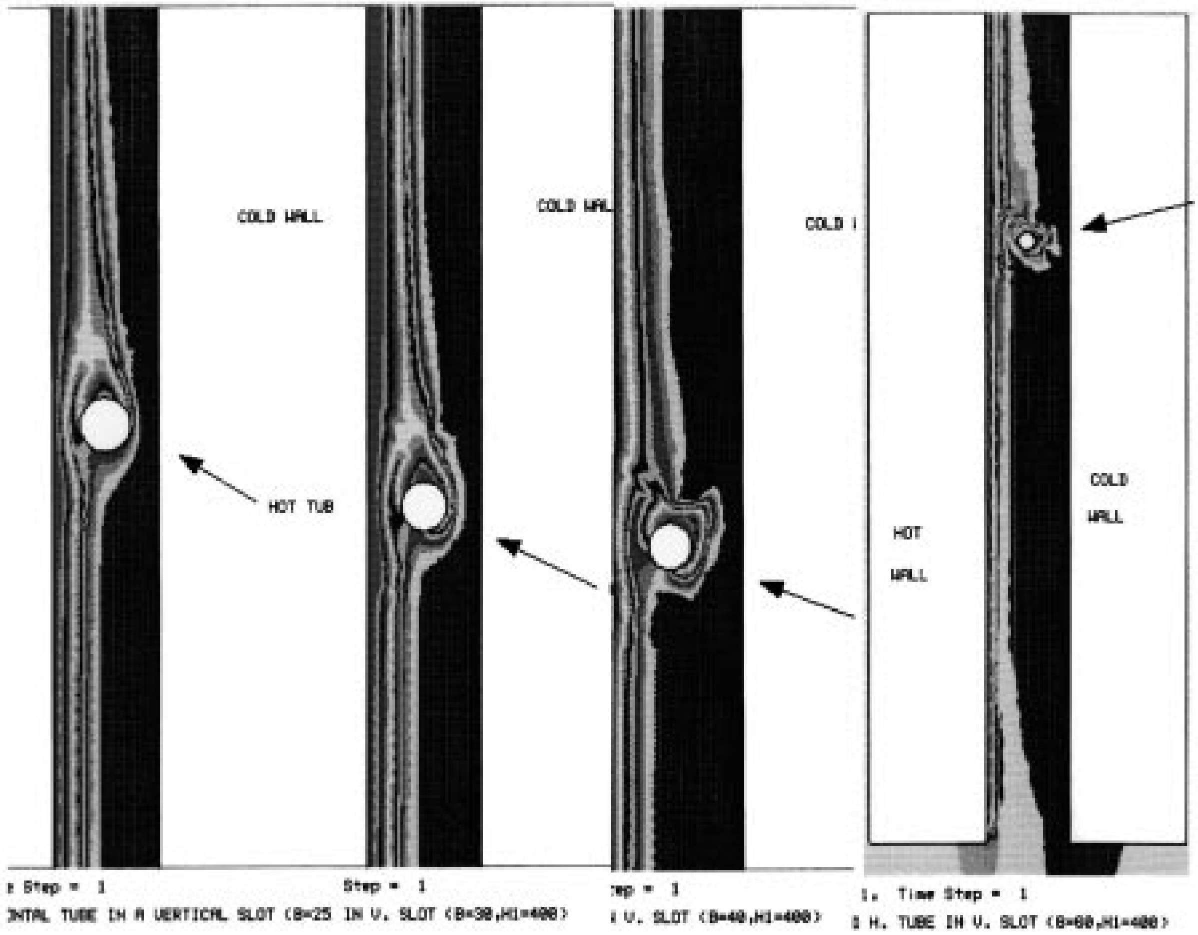

Fig. 9. Field of temperature round horizontal tube in the slot, from left: $B_1/D = 1.25$, 1.5, 2.0, 3.0.

2.1. Numerical analysis

In numerical analysis of the problem we adopted simplifying assumptions:

- two-dimensional model was analysed because of negligible effect of heat transfer on the depth of the slot (according to earlier experiments);
- Boussinesque approximation was used;
- viscous dissipation can be omitted (due to small speed of air);
- steady model was investigated;
- stability of thermophysical properties;
- incompressibility of the fluid;

This was provided for model which is shown in Fig. 8. In this figure the boundary condition and all countable parameters are signed.

For numerical analysis commercial code NISA386 [2] was used. This code is based on Finite Element Method with Petrov–Galerkin algorithm.

For comparison of experimental results numerical calculations for horizontal tube in the slot for coefficient $B_1/D = 2.0$, 2.5, 3.0 was made. A comparison of the average values of heat transfer coefficient compared with the experimental results are shown. Relative error between experimental results and results of numerical counting is approx. 20%. Numerical analysis did not give the same strong decline in heat transfer intensity (for $B_1/D = 2.5$) as in the experiment but the character of changes was saved.

