ANALYSIS OF THE TEST BENCH DESIGN INFLUENCE ON THE COOLING PERFORMANCE OF A RAIL VEHICLE BRAKE DISC

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Resume

The presented article aims at assessment of the influence of test bench design on the cooling performance of a rail vehicle brake disc. The test bench is designed for experimental analyses of rail vehicles' brake components. The work includes an introduction to the solved problem, as well as a description of importance of airflow simulations in virtual environment. Creation of a geometrical model and its most important features are described. The research is performed by means of the computational fluid dynamics. For this method, it is important to set-up the boundary conditions, which are presented in the next part of the article. The results of the computational fluid dynamics simulations are presented for the solid disc and for the ventilated disc. The simulations of the solid disc showed the better convergence of calculations for various time-steps in comparison to the ventilated disc.

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1 Introduction

Safety is an extremely important point of view during design, development or any other lifecycle stage of all the types of vehicles. Concerning safety, braking plays vital role. This is even more important for rail vehicles, as these can operate at higher velocities and with higher masses than cars. Even though the rail vehicles can be equipped with brakes of multiple types, a frictional brake should always be present. One type of frictional brakes is the disc brake, now widely used for all the high-speed trains, as well as for passenger wagons of casual trains.

To ensure proper functionality during operation, certification procedure is standardized in UIC (International railway union) leaflets [1-2]. These certification procedures aim to provide set of simulations of certain conditions for different types of brakes. Another leaflet [3] regulates certification requirements for the test benches themselves. Requirements for performance is present in that leaflet, however, no exact design is required by it. As a consequence, various design solutions for various test benches might satisfy the same performance requirements [4-5]. An integral part of the test bench is its ventilation subsystem that provides air from environment with satisfying parameters. This article discusses a particular solution of a ventilation subsystem installed on the test bench. Each design might influence the experiments differently, so in the leaflet, there is a standardized way to check that the cooling performance of the ventilation system is similar enough throughout all the test benches and results are comparable [6-8].

There are several ways of how the design of the ventilation system can affect the cooling performance of the brake disc. First, with the total mass flow around the brake disc (or pads), the velocity of the air changes and so do the parameters of forced convection. Moreover, the parameters of the flow (e.g. temperature) have serious impact, as well. However, these influences are subject to the test described in the leaflet, so after successful certification, there is no need to discuss them further [9-10].

One of the ways the airflow can affect the cooling of the brake disc, is that the flow direction on the local scale (turbulences, eddies) can have some impact on results of the experiment by the means of providing slightly different airflow on opposite friction surfaces of the disc. To assess this influence, the computational fluid dynamics (CFD) can help to isolate this effect from others on the disc (for example inevitable geometry imperfections in production, material imperfections, test bench geometry imperfections etc.). As a numerical method, a CFD simulation is suitable for observation and evaluation of concomitant effects, as suggested by many authors, for example [11-14]. The best metric to compare the results of the simulations seems to be the same as is used for experiments - averaging temperature on the friction surfaces of the disc [15-17].

From the above mentioned, the following two objectives arose. The first and main objective is to assess the influence of a test bench geometry, namely the inlet pipe's shape, on the cooling performance on the brake disc via the CFD simulation. The secondary objective is a side objective to roughly assess the maximum time step size for such simulations, so that the solution can converge and provide acceptable results.

2 Model creation, geometry

The room of the test bench consists of four main parts. An inlet pipe serves as the source of cooling air driven from outside the building to the brake mechanism. In the beginning of the pipe is a primary fan, which is regulated to maintain the proper velocity at the outlet of the inlet pipe. However, this fan is incapable of providing enough mass-flow for the highest velocities, and so in the final part of the inlet pipe, the secondary fan is installed, which is not regulated and is active only at simulated train speed of 160 $km \cdot h^{-1}$ and above.

