Interference Effects of Cooling Air-Flows with External Aerodynamics

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ABSTRACT: State-of-the-art vehicles already show an aerodynamically well improved bodywork with high-level efficiency. To further improve the aerodynamic drag, one potential area is the interaction between the underhood and the external flow. This study presents an experimental and numerical investigation of the interference effects of the cooling-air flow with the external-aerodynamics. Extensive measurements, like forces and total-pressure were accomplished. A simple dependency between drag and cooling-air mass-flow is derived, which will help for daily wind-tunnel work.

KEY WORDS: (Standardized) heat aerodynamic, wind tunnel test, computational fluid dynamics, engine (Free) interference effects, underhood flow [D1]

1. INTRODUCTION

The interaction between underhood-airflows and external aerodynamics is not yet fully understood. Investigating vehicles with the same underhood compartment, including the same inlet and outlet positions of the cooling-air can deliver different aerodynamic drag for different rear-end shapes. Furthermore it is well known that certain aerodynamic components operate for some rear-end shapes, while they do not show any optimization for another vehicle. In this study a correlation between cooling-air mass-flow and cooling-drag is presented on an experimental data basis and compared with the literature. Additionally one case of optimization will be shown, which has advantages for a notchback model, while it is neutral for a squareback vehicle.

2. EXPERIMENTAL AND NUMERICAL SETUP

2.1. Wind-tunnel model

Investigations were performed using a quarter-scale vehicle-model derived from a mass-production car. The vehicle was equipped with an underhood-section, see figure (1). The radiator was simplified by using mesh screens for a pressure drop and flow straighteners for rectifying the flow through the radiator. The mesh screens are exchangeable. With different porosities within the model radiator, different amounts of cooling-mass flow could be adjusted. It is similar to the radiator used at (1), and it inherits pressure sensors in a frame which was used at (2). Behind the radiator section, an engine and a gear-box are located, figure (1). Since for manufacturing reasons the model radiator is longer than in the mass-production vehicle, the engine was shortened to keep the distance between radiator and engine in scale, which is aerodynamically relevant. The gear box itself is correctly scaled in size and position especially to take account for the blockage of the underhood flow exiting the vehicle through the transmission-tunnel. While the cooling air inlet is at the grill, the cooling air outlet positions are at the transmission-tunnel and at the wheel-houses, which is state-of-the-art for conventional mass-production vehicles, figure (2). In this study, the standard configuration “STD” was to open both exits, the wheel-houses and the transmission-tunnel. One variation was to close the transmission-tunnel “TTC” (all air exits through the wheel-houses). Another option was to cover the wheel-houses “WHC” (all air exits through the transmission-tunnel). The underbody was built similar to a production car. Finally the rear-end shape is exchangeable to a notchback “NB” and squareback “SB” shape, compare figure (1).

Fig. 1 Schematic cutting plane through the aeromodel with both rear-end shapes to visualize the underhood section including radiator, engine and transmission.
Fig. 2 Underbody view of the wind-tunnel model. In red the closing of the transmission-tunnel “TTC”, in blue the closing of the wheel houses “WHC”. Both are optional.

2.2. Wind-tunnel

Experiments were performed in the Wind Tunnel B of the Technical University of Munich (TUM) at the Institute of Aerodynamics and Fluid Mechanics. It is a ¾-open-jet wind-tunnel of Goettingen-type with a ground plane and a scoop between nozzle and wind-tunnel table. The blockage accounts for approximately 7% with the here used aeromodel. Forces were measured using an external 6-component balance from Pfister. The wind-tunnel features a traversing system, whereby field measurements like the total-pressure measurements in the wake of the vehicle were performed. Measurements presented in this study were conducted at a free stream velocity of \( u_{\infty} = 50 \text{ m/s} \), which corresponds to a Reynolds number of about \( \text{Re} = u_{\infty} \cdot l / \nu = 3.8 \times 10^6 \) based on the vehicle length \( l \). Measurements and simulations were performed without moving ground and without rotating wheels. The standard deviation of is estimated to 0.003 in drag coefficient.

