Influence of asymmetric airflow on the cooling performance of brake disc of railway vehicle

V Pavelčík¹, Y Fomína¹

¹ Department of transport and handling machines, University of Zilina, Univerzitna 1, 010 26 Zilina, Slovakia

vladimir.pavelcik@fstroj.uniza.sk

Abstract. The main objective of this article is to discuss the influence of asymmetric airflow on the cooling performance of the brake disc. This asymmetry might be caused by the air inlet pipe curvature of the UIC test bench for frictional components, whereas in reality this asymmetry might be caused by various factors. In the first part of the paper, a brief summary of effects of heat and temperature changes on the brake disc is presented. Additional information about UIC test bench is given with the focus on cooling effects of airflow on the disc. In the second part, a CFD simulation was made, and its settings and results are presented in detail. In the last part, temperature differences in various time points and positions on the brake disc are compared graphically in order to assess the influence of airflow from the inlet pipe and its curvature on the cooling performance of the brake disc.

Introduction

When it comes to vehicles, safety is one of the highest priorities regardless of the type of the vehicle. Generally, brakes are one of the most important safety components. Various types of brakes are used for various purposes. The most common type of brakes in railway vehicles (as well as automotive) is frictional brakes. These brakes are designed to convert kinetic energy of the vehicle to heat using friction force. The heat is then conducted away from the contact area (frictional surface) into the solid body (brake disc or wheel) and is further dissipated into environment via convection forced by flow of cooler air.

Disc brakes have their limitations. First, their braking power is constrained by adhesion between wheels of the vehicle and road/railway. Second, the heat must be dissipated into the environment via convection effectively, so that no spot in the body of brake disc/pad is heated above the critical temperature, which may cause cracks and failures. More about effects of heating on the brake disc is presented in the first chapter.

As a result of being important safety component, all parts of brake system must undergo certification process before being admitted to operation. [15] There are various tests prescribed in UIC (international railway union) leaflets 541-3 [1] for disc brakes and 541-4 [2] for block brakes. These leaflets, however, assume that the tests are performed on certified test bench satisfying requirements given in UIC leaflet 548 [3]. In the second chapter, the ventilation system requirements of such test bench are described in detail.
In the simulation part, symmetric and asymmetric airflow and their effects on cooling of brake disc is investigated. CFD simulations in Ansys Fluent have been performed and results are presented, mainly graphically.

1. Braking

The main purpose of disc brake is to convert kinetic energy of a vehicle to heat. The kinetic energy $E_k$ can be calculated as a sum of kinetic energy of translational ($E_{kt}$) and rotational ($E_{kr}$) motion:

$$E_k = E_{kt} + E_{kr} = \sum \frac{1}{2} m_i v_i^2 + \sum \frac{1}{2} I_i \omega_i^2;$$

(1)

where $m_i$ are masses in translational motion, $v$ is the velocity of translational motion, $I_i$ is the moment of inertia of i-th rotating component and $\omega_i$ is the angular velocity of corresponding rotating component. However, the energy is decreasing in time, so converting this problem to time-dependent is advantageous. The braking power can be calculated also from easy-to-measure quantities – relative velocity between pad and disc $v_v(t)$ and braking force $F_p$. With known friction coefficient $f$ and disc and pad dimensions, the heat generated during braking can be expressed according to Zhoujun [4]:

$$q(r, \tau) = f \cdot \frac{F_p}{\pi (r_{outer}^2 - r_{inner}^2)} \cdot \frac{v_v(t)}{R_d} \cdot r.$$

(2)

Because pads and discs are made from different materials, the generated heat is not divided equally in these components. The part of heat taken by disc can be calculated from the following equation [4]:

$$\xi = \frac{q_d}{q_d + q_p} = \frac{1}{1 + \frac{\rho_p c_p k_p}{\rho_d c_d k_d}};$$

(3)

where $k$ is thermal conductivity, $c$ is specific heat and $\rho$ is density of the material, with indexes $p$ and $d$ for pad and disc respectively. This approach is simplified, and the ratio is not constant in time [5].

Many studies have been published discussing methods to assess the temperature fields in brake discs during braking. For example, in the article [6], the author compares numerical and analytical calculation of the disc temperature. Duzgun [7] compares heat dissipation abilities of solid disc and different types of ventilated discs experimentally and numerically. Here, ventilated brake discs are proven to have better heat dissipation abilities, although they are more prone to stress concentration caused by temperature differences in the disc’s body. Study also [4] shows interesting fact, that other studies used constant heat transfer coefficient on the whole surface of the disc [8] or calculated this coefficient empirically for each time step in simulations [7,9,10]. However, none of those studies has considered that this coefficient varies across the surface of the disc in both time and space.

Important effect of thermal load on the disc is deformation. Heat concentrates in some specific areas of the disc determined by geometry, production imperfections and wear. Three most adverse situations of temperature fields are presented in the Figure 1.

