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THE INFLUENCE OF CONVECTION ON WEIGHING

C.H. MASSEN[1], M.J.F.P. PLUIJM[1], J.T.H. LAMMERS[2], E. ROBENS[3],
J.A. POULIS[1]
[1]Department of Physics, University of Technology, Eindhoven
[2]Department of Building Engineering, University of Technology, Eindhoven
[3]Institut für Chemie, Joh.Gutenberg-Universität, Mainz, W-Germany

ABSTRACT

Due to temperature differences along the wall of a balance case, convective currents may occur which can influence the weighing. Apart from their dependence on the magnitude of the temperature differences, these currents depend strongly on the location of the high and low temperature area. In the present paper an extreme situation will be discussed where the bottom of the balancecase is warmer than the top. Whether the convective flow is laminar or turbulent depends on the temperature difference. As result we find that at atmospheric pressures a temperature difference of 1K may lead to velocities of the order of centimeters per ,second. Such currents may cause extra forces which lead to mass discrepancies in the µg range.

INTRODUCTION

In an earlier paper (ref.1) we pointed out that temperature differences in a balance case can cause convection currents in the air and may so be a source of error in microweighing. In that paper we restricted ourselves to convection currents with a laminair flow pattern. In the present paper we shall set up estimations which may also be used beyond the limits of laminair flow.

ESTIMATION OF THE VELOCITY

An appropriate choice of the geometry of the balance case and of the temperature distribution can simplify the calculations considerably. For that reason we start from a balance case which allows for a two dimensional approach by having a large dimension b perpendicular to the plane of drawing. In the plane of drawing we take the balance case to be a square (see Fig.1) with length ℓ.

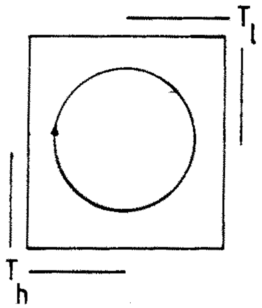

Fig.1: Schematic representation of the balance case

Regarding the temperature distribution we distinguish three parts of the balance case:

- one quarter of the surface of the balance case is at high temperature T_h,
- one quarter is at low temperature T_ℓ,
- the remaining is at roomtemperature $T_o = \frac{1}{2}(T_h + T_\ell)$.

It has to be mentioned that this configuration will cause large convection effects.
The driving force of the convection is caused by the density differences in the air due to temperature differences. This difference will be caused by heat transport to and from the low and high temperature regions of the case.
Let us for simplicity consider the situation where the gas is divided into two equal parts with temperatures $T_o + \frac{1}{2}\Delta T$ and $T_o - \frac{1}{2}\Delta T$ respectively. If we restrict ourselves to stationary state conditions we learn from the equilibrium of forces

$$F_g = \tfrac{1}{2}\ell^2 b\rho g \frac{\Delta T}{T_o} \tag{1}$$

where F_g is the friction force caused by the wall and ρ is the density of the gas.

For the heat transfer P in the LHS of the case from wall to gas we use

$$P = \ell b K\,(T_h - T_o - \tfrac{1}{4}\Delta T) \tag{2}$$

where K is the heat transfer coefficient.
The average temperature of the gas in the lower part of the case is taken to be $T_o + \frac{1}{4}\Delta T$. The heat balance for steady state conditions results in the equation:

$$P = \rho c\ \tfrac{1}{2}\ell^2 b\ \frac{dT}{dt} \approx \tfrac{1}{2}\rho c \ell^2 b\ \frac{\Delta T}{2\ \ell/v} = \tfrac{1}{4}\ \rho \ell b v \Delta T \tag{3}$$

Here the assumption has been made that the air warms up at a constant rate over half a cycle.
For the frictional force F_f we use under non-laminar flow conditions

$$F_f = f\ \tfrac{1}{2}\ \rho v^2\ 4\ell b \tag{4a}$$

where
v is the average flow velocity and
f is the drag coefficient.
One of the advantages of the geometry chosen is the fact that we may expect the convection current to approach a circular shape. This reduces conscientious objections due to neglecting internal frictional forces.

In the case of laminar flow we use

$$F_f = 4\ell b\ \frac{v}{\frac{1}{4}\ell}\ \eta = 16\ b\eta v \tag{4b}$$

where η is the viscosity of air.
In both cases we are left with four equations from which we can solve ΔT, P, F_f and v.