Figs. 9 and 10 present fields of temperature around horizontal tube in the slot, which was placed on high ratio $H_1/H_C = 0.33$. In the pictures from left to right it was shown how the temperature field was changed according to increasing width of slot coefficient B_1/D from 1.25 up to 7.5. All presented parts of the pictures show results of counting for the same thermal conditions. The aim was to present the influence of width of the slot on heat transfer intensity. For width coefficient $B_1/D < 2.0$ the heat transfer proceeded mainly

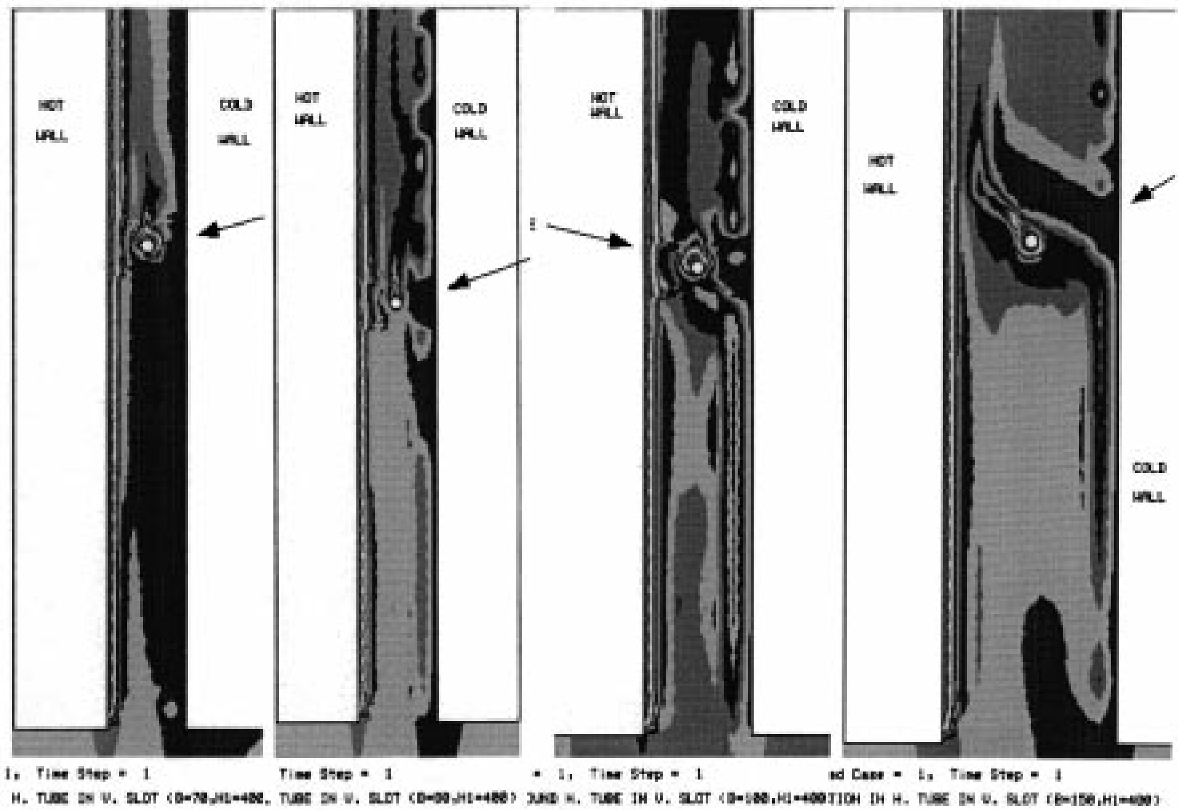

Fig. 10. Field of temperature round horizontal tube in the slot, from left: $B_1/D = 3.5$, 4.0, 5.0, 7.5.

by diffusion stream. The hot stream layer from the tube is strongly 'glued' to stream layer from the hot wall. In this case the convection movement cannot develop (see Table 1).

For coefficient $B_1/D = 2.0$ the influence of diffusion stream decreases and then convection movements are accelerated. Up to width of slot for coefficient $B_1/D = 3.0$ we can exactly mark out irregular temperature field which results from development of convection stream in heat transfer. In this point it is characteristic that the hot layers from the tube and the wall are still glued. The first 'ungluing' of layers is well visible in part of picture for $B_1/D = 3.5$ and for $B_1/D = 4.0$ there are not regular layers connected ('glued'). For bigger value of the coefficient in question the character of flow becomes more irregular and difficult to interpret. However, for higher coefficient (5.0 and 7.5) we can still see mutual influence of the tube and the walls.

The numerical analysis which was done for experimental cases confirms the fact that heat transfer intensity around horizontal tube placed between two isothermal vertical walls for width slot coefficient $B_1/D \approx 2.0$ decreases heat transfer coefficient value in numerical counting is smaller than in experimental results. This is probably the result of measurement errors. All measured parameters are average on length or surface.

Table 1

B_1/D	H_1/H_C	$h_{\mathrm{numerical}}$ (W/m^2 K)	$h_{\mathrm{experimental}}$ (W/m^2 K)
2.0	0.05	11.15	15.2
2.0	0.16	12.66	15.5
2.0	0.33	12.70	16.3
2.5	0.16	11.41	13.4
2.5	0.33	11.25	13.8
3.3	0.05	11.33	15.1
3.3	0.33	12.75	15.3
5.0	0.05	12.28	12.6

3. Conclusions

Summing the obtained results we state that:

1. For each slot the entry region fragment (reaching the height $H_1/H_C = 0.2$), where temperature stabilisation takes place, is identified. The influence of height at which the horizontal tube is placed (where

the width of the slot is insignificant) determines the inflow temperature on the tube. It is necessary to define the so-called inflow temperature in the analysis of the problem, because geometrical position of the tube and the slot affects the structure of the flow.

2. For the ratio $B_1/D < 2.0$ dominating impact of heat transfer is detected. In such case convection movements from isothermal walls are inhibited by the horizontal tube placed in the slot. The increase of this ratio above the quoted value causes a decrease in diffusion intensity, and development of convection.
3. The optimal value of heat transfer intensity is obtained for the horizontal tube, placed between two isothermal walls, for B_1/D ratio between 3.0 and 5.0.
4. At tube temperature close to the ‘hot’ wall temperature, we observed the phenomenon of ‘suction’ of hot air, rising from the horizontal tube, by warm layer of the gas rising from the ‘hot’ wall. This effect of ‘gluing’ of the layers is characteristic of the range in which the optimal heat transfer intensity takes place.

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