2.3. Numerical-Setup

Numerical simulations are performed based on the setup introduced in \(^{(3)}\). It is a simulation setup using the meshing tool Spider to get hexadominant meshes. The software package OpenFOAM® is used to solve the fluid-dynamic equations. The radiator is modeled using a porous-media to take account for the pressure drop within the radiator. Navier-Stokes-equations are used and turbulence is modeled by a RANS approach using the k-omega-SST turbulence model. The setup takes into account for the boundary conditions of the real wind tunnel, therefore the nozzle, the wind tunnel table including scoop between nozzle and table, the diffuser and the plenum was resolved in the discretization. The most important boundary conditions on bluff body experiments were simulated, for example the pressure gradient along the test section and the boundary layer distribution around the measurement position. Beside \(^{(3)}\), different other authors \(^{(4-7)}\) could show that this approach is reasonable to capture the correct wind-tunnel boundary-conditions.

3. RESULTS

3.1. Force Measurements of different cooling outlet positions

The aerodynamic optimization of the cooling outlet-position in wind-tunnel experiments is quite a difficult task, because the cooling mass-flow almost always changes on different outlet conditions. This is due to the geometric change of the outlet (the area changes), but also due to the different static pressure. Both, the outlet area and the outlet pressure strongly influence the cooling-air mass-flow and finally the cooling drag. The request arises to provide general conclusions, independent from the cooling-air mass flow. Therefore experiments have to be performed, where the cooling-air mass-flow could be changed without any geometric variations of the inlet and outlet positions. This was realized by different mesh screens within the radiator. Different radiator packages result in different flow resistance and therefore the cooling mass-flow in the underhood could be varied. Five different radiators were measured per each configuration. All three possible cooling outlet variations were investigated. Additionally both two rear-end shapes were measured. Figures (3-4) show the results of the measurements.

Fig. 3 Cooling drag \( \Delta c_{d_{c}} = c_{d_{(\text{Underhood})}} - c_{d_{(\text{MockUp})}} \) over cooling mass-flow \( c_{c} = \dot{m}_c / (\rho \cdot A_{ref} \cdot u_{\infty}) \) for the notchback model with different cooling-air outlet positions.

As one can see and as it was stated in different publications as well \(^{(8-10,12-13)}\), the cooling-drag shows a linear dependency on the cooling-air mass-flow. \(^{(11)}\) evaluated the cooling drag with semi-empirical equations. The result was a quadratic dependency of the cooling drag by the cooling mass-flow. Using experimental data showed that the quadratic term is smaller than the linear term, so that the equation reduces to a linear equation. The more cooling-air passes the underhood, the higher is the cooling drag. However different cooling outlet-positions show different gradients of the linear slope. The same occurs for
different vehicle rear-end shapes. But if the only difference between the different configurations occurs as the gradient of the linear slope, then it sounds reasonable to build the gradient by dividing the cooling drag with the cooling mass-flow. The resulting slope can be used to classify the possible cooling outlet-area, which was done by \(^{13}\) to compare different cooling paths. He found the relationship by closing the grill stepwise. For doing that, only the conventional cooling-drag measurement is necessary and not, like in this study, five different cooling mass-flow points. If the target mass flow is not hit, the value can be inter/extrapolated to the necessary cooling mass-flow and the resulting cooling-drag can be compared to the basic setup.

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Fig. 4 Cooling drag \(\Delta c_{dc} = c_d(\text{Underhood}) - c_d(\text{MockUp})\) over cooling mass-flow \(c_c = \dot{m}_c / (\rho \cdot A_{ref} \cdot u_w)\) for the squareback model with different cooling-air outlet positions.

In general it can be said that the configuration with a closed transmission-tunnel is the worst regarding mass-flow and drag balance. The baseline configuration of opened wheel-houses and opened transmission-tunnel delivers the highest cooling-air mass-flow. Interesting to notice is the case with closed wheel-houses. This case shows an improved performance for the notchback vehicle, while it does not improve the drag values at the squareback model. A possible explanation will be given in the next section 3.2.

