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To cite this article: I Kolesnichenko et al 2017 J. Phys.: Conf. Ser. 891 012075

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On boundary conditions in liquid sodium convective experiments

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Abstract. Turbulent convection of liquid sodium in a cylindrical cell, heated at one end face and cooled at the other, inclined to the vertical at angle 0 and $\pi/4$ is studied experimentally and numerically by solving the Oberbeck-Boussinesq equations with the LES (Large Eddy Simulation) approach for small-scale turbulence. The aspect ratio is one, i.e. cylinder length is equal to diameter L = D = 200 mm. The simulations were done using fixed heat flux thermal boundary conditions for the cylinder faces. To resolve the general problem of boundary condition in convective experiment with low Prandtl number liquids, a special kind of heat exchanger were designed for the experimental setup. Each heat exchanger is a temperaturecontrolled MHD (magnetohydrodynamic) stirrer, filled with sodium and separated from the convective cell by a thin copper plate. We demonstrate the efficiency of MHD stirring for the temperature control. In convective experiments the Rayleigh number, determined by the cylinder diameter, was in the range from $4.7 \cdot 10^6$ to $1.7 \cdot 10^7$. We show that the structure of the flow and the efficient heat transfer strongly depend on the inclination angle.

1. Introduction

The study of turbulent convection in liquid metals is largely stimulated by particularity of heat and mass transfer in fluids characterized by low Prandtl numbers and by their application as coolants in nuclear reactors and fusion reactors.

Experimental studies of liquid metal convection is a hard task and only few studies are known. Rayleigh-Benard convection (RBC) of mercury was studied in short vertical cylinders 0.03 < L/D < 0.22 (*L* is the length, *D* is the diameter) under moderate Rayleigh numbers $2 \cdot 10^4 < Ra < 10^6$ [1]. The more studied problem is the RBC of mercury in cylinders with unit aspect ration L = D [2, 3, 4], which cover a large range of Rayleigh numbers $10^5 < Ra < 10^{11}$.

Recently turbulent convection of sodium in relatively long cylinders with different orientation to gravity were performed for cylinder with aspect ratio L/D = 5 [5, 6] and L/D = 20 [7, 8]. Liquid metal experiments on turbulent convection are very labor-consuming and allow very few possibility for velocity measurement and for reconstruction of full 3D velocity and temperature field. Therefore the 3D simulations are of great interest, but again the case of low Prandtl number is the less elaborated [9]. An extensive numerical study of convection in a cylinder L/D = 1 for various orientation to gravity was done in [10] ($0.1 \le Pr \le 100$) and revealed a nontrival dependence of effective heat transport on Prandtl number and inclination angle.

The convection in cylinders with unit aspect ratio is the more studied case for RBC [11]. Thus numerical and experimental data for this aspect ratio are very required for low Prandtl number fluids $Pr \approx 0.01$. A general problem of liquid metal thermophysical experiments is the problem of thermal boundary conditions [1].

In our work we start the numerical and experimental study of sodium convection in a cylinder L/D = 1 under various inclination angle β to gravity. We present in this paper first examples of numerical simulations (for $\beta = 0$ and $\beta = \pi/4$), we described a new experimental setup in which we have designed a special heat exchanger to provide controllable boundary conditions, and we provide first experimental data, supporting the efficiency of technical solution proposed.



Figure 1. (a) – scheme of the experimental setup: 1 – convective cell, 2 – hot heat exchanger, 3 – cold heat exchanger, 4 – expansion vessel, 5 – inductor; (b) – photo of the setup.

2. Numerical simulations

Our simulations intend to reproduce the convective flow in the experimental setup to recover in simulations the flow characteristics, which are unattainable in laboratory experiments. Thus, we try to approximate the actual parameters of the setup designed for experiments. The computational domain is a cylinder with L = D = 200 mm (figure 1). The simulations are done for the mean sodium temperature 165°C with corresponding Prandtl number Pr = 0.0083.

The numerical code solves the Oberbeck-Boussinesq equations of thermogravitational convection. We use the LES (Large Eddy Simulation) approach for modelling turbulence, namely the Smagorinsky-Lilly model [12, 13] with the Smagorinsky constant $C_s = 0.17$ and turbulent Prandtl number $Pr_t = 0.9$. To resolve the boundary layers, a non-uniform numerical grid with a higher density of points near the boundaries was used. The number of grid points in the boundary layer should be not less than 5 [14, 15]. Simulations were carried with a numerical grid consisting of 1.8 million nodes. The terms with time derivatives are discretised using an implicit Euler scheme. The convective terms are calculated by the TVD (Total Variation Diminishing) scheme. All the simulations were run using the free and open source finite volume code OpenFOAM 4.1. The numerical simulations were performed using Uran supercomputer of IMM UB RAS. We considered the sodium convection in the cylinder under fixed heat power Q = 1 kW (the heat flux is $q = 31.8 \text{ kW/m}^2$) and Q = 5 kW (the heat flux is $q = 159.2 \text{ kW/m}^2$). The sidewalls are heat insulated. The no-slip velocity conditions are applied at all boundaries.

