Experimental evaluation of the cooling and heating dynamics for a vertical ventilation concept in a generic car cabin

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Abstract. A low-momentum ceiling-based ventilation concept is compared to dashboard ventilation in a fullscale generic car cabin (GCC). The dimensions of the cabin correspond to the interior of a typical mid-size car. To provide different boundary conditions, representing winter and summer, the GCC is equipped with a jacket cooling/heating. Thermal manikins simulate the heat release and the obstruction of passengers. With the objective to verify the thermal comfort and the global heat transport characteristics in the GCC, the velocities as well as the air and surface temperatures are recorded. In the present study, the cooling and heating dynamics of the low-momentum ceiling ventilation are examined and compared to a simplified dashboard ventilation with regard to thermal comfort and heating/cooling efficiency. It is found that the cooling and heating dynamics of the low-momentum ceiling differ significantly in comparison with the dashboard ventilation. Overall the low-momentum ceiling offers several benefits regarding the thermal comfort and efficiency.

1. Introduction
In recent years, the performance of ventilation concepts in cars has been investigated in terms of improved thermal passenger comfort and enhanced energy efficiency. The thermal comfort in vehicles is characterised by the latent and the radiative heat transfer as well as by the heat transfer of forced and thermal convection. These physical mechanisms are primarily determined by the fluid temperature, the surface temperature of the passengers, the air and the heat release of the passengers. For the assessment of the thermal comfort, we applied the concept of equivalent temperature based on the standard ISO 14505-2 [1]. State of the art car ventilation systems were investigated using Particle Image Velocimetry [2]. Previous studies focused on determining thermal passenger comfort using a local equivalent temperature measurement system with heated manikins [3]. In addition to the static performance, the dynamic behaviour with heat-up processes in winter is of vital importance [4]. When it comes to autonomous driving with e.g. rotated front seats including innovative and flexible cabin layouts, the most promising ventilation with regard to the thermal comfort for all passengers and efficiency is achieved by vertical ventilation concepts [5]. The study reveals an improvement of the thermal comfort for vertical ventilation concepts under static conditions in a mid-size generic car cabin (GCC) [6]. The measurements of this study were performed with four heated thermal manikins in order to simulate passenger’s heat release and obstruction. However, a detailed analysis of the thermal comfort is just provided for the driver manikin. In addition to the measurement under static conditions, a study of the cooling [7] and heating [8] dynamics under summer and winter conditions was realised in the GCC. A comparison of three different vertical ventilation concepts with a simplified dashboard ventilation the so-called mixing ventilation (MV) revealed several benefits for an air supply system based on a trickle ceiling, which is called low-momentum ceiling ventilation (LMCV) in the following. With regard to the cooling efficiency for the summer conditions, MV was identified as the most efficient and comfortable ventilation concept. Nevertheless, at least a similar result was obtained for LMCV. For the winter case MV was significantly less efficient compared to the vertical ventilation.
systems. Here, a combination of LMCV with cabin displacement ventilation revealed the highest potential in case of thermal comfort and efficient ventilation. The intention of the present study is a detailed discussion on the heating and cooling dynamics of MV and LMCV with regard to the requirements of future cars.

2. Materials and methods

2.1. Ventilation concepts

The two ventilation concepts MV and LMCV were investigated in the GCC. MV was realised as dashboard ventilation, which is most common for today’s car compartments. MV is characterised by a high degree of mixing by jets of fresh air enter the cabin at defined positions, e.g. in the dashboard. The used air leaves the car through the luggage compartment. As reference case for this study, a simplified dashboard ventilation was applied based on plain circular openings installed in the dashboard (figure 1(a)). In case of LMCV the fresh air enters the cabin through a trickle ceiling, interacts with the buoyancy flow induced by the heated thermal manikins (TMs), and leaves the cabin through outlets located in the front and the rear section of the ceiling, (figure 1(b)). Detailed information on the different ventilation concepts can be found in [6].

Figure 1. Illustration of the airflow for (a) MV and (b) LMCV. (c) Side view of the GCC including the positions of the temperature probes and the infrared camera.