In the case of non-laminar flow, using $\rho cv \gg K$, the solution for v reads

$$v = \left(\frac{\ell g k}{\rho f c} \frac{T_h - T_o}{T_o} \right)^{1/3} \tag{5a}$$

In the case of laminar flow we get

$$v = \frac{K}{2\rho c} \left(\sqrt{1 + \frac{\ell^2 \rho^2 c g}{2\eta K} \frac{T_h - T_o}{T_o}} - 1 \right) \tag{5b}$$

It is interesting to calculate when the two solutions for v are equal. Then equations (4a) and (4b) should be identical. This leads to the condition

$$\frac{v\rho\ell}{\eta} = \frac{8}{f} \tag{6}$$

Considering the fact that the LHS of eq. (6) represents Reynolds number (Re) and that 10^{-2} is a reasonable guess for the value of f, we get as a result

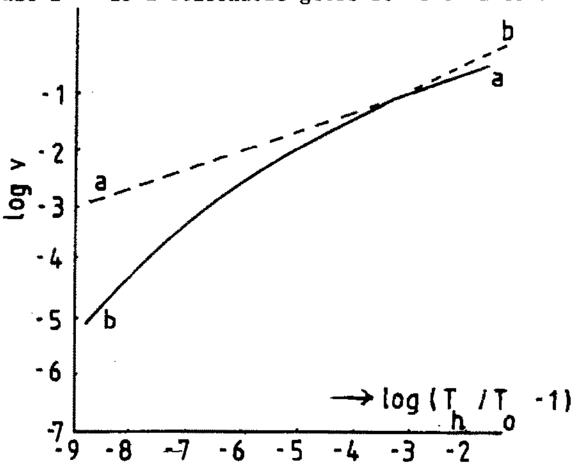

Fig.2 Velocities in the balance case as a function of (T_h/T_o-1)

that the limit of laminar flow should occur for Re = 800 which is quite acceptable. For our approximations we use the following values.
$K = 2{,}5\ Wm^{-2}K$; $\eta = 2\ 10^{-5}\ Nsm^{-2}$; $\rho = 1\ kgm^{-3}$; $C = 10^3\ WsK^{-1}kg^{-1}$;
$\ell = 0.3m$; $f = 10^{-2}$

Fig.2 shows the resulting velocities where the curves a and b refer to the equations 5a and 5b respectively.

ESTIMATION OF THE FORCE

In order to estimate the disturbing force caused by the convection flow we have to make assumptions about the shape of the part of the balance that is submitted to this flow. We have assumed this part to be a sphere. This should in principle allow for the use of Stokes' equation at very low Reynolds numbers and for the use of a quadratic relation in v at high Reynolds numbers. These two can be combined by using

$$F_d = \tfrac{1}{2}\,\rho A v^2 C_d \qquad (7)$$

Where the drag coefficient C_d is dependent on Reynolds number.
For the values of C_d we used the data presented in ref. (2). For A we use as a typical value $10^{-4}\,m^2$. The values of the force resulting from equation 7 are presented in Fig.3.

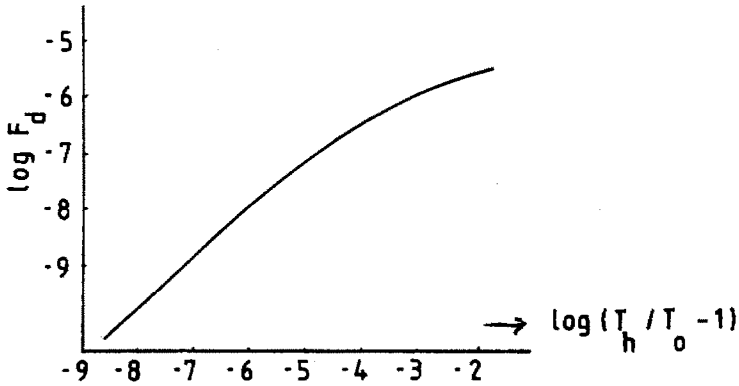

Fig.3. Disturbing force as a function of $(T_h/T_o - 1)$

When considering these results we should not forget that quite rough approximations have been used and that very unfavourable temperature distributions have been chosen.

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