Effect of Mean Wind on Turbulent Convection

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Introduction

Comparison of properties of laboratory turbulent convection with and without coherent structures (mean wind)

Production and dissipation in turbulent convection

Theoretical predictions and comparison with laboratory experiments

Properties of forced turbulent convection (without mean wind) and comparison with theoretical predictions

Conclusions and future studies
Closed cloud cells over the Atlantic Ocean
Open cloud cells over the Pacific Ocean
Coherent Structures (Mean Wind) in Laboratory Turbulent Convection
Laboratory Turbulent Convection

- In laboratory turbulent convection (in the Rayleigh-Benard apparatus) coherent organized features of motion, such as large-scale circulation patterns (the "mean wind") are observed.

- There are several open questions concerning these flows:
  - How do they arise?
  - What is the effect of mean wind on turbulent convection?
  - Is it shear-produced turbulence (due to the mean wind) or buoyancy-produced turbulence?
Experimental set-ups

Chamber A=4: $26 \times 58 \times 13 \text{ cm}^3$
Chamber A=2: $26 \times 58 \times 26 \text{ cm}^3$
Chamber A=1: $26 \times 26 \times 26 \text{ cm}^3$
Experimental Set-up for Forced Turbulent Convection
Experimental set-up
Particle Image Velocimetry System

Particle Image Velocimetry Data Processing
Experimental set-up for temperature measurements
Unforced Convection: $A = 1$

$\bar{U}(y,z)$

$\bar{T}(y,z)$
Unforced Convection: $A=2$
Experimental Set-up for Forced Turbulent Convection
Temperature Field in Forced and Unforced Turbulent Convection

Forced turbulent convection (two oscillating grids)  Unforced convection
Mean Velocity and Temperature Fields (One Grid Forcing)

2.2 Hz

4.4 Hz

6.4 Hz

10.4 Hz
Temperature Field in Forced and Unforced Turbulent Convection

Forced turbulent convection (two oscillating grids)

Unforced convection
Preferential Coherent structures for $A=2$

Preferential Coherent structures for $A=4$
Turbulent Kinetic Energy

\[
\frac{DE_K}{Dt} + \text{div} \Phi_K = -\langle u_i u_j \rangle \nabla_j U_i + \beta F_z - \varepsilon
\]

\[
E_K = \langle u^2 \rangle / 2, \quad D/Dt = \partial / \partial t + U \cdot \nabla,
\]

\[
P = -\langle u_i u_j \rangle \nabla_j U_i \text{ is the production caused by the shear}
\]

\[
\beta F_z \text{ is the production caused by the turbulent vertical heat flux } F_z = \langle u_z \theta \rangle
\]

\[
\varepsilon = E_K / \tau \text{ is the dissipation rate of the turbulent kinetic energy}
\]

\[
\beta = g / T_*
\]
Study of Velocity Field inside Coherent Structures
Study of Velocity Field inside Coherent Structures
Turbulence in large Rayleigh number convection with coherent structures is produced by shear, rather than by buoyancy (a main contribution to the turbulence kinetic energy production in turbulent convection with large-scale coherent structures is due to the non-uniform large-scale motions).

\[ P \approx \varepsilon \]

FIG. 5: Dependence of the shear-induced production term \( P = -\langle u_i u_j \rangle \nabla_j U_i \) of the turbulent kinetic energy versus the dissipation rate \( \varepsilon \) obtained in the experiments with turbulent convection in the chamber with \( A \approx 2 \) (triangles for two-cell flow pattern and circles for one-cell flow pattern) and \( A \approx 4 \) (squares for two-cell flow pattern). The dashed line corresponds to \( P = 1.1 \varepsilon \). The production rate \( P \) and the dissipation rate \( \varepsilon \) are measured in \( \text{cm}^2 \text{ s}^{-3} \).
Preferential Coherent structures for A=2

Preferential Coherent structures for A=4
Turbulent Kinetic Energy

\[
\frac{D E_K}{Dt} + \text{div} \Phi_K = -\langle u_i u_j \rangle \nabla_j U_i + \beta F_z - \varepsilon
\]

\[E_K = \langle u^2 \rangle / 2, \quad D/Dt = \partial/\partial t + U \cdot \nabla,\]

\[P = -\langle u_i u_j \rangle \nabla_j U_i \text{ is the production caused by the shear}\]

\[\beta F_z \text{ is the production caused by the turbulent vertical heat flux } F_z = \langle u_z \theta \rangle\]

\[\varepsilon = E_K / \tau \text{ is the dissipation rate of the turbulent kinetic energy}\]

\[\beta = g / T_*\]
Large-Scale Shear versus Temperature Difference

\[ S_{\tau_z} = \text{const} \]
\[ \tau_z = \ell_z / u_z \]
\[ S = \left[ (\nabla_z U_y)^2 + (\nabla_z U_x)^2 \right]^{1/2} \]

