Mitteilung

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EXPERIMENTAL STUDY OF TURBULENT MIXED CONVECTION IN A GENERIC AIRCRAFT CABIN UNDER HIGH-PRESSURE CONDITIONS

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The expression mixed convection (MC) refers to flows, where forced convection (FC) and thermal convection (TC) coexist and have the same order of magnitude. Often, MC occurs on large scales, for example in case of aircraft cabin ventilation [1]. The passengers emit heat and warm air rises due to buoyancy and interacts with FC from the air-conditioning. As a consequence, large-scale flow structures develop. The shape, the orientation and the dynamics of these structures determine the heat transport and thus the thermal comfort as well as the efficiency of aircraft cabin ventilation. With the objective to determine the characteristics of these structures an experimental study of MC is carried out as a function of the ratio of buoyancy to inertia forces.

MC is examined in a down-scaled generic cabin segment of an Airbus A380 upper deck. The spatial scaling of the cabin is based on a verified concept [2] in order to obtain full-scale characteristic numbers on laboratory scales. In accordance with the scaling concept, the volume flow $\dot{V}$, the heating power of the thermal manikins (TM) $\dot{Q}_{\mathrm{TM}}$ and the fluid pressure P are adjusted to obtain similitude between a down-scaled and the full-scale cabin. A photo of the configuration is shown in figure 1. The spatial scaling-factor for the cabin is $s_{\mathrm{L}} = 7$. Hence, the dimensions of the cabin are as follows: height $H = 0.31$ m, width $W = 0.726$ m and length $L = 1.0$ m.

The convective air flow and the heat transfer in the cabin are defined by a set of non-dimensional numbers including: the Grashof number $\mathcal{G}r = {}^{g\,\beta\,\Delta T\,H^3}\!/_{\nu^2}$, the Reynolds number $\mathcal{R}e = {}^{\dot{V}\,L}\!/_{A_{\mathrm{in}}\,\nu}$, the Nusselt number for the TMs $\mathcal{N}u = {}^{\dot{Q}_{\mathrm{TM}}\,H}\!/_{A_{\mathrm{TM}}\,\lambda\,\Delta T}$ and the Prandtl number $\mathcal{P}r = {}^{\nu}\!/_{\kappa}$. Here g denotes the gravitational acceleration, β the coefficient of thermal expansion, α the heat transfer coefficient, ΔT the characteristic temperature difference, L the characteristic length, H the characteristic height, ν the kinematic viscosity, κ the thermal diffusivity, λ the thermal conductivity, $\dot{V}$ the volume flow, A_{in} the area of the inlet slots and A_{TM} the surface area of a TM. In addition, the Stanton number $\mathcal{S}t = \mathcal{N}u/\mathcal{R}e\mathcal{P}r$ is used to characterise the ratio of the heat transferred into a fluid to the thermal capacity of the fluid.

With the objective to study the flow structure formation, caused by to the mutual interplay of forced and thermal convection, planar two-component particle image velocity (PIV) is conducted in the cross section $0.5 \times L$ in longitudinal direction. The measurements are performed in a high-pressure wind tunnel at an air pressure of $P = 19.6$ bar. Figure 1 illustrates the time-averaged velocity vector fields for $\mathcal{R}e = 2380$ under isothermal conditions (fig. 1a) and with heated TMs for $\mathcal{G}r = 8.6 \times 10^3$ and $\mathcal{S}t = 14.7 \times 10^{-3}$

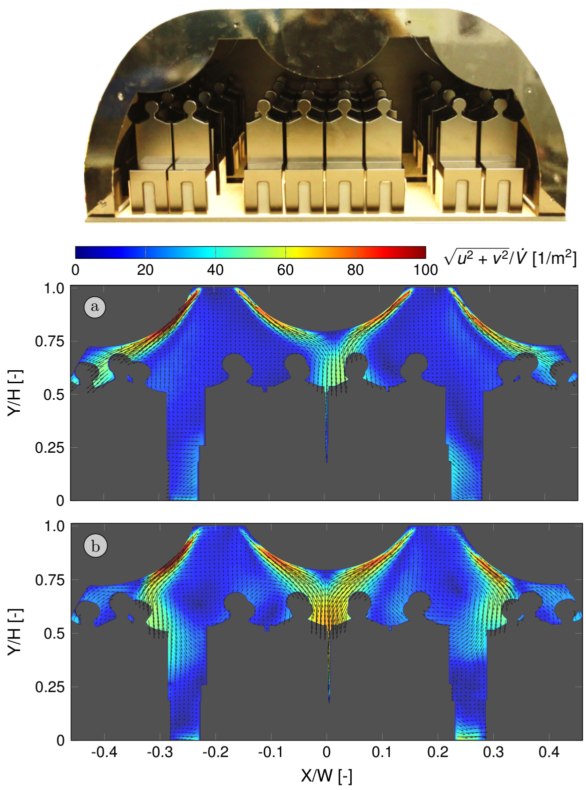

Figure 1: Photo of the scaled cabin segment with TMs (top). Time-averaged velocity vector field for $\mathcal{R}e = 2380$ under isothermal conditions (a) and with heated manikins for $\mathcal{G}r = 8.6\times10^3$ and $\mathcal{S}t = 14.7\times10^{-3}$ (b).