2.2. Test environment

Within the framework of the project Next Generation Car (NGC), a full-scale generic car cabin (GCC) with the dimensions of a mid-size car was constructed at the DLR in Göttingen. Figure 1(c) shows a side view of the GCC interior equipped with four TMs. For our study, the side wall of the GCC was equipped with a jacket cooling/heating system based on capillary tubes mounted on aluminium sheets. This system was integrated into the structure with the objective to simulate wall conditions representing summer and winter scenarios. In addition, the GCC was equipped with four heated thermal manikins (TM). The function of the TMs was to simulate the heat emission and obstruction of the passengers. Each TM was set to a heat release of 75 W. The temperatures of the airflow at the inlets (Tin), outlets (Tout) and walls (Twall) were monitored by 120 resistance temperature detectors (RTDs). A centrifugal fan in combination with a Venturi nozzle and four computer-controlled Peltier elements was used to ensure well-defined inflow conditions with a total volume flow of 28 l/s [9]. Additional detailed information for the experimental set-up were published in [6].

2.3. Experimental techniques and technologies

To determine the thermal environment in vehicles, previous studies focused on the investigation of single thermal parameters like air temperature, air velocity or surface temperature. However, radiation and thus the thermal sensation of people was not detected. To include these physical effects in the assessment of the thermal comfort, the equivalent temperature \( T_{eq} = T_S \cdot \dot{Q}/h_{cal} \) was determined using the surface temperature \( T_S \), thermal loss due to convection and radiation \( \dot{Q} \) and the combined heat transfer coefficient \( h_{cal} \) [1]. Therefore, a calibration method [10] was developed, in consideration of the guideline defined in ISO14505-2 [1], which allows to use the TM as \( T_{eq} \)-sensor. Based on the method, measurements of \( T_{eq} \) were performed for the TM on the driver seat.
In the following section, the cooling and heating dynamics for MV and LMCV will be discussed with regard to thermal comfort and heat transfer. The temperature boundary conditions for summer and winter scenario are listed in Table 1 and the inflow air velocities were 2.07 m/s for MV and 0.02 m/s for LMCV. All measurements started at thermal equilibrium. Therefore, the TM and jacket cooling/heating of the GCC operated at least 150 minutes before the ventilation with predefined air inlet temperature was started. In addition, Table 1 includes $\Delta T_{wall} = T_{wall}^{time=3000s} - T_{wall}^{time=0s}$, the total temperature change of the wall for the current measurement period as well as the time-averaged temperature ratio $\bar{T} = (T_{air} - T_{wall}) / T_{sys}$, where $T_{air}$ denotes the mean air temperature in the vicinity of the TM and $T_{sys} = (T_{in} + T_{out}) / 2$ the system temperature. A significant change of the wall temperatures was observed for MV in the summer scenario, while for LMCV $T_{wall}$ remained almost constant. In the winter scenario, smaller changes were also found for LMCV, however, $\Delta T_{wall}$ was four times higher in case of MV. The reason for these differences between MV and LMCV is attributed to the faster air velocities and the increased mixing in case of MV, which led to an increased heat transfer at the side walls. A detailed discussion on this effect can be found in the following section. In addition, the measurements show that the ratio $\bar{T}$ is always larger in case of LMCV compared to MV, indicating that for LMCV the incoming fresh air leads to temperature changes primarily close to the TM, while in case of MV significant temperature changes are caused at the surfaces. Consequently, almost the total enthalpy flux of fresh air is available for cooling/heating the TM in case of LMCV, while in case of MV just a part is used for the TMs.

### Table 1. Temperature boundary conditions as well as evaluation parameters for the efficiency of MV and LMCV in case of the summer and winter scenario.

|        | Summer |         |         | Winter |         |         |
|--------|--------|---------|---------|--------|---------|---------|
|        | $T_{in}$ | $T_{wall}^{time=0s}$ | $\Delta T_{wall}$ | $\bar{T}$ | $T_{out}$ | $T_{in}$ | $T_{wall}^{time=0s}$ | $\Delta T_{wall}$ | $\bar{T}$ | $T_{out}$ |
| MV     | 14.0   | 31.2    | 10.7    | 0.10(1) | 27.5    | 25.0    | 7.1     | 6.7 | 0.876(8) | 18.5    |
|        | 15.0   | 32.0    | 6.8     | 0.11(1) | 28.7    | 27.0    | 7.2     | 9.1 | 0.879(6) | 18.8    |
| LMCV   | 14.0   | 30.5    | 0.3     | 0.25(4) | 27.9    | 25.0    | 6.8     | 1.4 | 0.98(1)  | 23.2    |
|        | 15.0   | 31.1    | 0.3     | 0.29(4) | 30.3    | 27.0    | 6.9     | 2.8 | 0.96(1)  | 24.3    |

With the aim to verify the impact of the flow on the global heat transfer of the ventilation systems, we defined the parameter

$$\Theta = \frac{|T_{sys} - T_{wall}|}{|T_{sys}^{time=0s} - T_{wall}^{time=0s}|} - 1$$