FIG. 6: Measured large-scale shear \( S_{\tau_z} \) versus the temperature difference \( \Delta T \) between the bottom and the top walls of the chamber obtained in the experiments with turbulent convection in the chamber with \( A \approx 2 \) (triangles for two-cell flow pattern and circles for one-cell flow pattern) and \( A \approx 4 \) (squares for two-cell flow pattern). The temperature difference \( \Delta T \) is measured in K.
Turbulent convection is weakly inhomogeneous.

\[ \tau_z = \ell_z / u_z \]

FIG. 9: Dependencies of the vertical component \( u_z \) of the measured turbulent velocity (upper panel) and the vertical turbulent time \( \tau_z = \ell_z / u_z \) (lower panel) versus the horizontal \( y \) coordinate obtained in the experiments with turbulent convection in the chamber with \( A \approx 4 \) (squares correspond to measurements at the central \( yz \) plane at \( x = 15 \) cm, triangles are for the \( yz \) plane at \( x = 5 \) cm and circles are for the \( yz \) plane at \( x = 25 \) cm). The turbulent velocity is measured in cm s\(^{-1}\), the turbulent time is measured in seconds and the coordinate \( y \) in cm.
FIG. 8: Turbulent length scales $\ell_x$ (panel a), $\ell_y$ (panel b) and $\ell_z$ (panel c) versus the temperature difference $\Delta T$ between the bottom and the top walls of the chamber obtained in the experiments with turbulent convection in the chamber with $A \approx 2$ (triangles for two-cell flow pattern and circles for one-cell flow pattern) and $A \approx 4$ (squares for two-cell flow pattern). The turbulent length scales are measured in cm and the temperature difference $\Delta T$ is measured in K.
Turbulent Velocity versus Temperature Difference

\[ u_z \propto (\Delta T)^{0.45} \]

Theory:

\[ u \propto (\Delta T)^{1/2} \]

FIG. 7: Component \( u_x \) (panel a), \( u_y \) (panel b) and \( u_z \) (panel c) of the measured turbulent velocity versus the temperature difference \( \Delta T \) between the bottom and the top walls of the chamber (in log-log scale) obtained in the experiments with turbulent convection in the chamber with \( A \approx 2 \) (triangles for two-cell flow pattern and circles for one-cell flow pattern) and \( A \approx 4 \) (squares for two-cell flow pattern). The list-square fit for the experimental results is shown by dashed line. The turbulent velocity is measured in cm s\(^{-1}\) and the temperature difference \( \Delta T \) is measured in K.
Production and Dissipation Rate versus Temperature Difference

Theory:

\[ P \propto (\Delta T)^{1.26} \]

\[ P = \varepsilon \propto u^3/\ell \propto (\Delta T)^{3/2} \]

FIG. 11: Production rate \( P \) versus the temperature difference \( \Delta T \) between the bottom and the top walls of the chamber (in log-log scale) obtained in the experiments with turbulent convection in the chamber with \( A \approx 2 \) (triangles for two-cell flow pattern and circles for one-cell flow pattern) and \( A \approx 4 \) (squares for two-cell flow pattern). The list-square fit for the experimental results is shown by dashed line. The production rate \( P \) is measured in cm\(^2\) s\(^{-3}\) and the temperature difference \( \Delta T \) is measured in K.
Turbulent Velocity and Production

\[ \frac{\partial U}{\partial t} + (U \cdot \nabla)U = -\nabla \frac{P}{\rho} - \frac{g}{T^*} T + \nu_T \Delta U \]

Steady-state solution of Eq. for \((\nabla \times U)_x\)

\[(g/T^*) (\nabla y T) \sim -\nu_T \Delta (\nabla \times U)_x\]

\[\nabla_z U_y \propto \frac{L_z^2}{\ell^2/T^*} g \frac{\nabla y T}{T^*},\]

where: \(\nu_T \sim u \ell \sim \ell^2/\tau,\)

\[|\nabla_z U_y| \gg |\nabla y U_z|,\]

since \(S \sim \nabla_z U_y, \quad S_T = \text{const} \quad \text{and} \quad |\nabla y T| \propto (\delta T)_y/L_y \propto \Delta T\)

\[u \propto L_z \sqrt{g \frac{|\nabla y T|}{T^*}} \propto (\Delta T)^{1/2}\]

\[P = \varepsilon \propto u^3/\ell \propto (\Delta T)^{3/2}\]
Temperature Field inside Coherent Structure

\[ |\nabla_y T| \propto (\delta T)_y / L \propto \Delta T \]
Horizontal Temperature Difference versus Vertical Temperature Difference

\[ |\nabla_y T| \propto (\delta T)_y / L_y \propto \Delta T \]