We analyse the general structure of the turbulent convective flow that develops in the cylinder under fixed heat flux at vertical $\beta = 0$ and inclined at $\beta = \pi/4$ positions of the cylinder. The

doi:10.1088/1742-6596/891/1/012075



Figure 2. Results of simulations for Q = 5 kW: vertical cylinder $\beta = 0$ (upper row) and inclined cylinder $\beta = \pi/4$ (lower row). (a,e) – time-averaged vertical component of velocity in the xOy plane, z = 0.1 m; (b,f) – time-averaged velocity field in the vertical cross-section (plane with LSC); (c,g) – standard deviation of velocity field in the vertical cross-section; (d,h) – standard deviation of temperature field in the vertical cross-section.

Rayleigh number $Ra = \alpha g \Theta D^3 / (\nu \chi)$, defined through the diameter and the mean temperature difference Θ between the faces, varies with inclination from $4.7 \cdot 10^6$ to $1.7 \cdot 10^7$.

The mean velocity fields for fixed heat power Q = 5 kW are shown in figure 2. The largescale circulation (LSC) appears against the turbulent background in both cases. Figures 2a,e show the time-averaged vertical velocity component $\langle U_z \rangle$ in the horrizontal cross-section which demonstrates the orientation of LSC. In the vertical cylinder, the LSC and two counter rotating ring vortices exist near both end faces (figure 2b). In the inclined cylinder the LSC covers the whole cavity (figure 2f). In the latter case the LSC is much stronger and the maximum flow velocity is much higher (8.8 cm/s versus 3.6 cm/s).

The intensity of turbulence can be characterized by the distribution of the energy of turbulent pulsations or by the standard deviation (SD), shown in figures 2c,g. The most developed turbulence is observed in the vertical cylinder. The intense small-scale turbulence occupies the whole cylinder, and is most intense along the walls (figure 2c). In the inclined cylinder, the total intensity of turbulence decreases and its distribution differs. The most intense pulsations are observed in the areas close to "corners" (figure 2g).

Figures 2d,h show the corresponding SD fields for the temperature. The strongest pulsations occur in the vertical cylinder (figure 2d). The temperature pulsations at $\beta = \pi/4$ (figure 2h) are considerably lower than at $\beta = 0$ and its distribution is determined by the LSC, which carries up the hot fluid along the upper sidewall and carries down the cold fluid along the lower sidewall.

Standard thermal boundary conditions for RBC imply there is a fixed temperature at the horizontal boundaries. But it is difficult to measure the actual temperature field on the face end in sodium experiments. Thus we perform simulations with fixed homogeneous heat flux (the heater power is known) and then estimate the actual temperature difference between the face ends. For fixed heat flux q the Nusselt number is defined as $Nu = qL/(\lambda\Theta_q)$, where λ is the

IOP Conf. Series: Journal of Physics: Conf. Series **891** (2017) 012075 doi:10.1088/1742-6596/891/1/012075

thermal conductivity of sodium and Θ_q is the calculated difference of the average temperature over the heater and cooler plates under fixed heat flux q.

For $\beta = 0$, Q = 1 kW it was found that $\Theta_q = 10.1^{\circ}$ C and Nu = 7.7; for $\beta = 0$, Q = 5 kW we obtained $\Theta_q = 35.9^{\circ}$ C and Nu = 10.8; for $\beta = \pi/4$, Q = 5 kW we got $\Theta_q = 26.0^{\circ}$ C and Nu = 14.9.

3. Experimental setup

The experimental setup consists of a closed cylindrical convective cell 1 (figure 1), which is connected to the chambers of hot 2 and cold 3 heat exchangers. The cell and the chambers are filled with sodium. The cell and chambers are separated by 2 mm thick copper discs, which are the end faces of the cell. A heater with a maximum power of 15kVA is installed in the chamber of the hot heat exchanger 2. The cold heat exchanger has a copper needle-plate radiator located in the casing and forcedly blown by air. The chamber of the cold heat exchanger 3 is connected to the expansion vessel 4. The entire setup is located on the swing frame, by which the channel can be tilted at an angle of β from horizontal to vertical position.

The cell and heat exchanger are made of stainless steel pipe (D = 212 mm) with a thickness of 3.5 mm. The channel has 28 thermocouples with an isolated junction of 1 mm, installed in the cell at a distance of 89 mm from the channel axis. Thermocouples are located on 8 lines of 3-5 pieces with an equal in azimuth step $\pi/4$.

