

THE BEHAVIOR OF THE FLOW IN ROOMS NEAR WALLS - MEASUREMENTS AND COMPUTATIONS

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ABSTRACT

The purpose of the presented investigation is the comparison between measured data of the laminar and turbulent mixed convection and their approximation by wall functions.

New wall functions were implemented in a FVM-research-code using unstructured grids, which was developed by the author. Numerical results are compared with a turbulent closed cavity flow.

KEYWORDS

Boundary layer, Convective heat transfer, Forced convection, Natural convection

EXPERIMENTAL APPARATUS

An outline of the experimental apparatus is shown in figure 1. The heated surface consists of a 2 m high, 1 m wide and 20 mm thick aluminium plate.

The plate is heated from the backside by 14 horizontal mounted electrical heating tubes (1000 mm wide, 6 mm diameter). The 14 heaters were divided into 7 units. A uniform temperature distribution was obtained by controlling the voltage of each unit.

The inclination angle α can be varied from 0° to 360° .

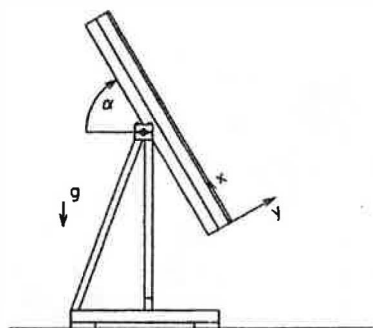
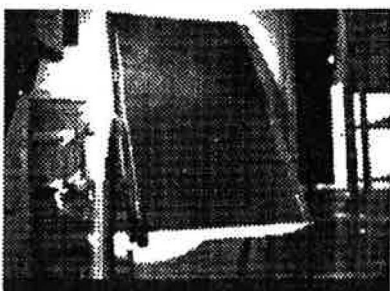


Figure 1 Experimental apparatus

MEAN VELOCITY PROFILES

Experiments in the presented work were carried out at a surface temperature of 80°C . The fluid temperature in the room was about 27°C .

Results for the mean velocity profiles in the boundary layer are shown in figure 2. The inclination in this case is $\alpha = 90^\circ$ (vertical plate). The laminar boundary layer was obtained in the range $x < 0.7\text{m}$, (Rayleigh number $< 2 \cdot 10^9$).

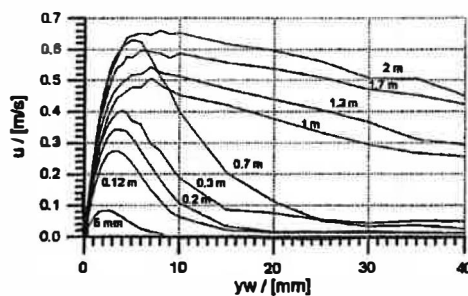


Figure 2 Mean velocity profiles
(vertical plate)

Structured eddies exist between $0.7\text{m} < x < 1\text{m}$, while the maximum velocity decreases by about 20%. A turbulent boundary layer was observed in the range of $x > 1\text{m}$.

Figure 3 shows a typical One-dimensional energy spectrum in the turbulent region. The energy decreases in the range of 1Hz to 5Hz. The energy spectrum in this region can be expressed as $e(f) \sim f^{-1.6}$.

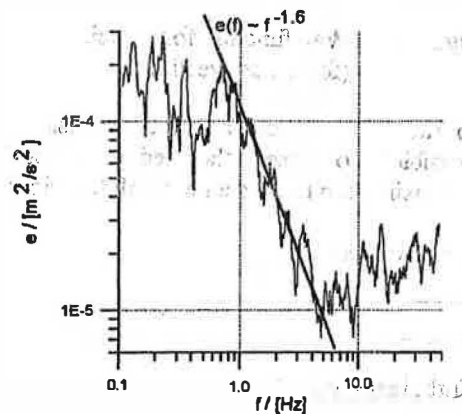


Figure 3 One-dimensional energy spectrum (vertical plate)

A further experimental study examined the air flow at the horizontal plate (inclination angle $= 0^\circ$).

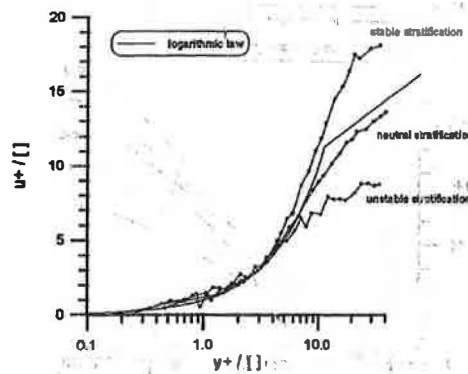


Figure 4 Dimensionless velocity distribution as a function of the dimensionless wall distance (horizontal plate)

The results for the velocity are shown in normalized coordinates in figure 4 for various stratifications.

Results of different inclinations are shown in figure 5. The measuring point was $x = 1.7\text{m}$. The boundary layer thickness decreases in the case of stable inclination ($90^\circ < \alpha < 180^\circ$). The maximum velocity in the labile regime ($0^\circ < \alpha < 90^\circ$) is smaller than in the stable regime ($\alpha > 90^\circ$).