(1)

It represents the temperature difference $T_{sys} - T_{wall}$ normalised by the starting conditions. Figure 2 shows $\Theta$ as a function of time. For the summer case (Figure 2a), the cabin was heated with $T_{wall} \approx 31^\circ C$ before starting the ventilation. With the onset of ventilation, the air was cooled down and thus an instantaneous and fast change of the mean air temperature $T_{sys}$ was obtained, while in contrast, the change of $T_{wall}$ was delayed and much slower. As a consequence, $T_{sys} - T_{wall}$ decreased in ratio to $T_{sys}^{time=0} - T_{wall}^{time=0}$ and $\Theta$ became negative. Here, for all cases $T_{sys} - T_{wall}$ reached its minimum after approximately one minute. In the following, the additional change of $T_{wall}$ led to a rise of $\Theta$. In case of LMCV $T_{sys} - T_{wall}$ became larger and $\Theta$ increased continuously which indicates that the temperature change and hence the cooling of $T_{sys}$ exceed the values at the wall. This means that the incoming enthalpy flow of fresh air is very effective cooling $T_{sys}$ and thus the TMs. In comparison, for MV, $\Theta$ started to decrease again after a period of about ten minutes. The reason for the smaller $T_{sys} - T_{wall}$ was found in the interaction of the high-momentum flow with the cabin surfaces and the resulting higher heat transfer at the surfaces. Here, after ten minutes the change of $T_{sys}$ was less significant than the change of $T_{wall}$, which means the incoming cold air primarily cools down the
cabin surfaces and the interior and less the TMs. As a consequence, MV is less energy efficient regarding the cooling of the passengers. For the winter case (figure 2b), the surfaces of the cabin had the mean temperature $T_{wall} \approx 7^\circ C$ before starting the heating ventilation. With the inflow of warm air, a strong gradient of $\Theta$ was obtained indicating an increase of $T_{sys} - T_{wall}$ for all cases. However, in case of LMCV $\Theta$ was always larger than for MV when comparing $T_{in} = 25^\circ C$ and $T_{in} = 27^\circ C$, respectively. For $t > 1000$ the difference was approximately 5% in case of $T_{in} = 25^\circ C$ and 4% in case of $T_{in} = 27^\circ C$. The high momentum flow for MV also led to a higher heat transfer between the surface and the fluid. As a result, all surfaces of the cabin were heated in case of MV, whereas in case of LMCV primarily the TM was heated. In conclusion, for both scenarios winter and summer LMCV was more efficient in comparison with MV regarding the cooling/heating of the TM. In case of MV, a noteworthy amount of energy was lost through the surface and due to the enhanced cooling/heating of the interior.

Figure 2. Temperature ratio $\Theta$ in case of the (a) summer and (b) winter scenario. MV is shown in magenta and LMCV in green lines for two different air inlet temperatures, respectively.

Figure 3. $\alpha_{eq}$ and $\Delta L R \alpha$ for MV (magenta) and LMCV (green) of three incoming air temperatures for summer (a) and winter (b), respectively. $\alpha_{eq}$ averaged over left/right body parts are shown in bars with the corresponding values on the left-hand axis, $\Delta L R \alpha$ normalised to $\alpha_{mean}$ are illustrated as dashed lines with corresponding values on the right-hand axis.

To quantify the dynamics of cooling and heating, the rate of temperature change $\alpha$, in the following termed as time coefficient, for the equivalent temperatures of the TM on the driver seat, was determined. Therefore, the temperature time series was approximated by means of $T(t) = \Gamma \cdot e^{-\alpha t + C}$ for the summer scenario and $T(t) = \Gamma \cdot \tanh(\alpha \cdot (t + 1)) + C$ for the winter scenario. For the sake of comparability, the equivalent temperature was normalised by $(T_{wall} - T_{in})$. Figure 3 depicts the averaged time coefficient ($\langle \alpha_{eq} \rangle$) of the left (L, striped bars) and right side (R, crossed
bars) (arms, hands, legs) as well as the normalised difference of the left and right side \( \Delta L_L \alpha = \frac{\alpha_e^L - \alpha_e^R}{0.5 \cdot (\alpha_e^L + \alpha_e^R)} \) (dashed lines) for MV (magenta) and LMCV (green) for (a) the summer and (b) winter scenario. For LMCV homogeneous cooling and heating dynamics were found. Here, \( \Delta L_L \alpha \) was always less than 2\% for the summer scenario and for the winter scenario even less than 1\%, indicating that the dynamics of \( T_{eq} \) were almost equal on the left and the right side. In contrast, significant differences of the dynamics regarding the left and right side were found for MV. The difference \( \Delta L_L \alpha \) was always larger than 3\% and even larger than 8\% in case of cooling with \( T_{in} = 16 \) and 9\% in case of heating with \( T_{in} = 23 \). The inhomogeneity of the dynamics or more precisely the larger differences of \( \Delta L_L \alpha > 8\% \) could negatively affect the thermal comfort of passengers. With the aim to study the energy efficiency, the temperature control efficiency \( \eta = \frac{T_{eq} - T_{air} - (\dot{Q}_{TM} \cdot \rho \cdot c_p)}{T_{in} - T_{air}} \) \[6, 7\] was calculated. It represents the ratio of the air enthalpy flow towards the passenger, corrected for its own heating power, and the air enthalpy flow into the cabin.