4. Heat exchanger

The peculiarity of our heat exchangers is that instead of traditional thick copper plates, thin plates intensively washed with liquid sodium, are used. The required flow of sodium in the chambers of heat exchangers is provided by travelling magnetic field (TMF), like in MHD stirrers [16, 17]. Each heat exchanger is equipped with six induction coils 5 (figure 1), shifted to the outer end face of the exchanger. The value of electromagnetic (em) force and, consequently, stirring velocity intensity are characterized by the value of electric current I feeding the coils. The em-force is directed oppositely to the OZ axis. A set of 9 thermocouples are installed in each heat exchanger, allowing to control the azimuthal and axial temperature distributions.

To optimize the parameters of heat exchangers we performed simulations of stirring using a mathematical model, based on the standard MHD equations and a semi-empirical $k - \epsilon$ turbulence model. The magnetic Reynolds number in this process is small, so the transfer of the magnetic field by the flow is insignificant. Three-dimensional calculations have been performed for pure liquid sodium using Ansys.

Calculations of the electrodynamic part showed that the magnetic field and the induced electric current are localized in the inductor region. The fields quickly decay in the direction towards the channel inside the chamber of the heat exchangers. Thus, the fields practically do not penetrate into the convective cell and do not affect the process under investigation.

Figure 3 shows an example of calculated velocity field for the feeding current I = 15A. The em-force generates a toroidal vortex, which is localized in the inductor region. The change of the TMF direction leads to change of the em-force and vortex rotation direction (figure 3a,b). The em-force is localized in a narrow layer near the chamber wall. The flow is not stationary and the pulsation level is high enough in the region nearby the convective cell. In this region the mean flow is practically absent (figure 3c). This should provide a good mixing of the temperature.

5. Experimental tests

The effect of the liquid sodium mixing in heat exchanger was studied experimentally by analysing the signals of 9 thermocouples located in the heat exchangers. The efficiency of mixing was



Figure 3. Examples of calculated velocity field in the axial cross-section of heat exchanger with upward (a) and downward (b) direction of em-force. Calculated profiles (c) of averaged velocity V_z : downward (1,2) and upward (3,4) direction of em-force, (1,3) – profiles located in the center of inductor, (2,4) – profiles located close to convective cell.

estimated by two characteristics:

$$A = \langle |\langle T \rangle_t - \langle \langle T \rangle_t \rangle_v | \rangle_v, \qquad B = \langle \langle rms(T - \langle T \rangle_t) \rangle_t \rangle_v,$$

here T is the temperature, rms(..) is the root mean square, $\langle .. \rangle_t$ is averaging over time, $\langle .. \rangle_v$ is averaging over volume (in fact, over the 9 thermocouples). The angle of the channel slope was $\beta = \pi/4$. At such angle, the effect of mixing can be reliably estimated, because the LSC is formed without stirring in the heat exchanger chambers [5]. Indeed, without mixing, the value of A is maximal (figure 4a). The enhancement of mixing leads to decrease of this parameter. This indicates a decrease in the difference between the thermocouple readings. The mixing does not have much influence on the value of temperature pulsations. With increasing the feeding current, the level of pulsation increases (figure 4b). With further increase the level gradually decreases, approaching the original value (without mixing).



Figure 4. Results of experiments: (a) – A vs feeding electric current I, (b) – B vs feeding electric current I; squares – hot heat exchanger, circles – cold heat exchanger.

6. Conclusions

This work continues a series of studies of liquid sodium convection in cylinders with different aspect ratio and different orientation to gravity [5, 7].

To study the convection in liquid sodium at the aspect ratio L/D = 1 and different angles of inclination, an experiment setup with improved control of the boundary conditions was created. The key element of the design is liquid metal heat exchangers with electromagnetic stirrers. Calculations of processes in heat exchangers have shown that electromagnetic processes do not influence the processes in the convective cell because of the sufficient distance of the inductor of the stirrer. The core of the flow is localized in the region of the inductor. In the region near the convective cell the flow is shapeless and nonstationary, which in this case is favorable for temperature homogenization.

Experiments have shown that mixing increases the temperature homogenization inside the chambers of liquid metal heat exchangers. After a certain value (approximately, I = 10A), there is no significant improvement in homogenization, so in the experiment we may use the moderate mixing. In this case, the level of temperature pulsations does not change significantly relative to the initial level at operating current values. Thus, the experimental setup significantly improves the control of the boundary conditions at the end faces of the convective cell, allows to realize a near-uniform temperature distribution.

Numerical simulations of the turbulent convection of liquid sodium (Pr = 0.0083) under Rayleigh numbers in the range from $4.7 \cdot 10^6$ to $1.7 \cdot 10^7$ in an inclined cylinder with aspect ratio L/D = 1, heated at one end face and cooled at the other, showed that the structure of the flow strongly depends on the inclination angle.

Acknowledgement

This study was supported by the project RFBR 16-01-00459. We thank Andrey Vasiliev, Alexander Shestakov, Alexander Pavlinov and Andrey Mamykin for help in experiment preparation and performance.

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