The second and the most important part is the test bench itself. It consists of the frame of the brake, the braking mechanism, bearings etc. The airflow is directed towards the brake disc and further direction is influenced by the disc and the braking conditions. The third part is the outlet pipe located behind the frame (flow-wise) and provides suction power to remove products of braking (heat and solid dust particles). The remaining part is a moveable covering. From airflow point of view, this covering separates the test bench

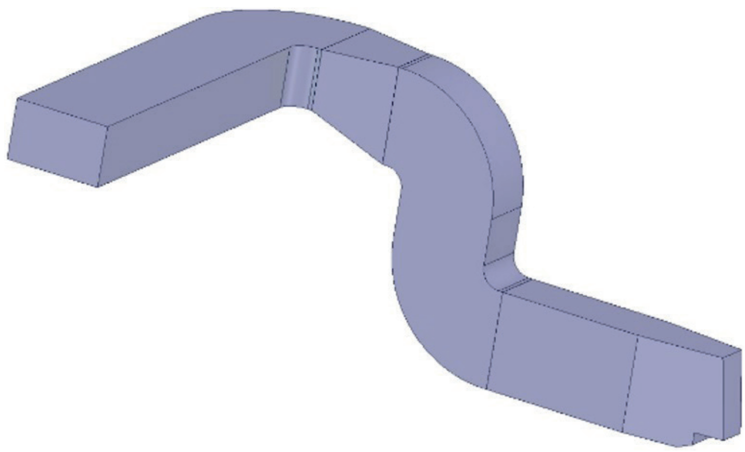


Figure 1 *A geometry for the CFD simulations - an inlet pipe*

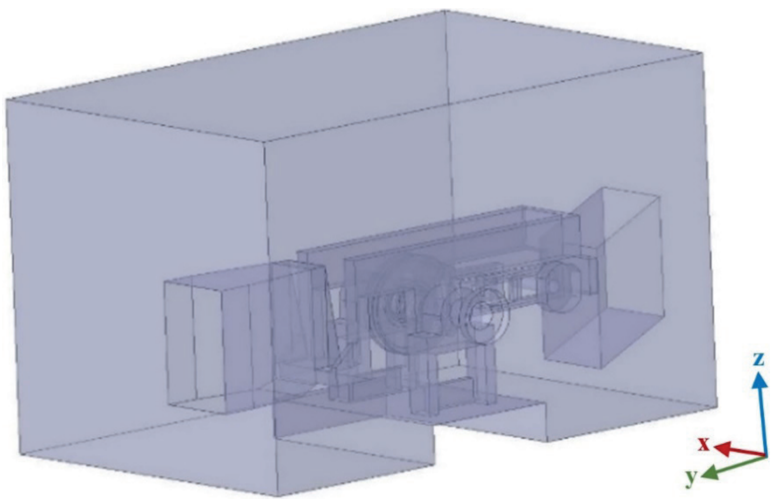


Figure 2 *A geometry for the CFD simulations - a test bench*

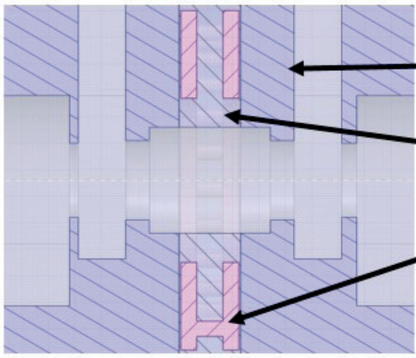


Figure 3 *Bodies in a model for the "sliding meshes" simulation*

Table 1 *Boundary conditions for simulation of airflow in the inlet pipe*

Boundary condition / setting	Value
Material in the domain	Air, constant density 1.225 $kg \cdot m^{-3}$
Mass flow inlet - primary fan	2 $kg \cdot s^{-1}$
Pressure outlet	0 Pa

Table 2 *Boundary conditions for simulation of airflow around the test bench*

Boundary condition	Value
Material - stationary + rotating domain	Air, constant density 1.225 $kg \cdot m^{-3}$
Angular velocity of rotating domain	357 rpm (equivalent to simulated train speed 60 $km \cdot h^{-1}$)
Velocity inlet	Velocity profile obtained by previous simulation
Suction - mass flow outlet	3.87 $kg \cdot s^{-1}$
Pressure outlet - floor	0 Pa
All walls	No roughness, stationary relative to adjacent cell zone
Time step size	0.00047 s (cca. 1° of rotation)

environment from the rest of the room and thus making the space out of the covering aerodynamically irrelevant for test bench.