![Figure 4](image)

**Figure 4.** Time coefficient of the temperature control efficiency in (a) zero balanced \( (|\eta - \eta_0|) \) for MV (continuous lines) and LMCV (dashed lines) and (b) the difference \( (\Delta \eta) \) between MV and LMCV in case of summer with \( T_{in} = 15^\circ C \) (red) and in case of winter with \( T_{in} = 23^\circ C \) (blue), respectively.

The corresponding variables of the temperature control efficiency were: the heat load \( \dot{Q}_{TM} \) of the TMs, the volume flow rate \( q_f \) as well as the density \( \rho \) and the specific heat capacity of the air \( c_p \). The temperature control efficiency served as a measure for the energy effectiveness. Higher values represent better effectivity and lower values worse effectivity. Figure 4(a) depicts \( \eta \) for MV and LMCV as a function of time under summer and winter conditions. Here, \( \eta \) is zero balanced for the sake of comparability. As previously reported in other studies performed under static conditions [6], a five times higher \( \eta \) was observed for cooling as compared to heating for both ventilation systems. For the present scenario of the dynamics, a faster increase of \( \eta \) was found for MV under summer as well as under winter conditions. For the heating case \( (T_{in} = 23^\circ C - blue) \), almost constant values were observed for both ventilation systems after a slight increase in the first 1000 s. The cooling case \( (T_{in} = 15^\circ C - red) \) revealed a continuous increase over the entire period after a strong rise in the first 1000 s. In addition, the differences of temperature control efficiency between MV and LMCV for the summer and winter scenario are depicted in figure 4(b). As already indicated in figure 4(a), the higher momentum at MV led to a significantly faster temperature change of the equivalent temperature and thus better cooling dynamics. However, in case of the heating dynamics, no major differences were found after 1000 s.

4. Conclusion

In the present study, a low momentum ventilation concept (LMCV) for car cabins based on a trickle ceiling was compared experimentally to a simplified dashboard ventilation (mixing ventilation MV). The system-relevant physical properties were determined using resistance temperature detectors, an
infrared camera as well as a volume flow meter. Four thermal manikins (TM) were used to simulate the heat release of real passengers. In addition, a jacket heating/cooling was integrated into the structure of a generic car cabin (GCC) to realise summer and winter conditions. The dynamics of the global heat transfer in the cabin were analysed with the focus on the physical mechanisms, which determine the cooling and heating efficiency as well as the thermal comfort of the TM on the driver seat. The examination of the heat transfer uncovered significant differences between the two ventilation concepts, in particular for the cooling dynamics. After the displacement of the hot air in the cabin at the beginning and the compensation of the thermal loads, the temperature difference between the wall and the air near the TM decreased as a function of time in case of MV. In contrast, for LMCV an increase was observed. As a consequence, an enhanced heat transfer was obtained at the side walls in case of MV which resulted in a noteworthy heat loss. For the heating scenario, MV and LMCV revealed almost similar dynamics. However, in case of MV, the temperature difference between the air near the TM and the walls was significantly lower, which in turn means that there is a higher heat loss through the side walls in case of MV. Moreover, the cooling and heating dynamics on the TM surface were significantly more inhomogeneous in case of MV, which negatively impinged on the thermal comfort. For the temperature control efficiency, MV revealed benefits for the passengers in the front section of a car as a result of the larger heat transfer on the TM surface caused by the higher air velocities near the TM.

In summary, for the present investigations, LMCV stands out with advantages with regard to energy efficiency as well as the right-left symmetries compared to dashboard ventilation. In a next step, the impact of direct solar radiation has to be addressed. Moreover, energy savings could be achieved with a personalised LMCV, e.g. only for the driver.

Acknowledgement
The authors acknowledge Annika Köhne for proofreading this manuscript.

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