![Figure 1. Types of hot spots on the brake disc (acquired using thermovision) [12]](image-url)
In the left side of the Figure 1., the first type is presented – local thermoelastic instabilities caused by uneven frictional force, wear, etc. These hot spots occur above critical relative velocity. In the middle of this figure., hot bands can be clearly identified as a result of further decreasing the contact area between disc and pad. Study [11] presents computational model for this case and predicts creation and evolution of the hot bands. The last type presents locations of big temperature gradients regularly placed around the disc. These hot spots are the main cause of disc failures, because higher temperature here causes material changes (to martensite) The creation and evolution are not the same for all cases, and the mechanism is described in the study [12].

There are more not negligible effects of braking on the braking components, for instance acoustic effects or solid particles emissions. These components must withstand all such conditions (thermal, mechanical stress, vibrations, etc.) during operation. This is ensured by certification process performed on certified test benches.

2. UIC test bench and its ventilation system
As has been mentioned before, UIC leaflet summarizes requirements test benches must meet in order to be certified and able to perform certification tests. In this chapter, some of those requirements are presented. The test bench has to satisfy a lot of criteria and it is wise to verify all the processes [17] and their effectivity using experiments [13,14,16] or various simulation types, for example structural analysis of various parts of test bench itself or tested components [18,19,20,21], dynamic [22,23] and modal analysis of the test bench as a whole or the vehicle’s components [24,25] and/or CFD simulation to investigate airflow, heat transfer and solid particles. Information in this chapter is then important for the CFD simulation setup part.

Every test bench has different speed limit (simulated speed of train calculated from revolutions taking into account wheel diameter). The test bench can simulate various braked mass, and this can be done using purely mechanical gyrating masses or using drive motor to simulate gyrating masses. Lastly, contact force perpendicular to friction surfaces can be adjusted in a range of values. Different braking regimes are then simulated via certain combination of these factors.

For certification purposes, contact force and tangential force are measured and friction coefficient is calculated for evaluation purposes. In addition to friction coefficient, temperature is measured and evaluated. For these measurements, six measuring points are located on the disc, three on each side 120° apart, as shown in the Figure 2.

![Figure 2. Points of temperature measurement according to UIC leaflet [1]](image-url)

Various test benches are assembled in various conditions and therefore might have different numbers, shapes and dimensions of inlet and outlet air ventilation pipes. For disc brakes, an axle-mounted brake disc is heated by a drag-brake application. Parameters of the test are summarized in the Table 1. The cooling curve is recorded from the thermoelements, and a cooling constant is calculated from it. This
test is repeated twice for different cooling air speeds. In order to accept the ventilation system as satisfactory, these cooling constants must be in range defined by the leaflet.

**Table 1. Test settings**

| Setting                        | Value                                      |
|--------------------------------|--------------------------------------------|
| Running speed                  | 60 km·h⁻¹, simulating wheel diameter 890 mm |
| Braking power                  | 25 kW                                      |
| Drag braking duration           | 30 minutes                                 |
| Cooling air speed, Test 1      | 30 km·h⁻¹                                  |
| Cooling air speed, Test 2      | 60 km·h⁻¹                                  |
| Brake disc                     | Knorr 640x80, cast steel                   |
| Brake pad                      | Cofren T30S                                |

From the leaflet, it is unclear, where the cooling air speed should be measured, and so the velocity will be a subject to simplifications at CFD simulation setup.

The 3D model of the test bench is shown in the figure 3. There, ‘1’ is the inlet pipe, where air enters the test bench’s environment, ‘4’ is the outlet pipe, ‘2’ is a box enclosing the test bench and ‘3’ is the test bench itself, for the purposes of CFD simulations simplified with brake pads, gyrating masses and other parts of the brake removed. Because no suction is mentioned in the leaflet and no effect of the curvature of the outlet pipe is expected, the outlet pipe is suppressed for the simulation.

![Figure 3. Geometry of the test bench used for simulations](image)

3. **CFD simulations setup**

The CFD simulations have been made in Ansys Fluent software [26] in order to assess the influence of the inlet pipe curvature on the cooling performance of the brake disc on this particular test bench. First, a reference plane has been selected (located at the point where inlet pipe color changes from red to blue). This reference plane cuts the inlet pipe into two parts. In the first simulation, the inlet pipe before this plane (from the airflow point of view) was suppressed and the cooling air speed of 60 km·h⁻¹ was applied as normal to this plane. This case is considered as ‘symmetric’ since the brake disc middle plane is coincident with the middle plane of the rest of the inlet pipe and the airflow inlet velocity has the same magnitude and direction vectors on both sides.

In the second simulation, the ‘cut’ inlet pipe (ending at the reference plane) was investigated. The inlet velocity was set, so that the total mass flow rate was the same as in the first case. The outlet velocity profile was taken as inlet velocity profile in the third simulation, which had the same settings as the first one, except for the inlet velocity. In this case, the inlet velocity reflected the curvature of the inlet pipe and is thus considered ‘asymmetric’.

