Conference or Workshop Item:
Limmer, F, Barton, D, Brooks, P et al. (Accepted: 2021) Development of a Small-scale Test Bench for Investigating the Tribology and Emission Behaviour of Novel Brake Friction Couples. In: Eurobrake 2021, 17-21 May 2021, Online. (Unpublished)

SUBMISSION NUMBER (EB20XX-XXX-XXX)

Development of a Small-Scale Test Bench for Investigating the Tribology and Emission Behaviour of Novel Brake Friction Couples

Fabian Limmer* [1], David Barton[1], Peter Brooks[1], Carl Gilkeson[1], Shahriar Kosarieh[1]

[1]*University of Leeds, School of Mechanical Engineering, Woodhouse Lane, Leeds LS2 9JT, United Kingdom (E-mail: mnfli@leeds.ac.uk)*

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ABSTRACT: Testing of brake friction couples is a very costly, energy and time-consuming process, that only allows for a very limited range of material concepts to be considered. Small-scale testing offers great advantages in these areas but often can't be applied due to limitations in operating conditions and comparability of the results to full-size systems. A novel small-scale test bench has been developed at the University of Leeds which aims to provide the ability to screen friction materials under realistic braking conditions. This study reports on the development process of some of the key features of the test bench. Details of the scaling approach that was used for the sizing of the friction couple are described. Different controlling strategies are presented which can be applied to replicate a variety of realistic braking scenarios. The test bench also incorporates an isokinetic sampling system to measure airborne particulate matter (PM). Some details on the CFD guided development process of this PM capture system are presented.

KEY WORDS: Small-scale testing, real-world driving cycle, emission measurement, friction material screening, tribology

1. INTRODUCTION

The development of friction brakes is a complex process with ever changing demands. Starting over 100 years ago with rising vehicle weights and travelling speeds that left a need for improvement of the brake systems performance parameters such as the friction coefficient and its stability at elevated temperatures. Additional requirements now include acoustic considerations, with quiet operation becoming increasingly important, further adding to the demand portfolio. Recently, a popular area for development is for lighter, corrosion-resistant and more eco-friendly brake systems [1]. Apart from health and environmental issues, the main drivers for this development are the changing load profiles arising from the megatrends of electrification and autonomous driving. These new demands are again creating a race to finding solutions as well as adding additional evaluation criteria such as particulate emissions.

Since testing and development of new brake designs and materials on vehicles is very time consuming, costly and to some extent dangerous, test benches were created to assess the performance of brake systems as early as the 1890s where water-driven dynamometers were used to test new friction materials [2]. Several attempts have been made over the years to design test benches that utilize scaled-down brake systems to further cut down on cost, time, and complexity. Due to limitations in their operating flexibility and lack of comparability of the results to full-sized brake systems, small scale test benches have traditionally been limited to general research purposes.

This paper reports the development of a novel small-scale test bench system that aims to replicate real braking conditions and allows for measurement and analysis of particulate matter emissions, generated at the friction interface.

2. SCALING APPROACH

One of the most important aspects when designing a scaled test bench is the size and geometry of the friction couple. The overall goal is to match any operating conditions of a full-sized brake system as closely as possible. Past research has shown that scaling based on a constant energy density approach delivers the best results [3]. A scaling methodology is normally applied and it introduces a fixed scaling factor based on one chosen scaling parameter. The approach to keep the input energy per unit of nominal contact area constant is usually achieved by scaling the brake system via the brake pad friction area A_{pad} (mm²), with A'_{pad} being the area of the scaled pad. This allows the introduction of a dimensionless scaling factor, f, which is then used to scale other parameters as well. This is given by:

$$f = \frac{A_{pad}}{A'_{pad}} \tag{1}$$

Figure 1 shows the surface geometry of the pad and the friction ring of the disc from a full-sized brake system (from a Ford Focus) chosen for the scaling exercise.

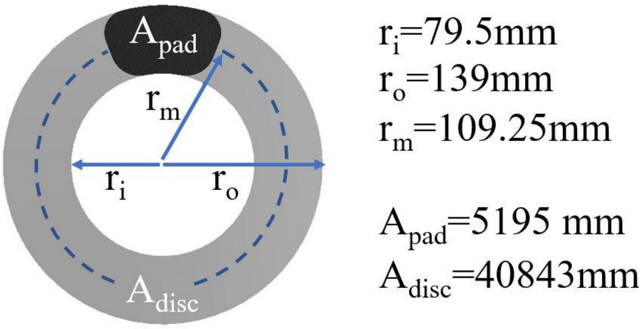

Figure 1: Geometric parameters of a typical full-sized brake system.

The purpose is to duplicate the conditions existing inside the pad/disc sliding interface of this full-sized brake system as closely as possible so that the same friction phenomena occur in the scaled system. Three geometric features that affect the sliding conditions of the brake interface were chosen to guide the scaling process.

The first feature is the ratio, R_A, of the friction surface area of disc to pad, namely:

$$R_A = \frac{A_{disc}}{A_{pad}} \tag{2}$$

This ratio defines the area where the friction is generated (and therefore where the heat is produced), in relation to the area where this heat can be dissipated (i.e. the disc) into the environment; this aspect has been shown to be important for the thermal response of a brake system [4]. Another important geometric feature of a brake system is the sliding velocity gradient along the friction radius. The velocity $v_s(r)$ is greatest at the outer radius, r_o (m), and lowest at the inner radius, r_i, as depicted in Figure 2. Accordingly, this causes the power input to be greater at the outer radius, assuming that the full pad area is in contact with the disc and a uniform braking pressure is applied.