The exact CAD model was created from the first three parts and this model was heavily simplified for the purpose of CFD simulations [18-20]. However, even after these simplifications, the computational costs of this problematics remained relatively high. The CAD model was then converted into a CFD model - enclosure of the bodies, while keeping the size of the enclosure same as the size of the real covering.

Various discs are mounted in the brake test bench, so two different simplified discs have been examined in simulations. The simplest disc is solid with a groove and the second is simplified brake disc with radial vanes. These two discs represent the edge cases for effectivity of cooling from ventilation point of view.

For the selected method of simulating the disc rotation - "sliding meshes" - another body had to be created inside the domain. This body needs to cylindrically enclose the disc.

The basic components of an analysed CFD simulation model are shown in Figures 1 to 3. Figure 1 depicts a geometry of an inlet pipe. A geometry of the test bench, which is used for the simulation computation, is shown in Figure 2. Figure 3 depicts a cross-section view of the solid disc model with individual components.

3 Simulation settings, boundary conditions and mesh

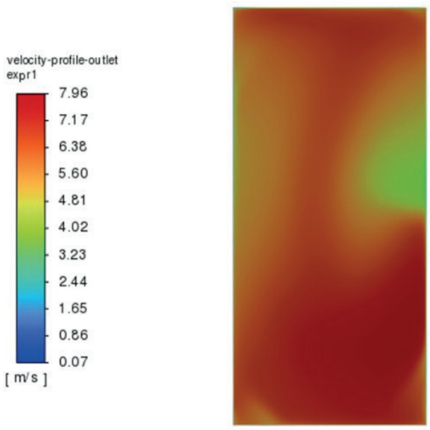
The airflow analysis was split between the simulation of airflow around the test bench and the simulation of airflow in the inlet pipe. For the latter one, the domain consisted only of the first part of the geometry (inlet pipe). The goal of this sub-simulation is only to find the velocity profile at the outlet of the inlet pipe. Settings and boundary conditions for this sub-simulation are presented in Table 1.

Due to the complexity of the problem, both brake discs were simulated in a single constant regime. Rotation was simulated using the principle of sliding meshes with angular velocity of rotating domain simulating particular train velocity. Settings and boundary conditions of this regime are summarized in Table 2 and the thermal properties for cooling are summarized in Table 3.

The mesh used in all bodies was made using the Mosaic meshing in Ansys Fluent Meshing, the

***Table 3** Thermal properties and time-dependent simulation settings*

Property	Body	Value
Density	Air	1.225 kg·m^{-3}
	Disc	8030 kg·m^{-3}
Thermal conductivity	Air	0.0242 W·m^{-1}·K^{-1}
	Disc	40 W·m^{-1}·K^{-1}
Specific heat	Air	1006.43 J·kg^{-1}·K^{-1}
	Disc	502.48 J·kg^{-1}·K^{-1}
Initial temperature (homogenous throughout the body)	Air	288 K
	Disc	600 K



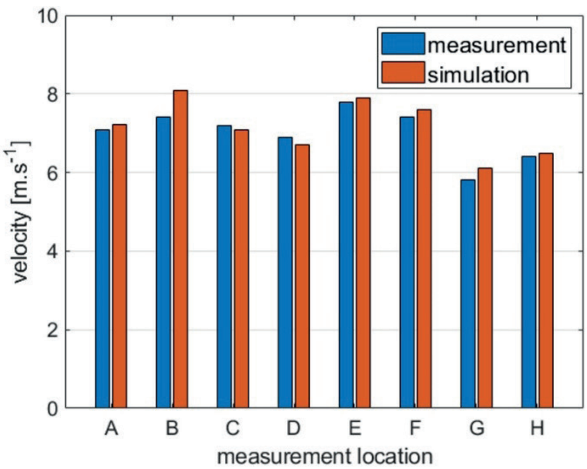


***Figure 4** A velocity profile on the outlet of the inlet pipe (left), the comparison of the measured vs. simulated values (right)*

final mesh contained polyhedral prisms near the wall and hexahedral elements farther from the wall with polyhedral buffer layers.