Both simulations were done only for a single cooling air speed 30 km·h⁻¹. Both cases are repeated for solid brake disc and for very simplified ventilated brake disc (simplified due to computational cost reduction). In all situations, heating is neglected, and initial temperature of 600 K is applied on the whole body of the disc. The reason for this is high complexity of the problematics as described above. Material
properties are kept constant throughout all simulations and are summarized in the Table 2. Rotation is simulated using sliding mesh interface and mesh motion. An overview of boundary and cell zone conditions is presented in the Table 3. Mesh parameters are summarized in the Table 4.

**Table 2. Material properties**

| Property             | Brake disc | Cooling Air |
|----------------------|------------|-------------|
| Density [kg.m\(^{-3}\)] | 7500       | 1.225       |
| Thermal conductivity [W.m\(^{-1}\).K\(^{-1}\)] | 50         | 2.42\(e\)-2 |
| Specific heat [J.kg\(^{-1}\).K\(^{-1}\)] | 502        | 1006        |
| Viscosity [kg.m\(^{-1}\).s\(^{-1}\)] | -          | 1.7894\(e\)-5 |

**Table 3. Boundary conditions**

| Boundary condition       | Case ‘symmetry’ | Case ‘asymmetry’ |
|--------------------------|-----------------|-----------------|
| Velocity inlet           | 30 km\(h\)^{-1}\* | Varying\*       |
| Pressure outlet           | 0 Pa            | 0 Pa            |
| Disc initial temperature | 600 K           | 600 K           |
| Simulated speed of train (mesh motion) | 60 km\(h\)^{-1} | 60 km\(h\)^{-1} |

*Total mass flow rate of cooling air was kept same in both cases

**Table 4. Mesh parameters and quality**

| Parameter                  | Ventilated disc | Solid disc |
|----------------------------|-----------------|------------|
| Type of mesh               | Polyhedral      | Polyhedral |
| Number of elements         | ca 2.7\(\times\)10^6 | ca 1.5\(\times\)10^6 |
| Min. orthogonal quality    | 0.277           | 0.274      |

\(y^+\) criterion satisfied according to requirements of k-\(\omega\) SST turbulence model

4. Results

In the Figure 4, the streamlines of the airflow in the inlet pipe are shown. The flow direction turns twice at 90\(^\circ\) and this creates two areas with slower, more turbulent flow. It can be clearly seen that the straight end section of the pipe is too short, and the flow is not stabilized at the end of it. The velocity at the outlet of this pipe is then used as an inlet boundary condition for the ‘asymmetry’ case. The velocity contours at the inlet for both cases can be compared in the Figure 5. There, on the left side, the inlet boundary condition for ‘symmetry’ case is shown, with the red area of velocity magnitude according to Table 1 (velocity decreasing near wall due to no-slip condition on the pipe wall). On the right side, the airflow is influenced by the pipe curvature strongly and the velocity throughout the profile varies rapidly.
Figure 5. Velocity contours (inlet boundary condition) for ‘symmetry’ (left) and ‘asymmetry’ (right)

In the following figures, the influence of asymmetric airflow on the cooling of friction surfaces is shown for the braking regime described in the previous chapters. First, in the Figure 6. The cooling curves measured on the solid brake disc are shown for both cases and for both sides of the disc (from the airflow point of view). The difference between symmetry and asymmetry case is clearly visible after 240 seconds of cooling, however, the difference is of only negligible importance for the accuracy of real experiment. In the Figure 7., the same comparison is presented for very simplified ventilated brake disc. Here, the distance between single curves is even smaller.

Figure 6. Cooling curves – solid disc

Figure 7. Cooling curves – simplified ventilated brake disc
In the next two graphs, the average temperature of the friction surfaces as a function of radial distance from the rotation axis is shown. In the Figure 8., these curves on solid disc are presented in three certain time points. While both curves representing symmetry case are similar in all time steps, the curves representing asymmetry case show bigger difference. However, majority of the ‘asymmetry’ curves’ length lies under those of ‘symmetry’ case, so a slightly better cooling performance is expected there. In the Figure 9., the same is presented for simplified ventilated disc. Here, all curves in a single time step are very similar in both shape and values. For asymmetry case, the difference between friction surfaces is bigger than for symmetry case. These differences are subtle from the cooling process point of view, though.

![Figure 8. Temperature as a function of radial distance in different time steps – solid disc](image)

![Figure 9. Temperature as a function of radial distance in different time steps – simplified ventilated disc](image)
Conclusion
Brake components heating and cooling is extremely complex problematics, however, solutions to these issues are very important by the means of their proper functionality. Testing and certification of brake components is vital for safety of railway vehicles. Test benches designed for such purposes must satisfy requirements given in the standardization leaflet to simulate the conditions correctly.

From the results of the CFD simulations, a conclusion can be made that for this particular test bench, its ventilation system and more specifically its inlet pipe’s curvature influence on the cooling performance is only weak. The biggest differences are observed while braking with solid brake disc, however, only minority of vehicles in real operation is equipped with solid discs. The ventilation effect of the simplified ventilated disc modifies the airflow around the test bench in a way that most of the differences between right and left side (from the airflow point of view) diminish, so a prediction can be made that non-simplified ventilated disc would cause these diminishing to be even stronger, the side differences even lower and the cooling performance will be less influenced by asymmetry of the airflow.

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