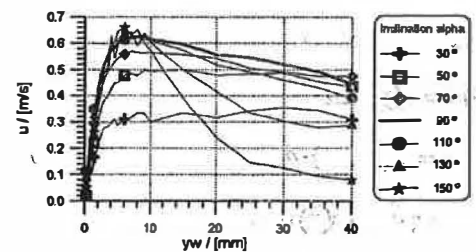


Figure 5 Mean velocity profiles

WALL FUNCTIONS

An analytical investigation of traditional wall functions shows that they are not able to describe the velocity profile near a heated wall exactly.

They are based on a constant shear stress in the boundary layer. This is not sufficient for flows near heated walls. An extension of the model including the influence of temperature has to be provided. Therefore calculations at the wall were performed which take into consideration buoyancy effects near heated walls.

The shear stress is not constant in this case $\left(\frac{Fr1}{dA} \neq \frac{Fr2}{dA} \right)$. The reason for this is the buoyancy force (Fr) added to the flow field in the control volume (figure 6).

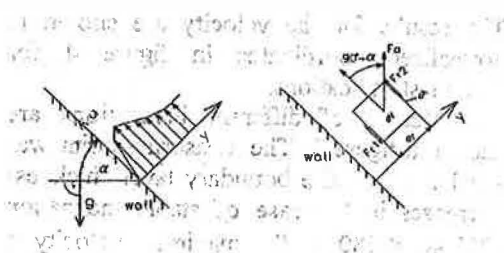


Figure 6 Momenta at a control volume near a heated wall

The momenta are

$$Fr1 = \tau(y) \cdot dA \quad (1)$$

$$Fr2 = \tau(y+dy) \cdot dA \quad (2)$$

$$Fa = g \cdot (\rho - \rho(y)) \cdot dV \quad (3)$$

With the equilibrium of forces

$$-Fr1 + Fa \cdot \cos(90^\circ - \alpha) + Fr2 = 0 \quad (4)$$

the shear stress gradient is calculated.

$$\frac{d\tau}{dy} = g \cdot (\rho(y) - \rho) \cdot \sin(\alpha) \quad (5)$$

The shear stress gives the fundamental change in the wall calculation procedure.

The next step is to solve the equation

$$\tau(y) = \rho \nu_{\text{lam}} \frac{dU}{dy} + \rho \nu_{\text{turb}} \frac{dU}{dy} \quad (6)$$

with an eddy viscosity approximation for ν_{turb} [Albring, 1981]

$$\nu_{\text{turb}} = \nu_{\text{lam}} \frac{Re}{R^2} \quad (7)$$

The numerical results of the wall calculation are presented in figure 7. If the temperature of the wall is equal the temperature in the room ($T_w = T_R$), the shear stress is constant and the calculated velocity profile ($R=6.25$) is similar with the wall function of Yuan [1992] (figure 7).

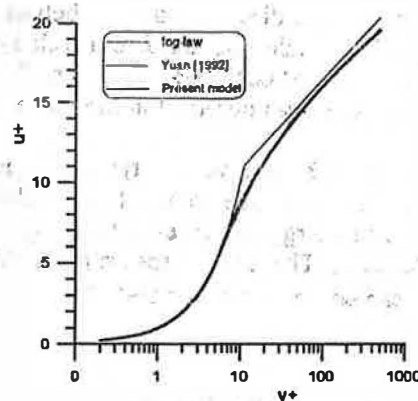


Figure 7 Wall function for velocity (forced convection)

In the case of constant shear stress it is possible to use the eddy viscosity approximation to give a new wall function:

Laminar layer:

$$u^+ = y^+ \quad y^+ \leq R \quad (8)$$

Turbulent layer:

$$u^+ = R \sqrt{2 \ln \left(\frac{y^+}{R} \right) + 1} \quad y^+ > R \quad (9)$$

The advantage of this method is based on one continuous function in the turbulent region (figure 8).

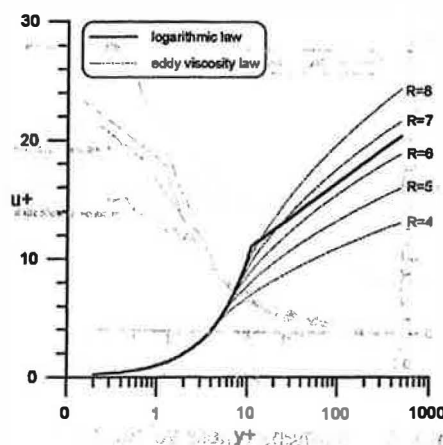


Figure 8 Wall function for velocity (forced convection)

A large temperature difference between the wall and the room increases the velocity gradient near the wall. The heat transfer increases in the same manner.

It can be seen from figure 9 that the velocity profile obtained by the new wall function fits well to the measured data. Traditional wall functions are not applicable in the range from 0 to 100 mm.