3.2. Total-Pressure Measurements in the Wake of the Bluff Body

Total-pressure measurements were performed in the wake of the vehicle. Four configurations were determined. Both rear-end shapes and the version with and without underhood-flow in the standard configuration were investigated. The measurement planes were 100 mm and 500 mm behind the end of the vehicle, tangential to the main flow direction. Measurements were done using a Venturi-Pitot probe, which is insensitive to yaw angles up to 60° \(^{16}\), while the probe used here measures up to 50°. Measurement accuracy is estimated to 0.014 in total-pressure coefficient.

Fig. 5 Total-pressure in the wake of the notchback. On top, the car without cooling air, on bottom with underhood flow. The measurement grid is made visible.

Fig. 6 Total-pressure in the wake of the squareback. On top, the car without cooling air, on bottom with underhood flow. The measurement grid is made visible.
Figures (5,6) show the measurement results. As one can see, the difference between the vehicles with and without cooling-air is hard to identify. Therefore figures (7,8) show the total-pressure cooling-air difference, which means total-pressure with cooling minus total-pressure without cooling. As can be seen, in the side wake of the vehicles the biggest amount of total-pressure change occurs. Although cooling air exits as well through the transmission-tunnel, the alteration of the flow field in the center of the wake is small. \(^{(13)}\) could measure a similar total-pressure drop for another vehicle, which origins from the front wheel-houses. Interesting to notice is that the pressure loss is for both rear-end shapes similar in size, but the position seems to be more outside for the notchback. One explanation is that the main wake vortices rotate in different directions for different rear-end shapes, \(^{(8)}\). The different main flow-direction (in y-direction) of the wake leads to different suction of the wheel-house vortex due to cooling-air which results in different positions. Furthermore the different wake structures and its interaction with the wheel-house vortex can explain the different influence of the rear-end shape on the cooling drag.

3.3. Total-pressure drop from computational fluid dynamics

The simulation shows that the total-pressure drop arises at the wheel-houses, figures (9,10). From this source, it extends downstream to the wake of the vehicle. In the bottom view it can be seen that the underbody is mixed by total-pressure drop and total-pressure gain by the cooling-air flow. Figures (9,10) show the standard cases for both rear-end shapes. Isopanes of

$$\Delta p_{\text{Total}} = p_{\text{Total(Underhood)}} - p_{\text{Total(MockUp)}}$$

are shown.

Figures (11,12) show the total-pressure drop for a vehicle with wheel-house closing. As it can be seen, the total-pressure drop from the wheel-house disappears. There is a total-pressure drop still existing, which arrises from the underbody.
Wäschle (15) could show from measurements and from the numerical simulation that the front wheel-house vortices change on the type of ground simulation. It is assumed that the vortex system will be further changed due to the cooling-air exiting the front-wheel houses. To investigate the interference effects of cooling air-flow with rotating wheels, force measurements with ground simulations were performed, see section 4.

Fig. 10  Total-pressure loss around the squareback vehicle from simulation. Blue is $\Delta c_{pTotal} = -0.1$, red $\Delta c_{pTotal} = +0.1$.

Fig. 11  Total-pressure loss around the notchback vehicle with wheel-house closing from simulation. Blue is $\Delta c_{pTotal} = -0.1$, red $\Delta c_{pTotal} = +0.1$.