4 Results and discussion

From the simulation of airflow in the inlet pipe, the velocity profile is obtained at the interface between the inlet pipe and the main stationary domain. This velocity profile is presented in Figure 4.

A measurement was performed to prove the results of this simulation. A total of eight points were selected (and spaced almost equally) near the inner perimeter of the profile and velocity of flowing air has been measured using simple handheld anemometer. A comparison of measured and simulated values can be found in Figure 4. Generally, the correlation between them is good and the velocity profile is accepted as an input to the main simulation.

During the real measurements, the temperature is measured 1 mm under the frictional surfaces, in three locations on both sides of the disc and the temperature is averaged between those three sensors. In these simulations, temperature is averaged on the whole surface of friction.

As for the minor objective - the minimum applicable time step size - for solid disc, the disc temperature contours after one minute of cooling are presented in Figure 5. The gradient in the tangential direction in the middle of the friction surface is clearly visible in the contours created with time step size 1 s and 0.5 s but are not present when the time step size is 0.05 s. Therefore, the time step 0.05 s can be considered as the maximum time step size for model with solid disc. Figure 5 shows an example of the simulation computations results of a solid disc. It includes three various temperature distributions in the solid disc after the 1st minute. Figure 6 depicts the results of simulation computation for the ventilated brake disc. As the numerical simulation was successful only for the time step of 0.005 s, it includes only one illustration of temperature distribution in the disc body.

On the other hand, the model with ventilated disc has the only single time step size successfully simulated - 0.005 s. All the other (bigger) time steps showed poor convergence or no convergence at all. No significant tangential gradients are present in this case, as can be seen in Figure 6. For ventilated disc, its aerodynamic drag was used as an additional criterion of convergence and was in good proximity with values presented by [17]. In addition, the results were in good accordance with the rig tests described by [10].

To compare the influence of turbulent and swirling

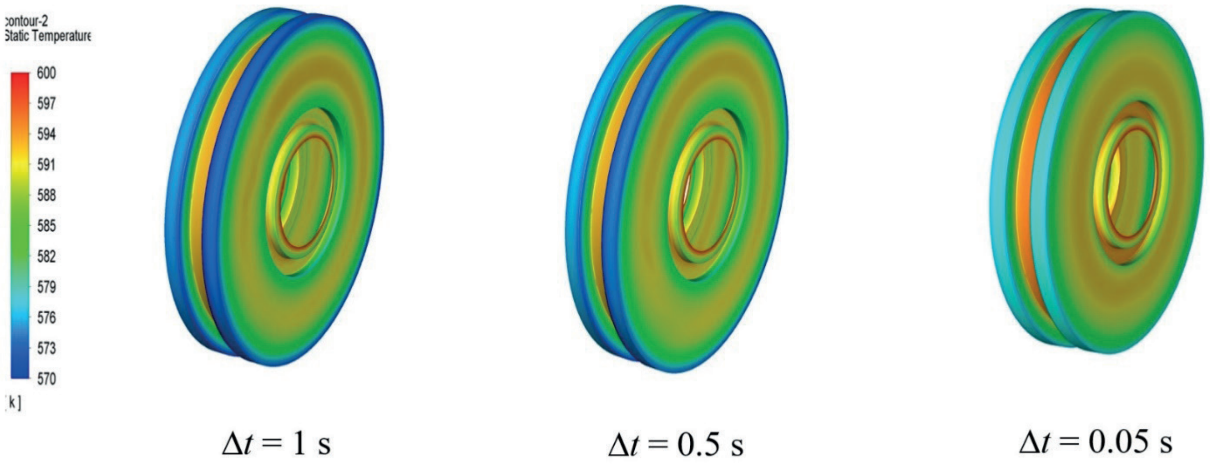


Figure 5 *Temperature contours on the surface of a solid disc at different time steps after the 1st minute*