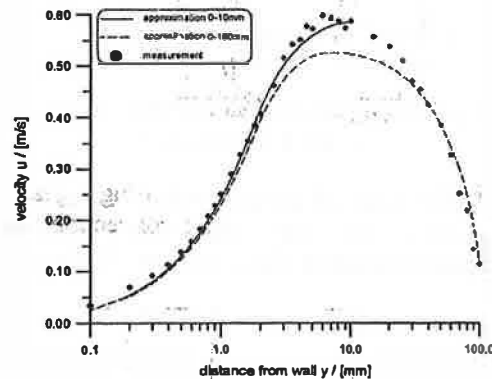


Figure 9 Wall function for velocity (natural convection)

The next step is the determination of the constant R . In [Strehle, 1979] numerical data are shown which can be approximated by the function:

$$R = R_{\text{neutral}} \cdot e^{(K_s \cdot Ri)} \quad (10)$$

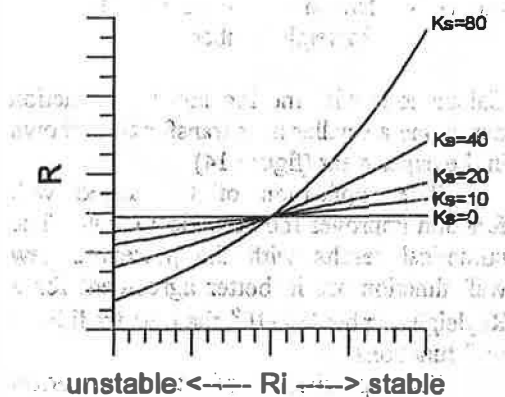


Figure 10 The constant R as function of the Richardson number

The neutral, stable and unstable stratifications results in figure 11 were calculated with the function (10). The stable stratification has a 10mm thick laminar sublayer. The thickness of the unstable sublayer is 2mm.

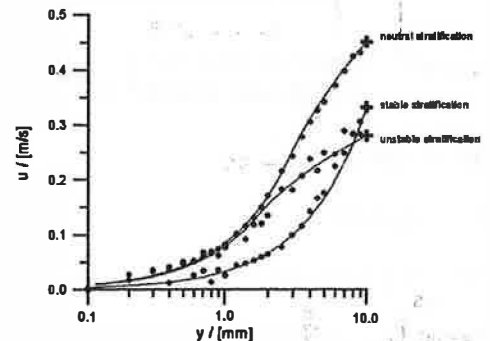


Figure 11 Velocity distribution as a function of the wall distance (horizontal plate)

NUMERICAL RESULTS

The FVM-code uses a standard $k-\epsilon$ -turbulence model [Rodi, 1980]. The new wall functions were applied to a turbulent closed cavity flow [Choesewright et al, 1986].

The geometry and the unstructured grid is shown in figure 12. In this present work a coarse grid is applied to compute the closed cavity flow (minimum edge-length: 30 mm).

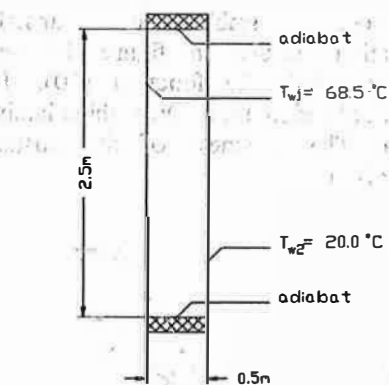


Figure 12 Geometry and grid of the turbulent closed cavity flow

The numerical results are based on three models: log law, forced convection law, mixed convection law. These models are compared to the experimental data given by Cheesewright et al.

Figure 13 shows the vertical velocity calculated using the log law and the mixed convection law.

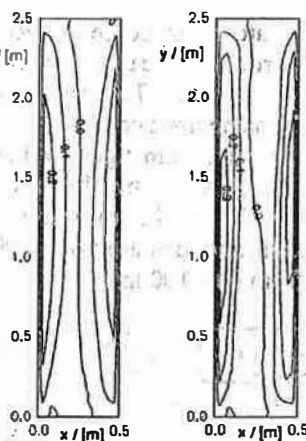


Figure 13 Vertical velocity using log law (left) and mixed convection law (right)

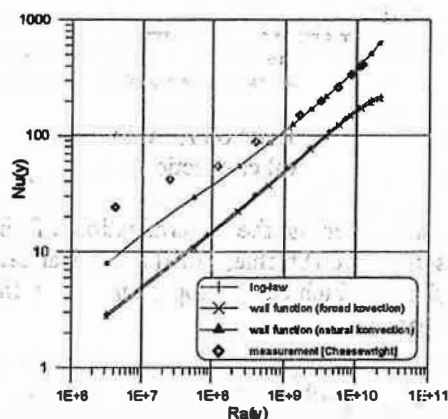


Figure 14 Nusselt number as a function of Rayleigh number

Calculations with the log law wall function determine a smaller heat transfer than shown in the experiment (figure 14).

The application of the tested wall function improves the numerical results. The numerical results with the presented new wall function are in better agreement for a Rayleigh number $> 1 \cdot 10^9$ than the traditional wall functions.

The precision of the numerical simulation increases in case of a finer grid [Borth, 1993, Neitzke, 1994].

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