Fig. 12  Total-pressure loss around the squareback vehicle with wheel-house closing from simulation. Blue is $\Delta c_{pTotal} = -0.1$, red $\Delta c_{pTotal} = +0.1$. 
4. THE INFLUENCE OF THE GROUND SIMULATION

To further understand the interaction between the opened and closed wheel-houses for real-road conditions, experiments with ground simulation at the model wind-tunnel at the FKFS (Forschungsinstitut für Kraftfahrwesen und Fahrzeugmotoren) Stuttgart were performed. A 5-belt system is installed in the Goettingen-type wind-tunnel (14). The activation of the ground simulation can be done step by step, which means 1) no ground simulation “CGS”, 2) boundary layer suction system is added “BLS”, 3) centerbelt is added “CB”, 4) rotating wheels including mini-belts are added “WR”. The different configurations of cooling outlets were investigated for both rear-end shapes. Furthermore different rims were installed, figure (13). Due to limited experimental time in this wind-tunnel, the force measurements were reduced to one radiator. Field measurements could not be performed. Figure (14) shows the results of this experiment.

![Fig. 13](image)

As it can be seen, the cooling drag for the configurations 1)-3) does not alter very much. Adding the wheel rotation changes this behavior especially for the cooling-outlet located at the wheel-houses. The drag can increase strongly, dependent on the vehicle shape. The interference effects of the notchback are higher than for the squareback, which could be explained by the total pressure measurements and its interaction with the wake of the vehicle. The highest cooling drag could be measured for the closed transmission-tunnel, where all cooling air had to exit the wheel houses. Interesting to notice is that there is no additional cooling drag due to wheel rotation for the case with closed wheel-houses. A presentation of the influence of many different vehicle types on cooling drag and its interaction with ground simulation is presented by Wickern (17).

5. CONCLUSION

In this study a quarter-scale vehicle model was investigated regarding aerodynamic drag due to cooling-air. Measurements were performed using state-of-the-art outlets of cooling air, namely the transmission-tunnel and the wheel-houses. It could be observed, that there exists a linear relationship between the cooling drag and the cooling mass-flow. The gradient of the linear slope changes with different outlet positions as well as different rear-end shapes. A possible explanation of the different drag contributions could be given with total-pressure measurements in the wake of the vehicle. It could be observed that the highest total-pressure loss occurs behind the wheels.

Table 1 Cooling drag interaction of no ground simulation (CGS) and full ground simulation (WR) with closed rim design.

<table>
<thead>
<tr>
<th></th>
<th>1) CGS</th>
<th>4) WR</th>
</tr>
</thead>
<tbody>
<tr>
<td>NB STD</td>
<td>0.023</td>
<td>0.024</td>
</tr>
<tr>
<td>SB STD</td>
<td>0.019</td>
<td>0.019</td>
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Fig. 14 Influence of the ground simulation on the cooling drag.

One further measurement is shown in table (1), where the influence of the rims on the cooling drag can be interpreted. While the standard measurements of chapter 2 were performed without ground simulation and with closed rims, the influence of the opened/closed rims was investigated as well. Table (1) shows the standard measurements. For closed rims, no additional drag due to rotating wheels could be observed in this configuration, but the cooling drag itself is slightly higher on the setup without ground simulation compared to the open rims. The influence of the rims and the ground simulation will be subject of future investigations.
which is originated from the cooling-air exiting the wheelhouses. The interaction of this wheel vortex with the wake vortices is different for different rear-end shapes and therefore different cooling-drag can occur. Finally an outlook is given of force measurements with ground simulation. It could be shown that the cooling air exiting the wheelhouses has a higher cooling drag with rotating wheels compared to conventional ground simulation. It has to be pointed out that this higher cooling drag due to rotating wheels is influenced by the rim design. For closed rims, the cooling-drag does not correlate with the wheel rotation.

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REFERENCES


NOMENCLATURE

| \( A_{ref} \) | car frontal surface | m² |
| \( c_c \) | cooling coefficient | - |
| \( c_d \) | drag coefficient | - |
| \( \Delta c_{dc} \) | cooling-drag coefficient | - |
| \( c_{pTotal} \) | total-pressure coefficient | - |
| \( \Delta c_{pTotal} \) | delta total-pressure coefficient | - |
| \( \dot{m}_c \) | cooling-air mass-flow | kg/s |
| \( Re \) | Reynolds-Number | - |
| \( u_{\infty} \) | free-stream velocity | m/s |
| \( \rho \) | fluid density | kg/m³ |