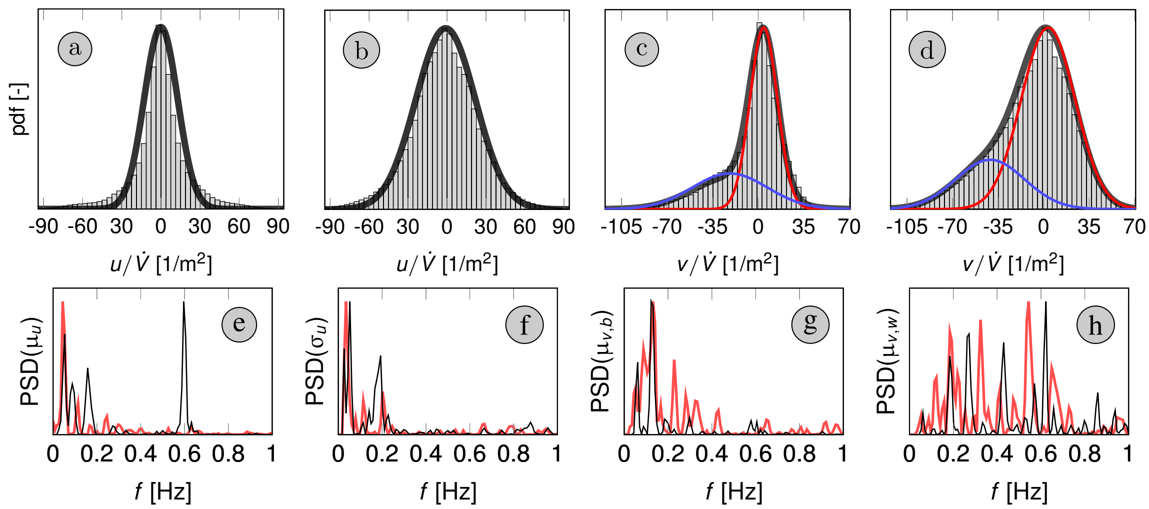

Figure 2: Proper density functions of the time-averaged velocity vector fields for the u- and v-component, respectively. (a) and (c) isothermal case at $\mathcal{R}e = 2380$. (b) and (d) thermal case at $\mathcal{R}e = 2380$, $\mathcal{G}r = 1.2 \times 10^4$ and $\mathcal{S}t = 14.7 \times 10^{-3}$. (e) - (h) PSD calculated by means of the time series of μ and σ of the corresponding PDFs. Here, red is used for the PSD of MC and black for the isothermal condition.

(fig. 1b). The velocity magnitude $\sqrt{u^2 + v^2}$ is colour-coded and standardised to the volume flow $\dot{V}$ of the incoming air. In both cases the air enters the cabin at the ceiling near the luggage compartment, follows the contour and detaches. However, in case of the flow with heated TMs, the additional buoyancy forces impact the wall jets significantly. For all wall jets a broadening and a previous detaching of the wall jet at the left and right side can be observed.
Since mixed convection in a cabin is generally turbulent, far from equilibrium or shows erratic behaviour, the time series of the velocity vector fields are analysed using several static methods such as Property Density Functions (PDF) of the velocity or Proper Orthogonal Decomposition (POD). Sample results of the PDF analysis are depicted in figure 2. The histograms 2(a-d) show the velocity distributions of the time-averaged velocity vector field (fig. 1(a) and (b)). Figures 2(a) and (b) depict the distribution of the u-component and figures 2(c) and (d) the distribution of the v-component for the isothermal and thermal case, respectively. Based on the assumption that the velocity components are normally distributed, the mean value μ and the variance σ are calculated by means of a least-square-fit. For the u-component it is found that the distribution of the isothermal and the thermal case are almost normally distributed with $\mu_u = 0$. However, in case of MC the interaction of the buoyancy flow with the wall jets leads to a broadening of the PDF. For the isothermal flow the variance is $\sigma_u = 2.18 \times 10^{-3}$ and for MC it is $\sigma_u = 3.89 \times 10^{-3}$. In case of the v-component the sum of two Gauss functions is fitted, with the blue-coloured graph representing the distribution of the wall jet and the red one representing the distribution for the bulk. Similarly, for the v-component, a broadening of the PDF at MC is found for the wall jet as well as for the bulk. In addition, the PDFs are calculated for each instantaneous velocity field (recorded at a repetition rate of 2 Hz). Based on these time series the power spectral density (PSD) of μ and σ is computed by means of a fast Fourier transformation (FFT). Here, the red graphs represent the MC cases and the black graphs those of the isothermal conditions. Figures 2(e) and (f) depict the PSD of μ_u and σ_u for the u-component and figures 2(g) and (h) show the PSD for μ_v of the v-component for the bulk and the wall jet, respectively. The PSD discovers a plethora of characteristic frequencies representing primarily the dynamics of the wall jet detachment as well the dynamics of large-scale circulations. A detailed discussion of these frequencies, to be exact the large-scale flow structure formation and its dynamics as well as its effect on the heat transport, will be presented at the conference. Furthermore, we will introduce the concept of scaling and the experimental configuration for the measurement under high-pressure conditions.

[1] M. Kühn, J. Bosbach, and C. Wagner. Experimental parametric study of forced and mixed convection in a passenger aircraft cabin mock-up. *Building and Environment*, 44(5):961 – 970, 2009.

[2] Andreas Westhoff. *Spaitial Scaling of Large-Scale Circulations and Heat Transport in turbulent Mixed Convection*. PhD thesis, Georg-August Universität Göttingen, 2013.