FIG. 10: Dependence of the horizontal temperature difference \((\delta T)_y = |\nabla_y T| L_y\) inside one cell of the two-cell coherent structure versus the vertical temperature difference \(\Delta T\) between the bottom and the top walls of the chamber obtained in the experiments with turbulent convection in the chamber with \(A \approx 4\) (the filled squares denote left cell and the unfilled squares denote right cell). Here \(L_y\) is the horizontal size in the \(y\) direction of the one cell of the coherent structure. The list-square fit for the experimental results is shown by dashed line. The temperature differences \((\delta T)_y\) and \(\Delta T\) are measured in K.
Turbulent Velocity and Production

\[ \frac{\partial U}{\partial t} + (U \cdot \nabla)U = -\nabla \left( \frac{P}{\rho} + \frac{g}{T_*} \right) + \nu_T \Delta U \]

Steady-state solution of Eq. for \((\nabla \times U)_x\)

\[ \left( \frac{g}{T_*} \right) \left( \nabla_y T \right) \sim -\nu_T \Delta (\nabla \times U)_x \]

\[ \nabla_z U_y \propto \frac{L_z^2}{\ell^2/\tau} g \frac{\nabla_y T}{T_*}, \]

where:

\[ \nu_T \sim u \ell \sim \ell^2/\tau, \]

\[ |\nabla_z U_y| \gg |\nabla_y U_z|, \]

since \( S \sim \nabla_z U_y, \quad S_T = \text{const} \)

and

\[ |\nabla_y T| \propto (\delta T)_y/L_y \propto \Delta T \]

\[ u \propto L_z \sqrt{g \frac{|\nabla_y T|}{T_*}} \propto (\Delta T)^{1/2} \]

Experiments:

\[ u \propto (\Delta T)^{0.45} \]

\[ P \propto (\Delta T)^{1.26} \]

Theory:

\[ u \propto (\Delta T)^{1/2} \]

\[ P = \varepsilon \propto u^3/\ell \propto (\Delta T)^{3/2} \]
Forced Turbulent Convection
Temperature Field in Forced and Unforced Turbulent Convection

Forced turbulent convection
(two oscillating grids)

Unforced convection
FIG. 2: Components $u_y$ (triangles) and $u_z$ (squares) of the measured turbulent velocity versus the frequency $f$ of the grid oscillations for the unstably stratified turbulent flow at the temperature difference between the top and bottom walls $\Delta T = 50$ K. The turbulent velocity is measured in cm s$^{-1}$ and the frequency $f$ is measured in Hz.
FIG. 3: Turbulent length scales $\ell_y$ (triangles) and $\ell_z$ (squares) versus the frequency $f$ of the grid oscillations for the unstably stratified turbulent flow at the temperature difference between the top and bottom walls $\Delta T = 50$ K. The turbulent length scales are measured in mm and the frequency $f$ is measured in Hz.
Non-Dimensional Thermal ratio versus Frequency of the Grid Oscillations

For the high frequency $f > 10$ Hz

$$\frac{\ell_z |\nabla_z T|}{\sqrt{\langle \theta^2 \rangle}} \approx \text{const}$$

FIG. 7: The non-dimensional ratio $-\ell_z \nabla_z T/\sqrt{\langle \theta^2 \rangle}$ versus the frequency $f$ of the grid oscillations for the unstably stratified turbulent flow at the temperature difference between the top and bottom walls $\Delta T = 50$ K.
Budget Equation for Temperature Fluctuations

$$\frac{DE_\theta}{Dt} + \text{div} \Phi_\theta = -(F \cdot \nabla)T - \epsilon_\theta$$

$$E_\theta = \langle \theta^2 \rangle / 2,$$

$$F = \langle u \theta \rangle = -D_T \nabla T$$ is the turbulent heat flux;

$$\Phi_\theta = (1/2)\langle u \theta^2 \rangle$$ is the third-order moment that determines flux of $$E_\theta;$$

$$\epsilon_\theta \approx E_\theta / \tau_0$$ is the dissipation rate of $$E_\theta;$$