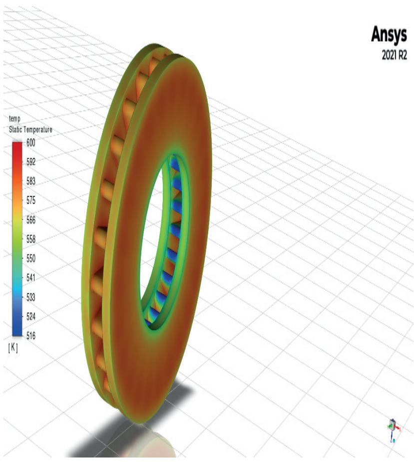


Figure 6 *Temperature contours on the surface of a ventilated disc for the time steps 0.005 s after the 1st minute*

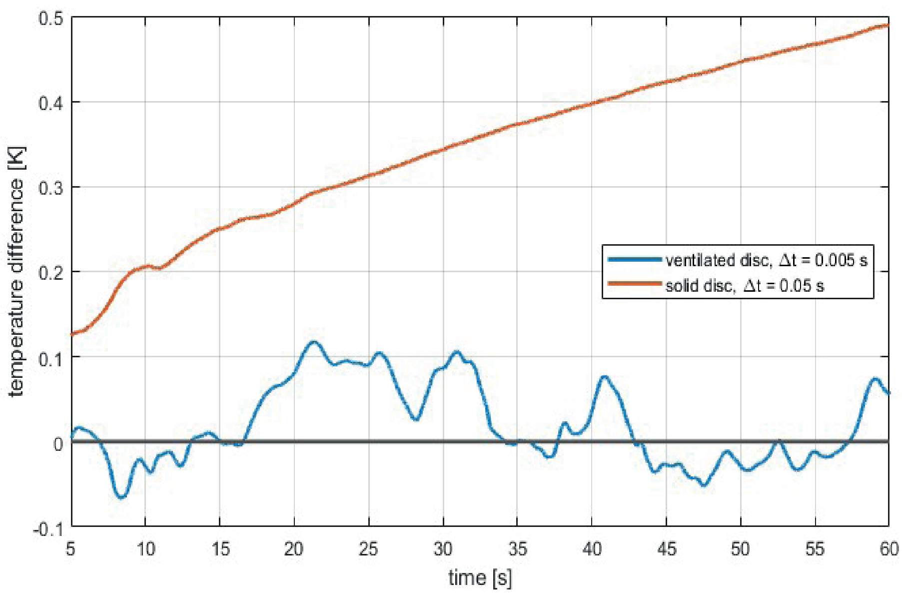


Figure 7 *Evolution of temperature differences between the friction surfaces of the brake discs in terms of time*

flow from the inlet pipe on the cooling performance of the either of the brake discs, curves for both successful cases are presented in Figure 7. The two curves are depicted - for the solid disc (a red curve) and for the ventilated disc (a blue curve). For the non-ventilated disc, the trend is clearly visible, as the difference of temperatures on frictional surfaces increases in time. Contrary to the solid disc, the curve for ventilated disc shows neither increasing nor any other trend in time.

5 Conclusions

From the results, for the goals stated at the end of the introductory part, following statements can be concluded:

- For the non-ventilated disc, the maximum allowable time step is (in simulated case) 0.05 s (cca 10 degree of revolution), whereas for ventilated disc, this value is 0.005 s (cca 1 degree of revolution). Bigger time steps caused the simulation either to diverge or to provide unrealistic results.
- The airflow provided by the inlet pipe causes different sides of the solid disc to be cooled at different pace, while the ventilated disc makes these differences diminish and no trend was observed over time.
- Combining the previous statements, simulations with ventilated disc can be heavily simplified by removing most of the test bench geometry, as it shows little to no influence on its cooling performance.

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Conflicts of interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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