In a steady-state the budget equation yields:

$$\langle \theta^2 \rangle \approx -2\tau_0 (F \cdot \nabla)T \approx 2\ell_0^2 (\nabla T)^2$$

$$\Rightarrow \frac{\ell_z |\nabla z T|}{\sqrt{\langle \theta^2 \rangle}} \approx \text{const}$$
Non-Dimensional Velocity ratio versus Frequency of the Grid Oscillations

For the high frequency $f > 10$ Hz:

$$\frac{u_z^*}{u_z} \simeq [1 + \frac{2\ell_z \beta}{(u_z^*)^2} \sqrt{\langle \theta^2 \rangle}]^{-1/2}$$

FIG. 8: Ratios $u_z^*/u_z$ (squares) and $u_z^*/\tilde{u}_z$ (circles) of the measured turbulent velocity versus the frequency $f$ of the grid oscillations. Here $u_z^*$ is the vertical component of the measured turbulent velocity in the isothermal turbulence, $u_z$ is the vertical component of the measured turbulent velocity in the unstably stratified turbulent flow with the temperature difference between the top and bottom walls $\Delta T = 50$ K, $\tilde{u}_z$ is the vertical component of the effective turbulent velocity, $\tilde{u}_z = [(u_z^*)^2 + 2\ell_z \beta \sqrt{\langle \theta^2 \rangle}]^{1/2}$, that takes into account the production of the turbulence by buoyancy. The turbulent velocity is measured in cm s$^{-1}$ and the frequency $f$ is measured in Hz.
Turbulent Kinetic Energy

\[
\frac{DE_K}{Dt} + \text{div} \Phi_K = -\langle u_i u_j \rangle \nabla_j U_i + \beta F_z + \langle u \cdot f_f \rangle - \varepsilon
\]

\[E_K = \langle u^2 \rangle / 2, \quad D/Dt = \partial / \partial t + U \cdot \nabla,\]

\[P = -\langle u_i u_j \rangle \nabla_j U_i \text{ is the production caused by the shear} \]

\[\beta F_z \text{ is the production caused by the turbulent vertical heat flux } F_z = \langle u_z \theta \rangle \]

\[\langle u \cdot f_f \rangle \text{ is the production of turbulence caused by the grid oscillations} \]

\[\varepsilon = E_K/\tau \text{ is the dissipation rate of the turbulent kinetic energy} \]

\[\beta = g/T^* \]
In a steady-state the budget equation yields:

\[ \langle u^2 \rangle = 2\tau_0 \left[ \nu_T S^2 + \langle u \cdot f_f \rangle + \beta F_z \right] \]

\[(u^*)^2 = 2\tau_0 \left[ \nu_T S^*_x + \langle u \cdot f_f \rangle \right] \] is for the isothermal turbulence;

\[ F_z = -D_T \nabla_z T \] is the vertical turbulent heat flux;

\[ u^*_h \simeq (u^*_h)^2 \] is for the horizontal velocity;

For the high frequency \( f > 10 \text{ Hz} \):

\[ \langle \theta^2 \rangle \approx 2\ell_0^2 (\nabla_z T)^2 ; \]

\[ \frac{u^*_z}{u_z} \simeq \left[ 1 + \frac{2\ell_z \beta}{(u^*_z)^2} \sqrt{\langle \theta^2 \rangle} \right]^{-1/2} \]
References


Conclusions

- We found that a **main contribution** to the turbulence kinetic energy (TKE) production in turbulent convection with **mean wind** is due to the **large-scale shearing motions** in the **mean wind**. Turbulence in large Rayleigh number convection with **mean wind** is produced by shear, rather than by buoyancy.

- We determined the **scalings of global parameters** (e.g., the production and dissipation of TKE, the turbulent velocity and integral turbulent scale, the large-scale shear, etc.) of turbulent convection versus $\Delta T$. These scalings are in an agreement with our theoretical predictions.

- We studied experimentally a **forced turbulent convection** and observed **two different regimes**: (i) when the frequency of the grid oscillations is large, the mean wind is **totally destroyed**, the mean temperature gradient in the central flow region in the vertical direction $|\nabla_z T| \gg |\nabla_y T|$, and $\ell_z |\nabla_z T|/\sqrt{\langle \theta^2 \rangle} \approx \text{const}$, in agreement with the theoretical predictions; (ii) for the low frequency of the grid oscillations the thermal structure inside the mean wind is inhomogeneous and anisotropic, and $|\nabla_z T| < |\nabla_y T|$.

[see also Shang, Qiu, Tong and Xia, PRE (2004)].
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