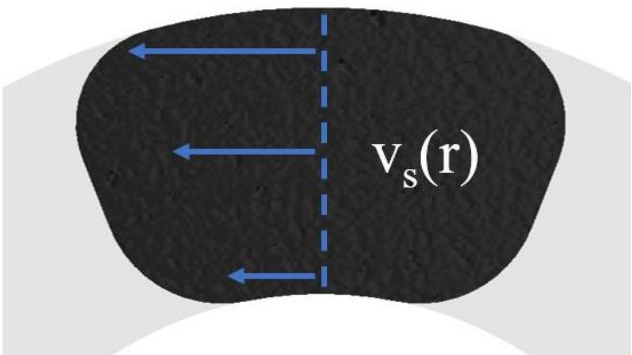

Figure 2: Velocity gradient along the sliding radius.

Therefore, the sliding velocity ratio, R_v, can be determined from the sliding radiuses for the full-sized brake system through:

$$R_v = \frac{r_o}{r_i} \tag{3}$$

The third geometric feature that has shown to affect the thermal response of a brake system is the pad aspect ratio, which is defined as the width to length ratio of the brake pad itself [5]. While the width of the pad determines the velocity gradient throughout the radial direction of the sliding interface, the length of the brake pad determines the angular coverage of the disc friction track and therefore the change in the velocities direction throughout the circumference of the sliding interface (Figure 3).

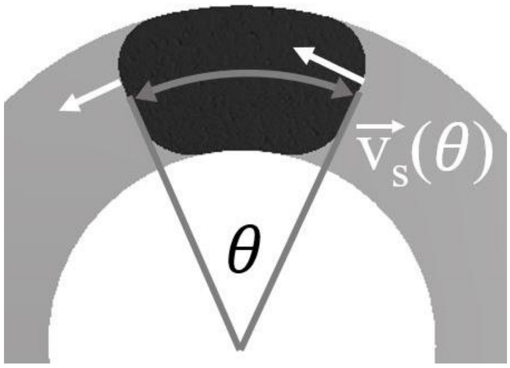

Figure 3: Velocity direction along the sliding circumference.

While the angular coverage is relatively constant over the sliding radius for the full-sized brake system due to the shape of the pad, the pad of the small-scaled version will have a rectangular shape for practical reasons. The angular coverage will therefore vary from the inner sliding radius to the outer sliding radius on the small-scale system and the angulare coverage at the mean sliding radius will be considered for scaling.

The device which serves as a basis for the test bench is the Bruker UMT TriboLab arranged in a pin-on-disc type configuration, equipped with a rotating drive at the bottom and a movable carriage at the top, see Figure 4. Two important limitations are given by the device specifications. The first one is the maximum torque output from the rotating drive's electric motor of 5Nm and the second is the maximum brake disc diameter of 95mm.

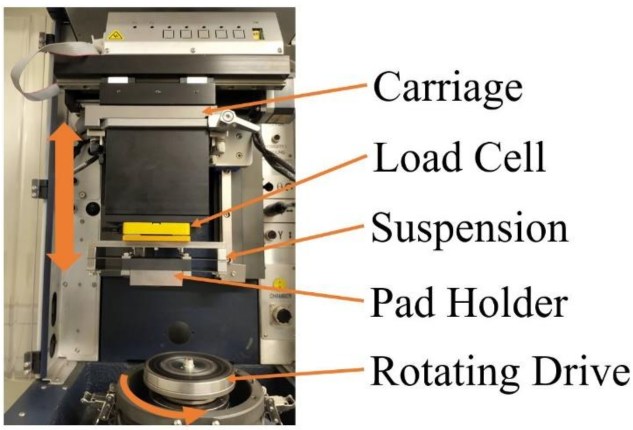

Figure 4: Bruker UMT configuration.

The size of the brake pad, as well as the mean sliding radius for the scaled test bench, were determined iteratively by matching the friction surface area ratio, R_A, the sliding velocity ratio, R_v (radial velocity gradient) and the angular coverage of the friction interface over the friction track, θ, within the limitations of the device. The brake pad area was calculated to be 320mm² with a length of 20mm and a width of 16mm. The mean sliding radius was calculated to be 29mm. It is worth noting at this stage that the brake pad of the full-sized system is chamfered at the leading and trailing edges. This means that the friction area of the brake pad will increase during its lifespan and the scaling factor (Equation 1) would increase from 11.3 up to 16.2. Applying the scaling rule,

$$r_s = r_s * \sqrt{f} \tag{3}$$

to determine the mean sliding radius would have led to a mean sliding radius between 27.1mm and 32.5mm. The surface area of the pad A_{pad}, shown in Figure 1 is the surface area of the brake pad after the chamfered edges have worn down. Since this is the surface area that was used to calculate the mean sliding radius through the geometric ratios, this implies that the mean sliding radius (29mm) is 10.8 % smaller than the mean sliding radius (32.5mm) that would be given through applying the scaling rule. Table 1 shows the three geometric parameters of the full-sized brake system (in the range of the brake pad being new with the chamfered edges and the brake pad worn down) and the calculated geometric parameters of the scaled system.

Table 1 Geometric features considered for scaling

	Full-Sized System	Scaled System
R_A	10.2 - 7.8	9.1
R_v	1.75	1.76
θ	37° - 49°	42°

3.CONTROL STRATEGY

The test bench can be controlled using the load cell (Figure 4) either by controlling the normal force, F_N (N), applied to the pad or by controlling the friction force, F_F. For test cycles including drag brake applications over extended periods the operating temperature is dependent on the frictional power of the system. When comparing different materials (pad or disc or both) for example, it is essential to keep the friction power and resulting energy dissipation constant for each test cycle to allow for direct comparability of the performance (wear, emissions, etc.) of the friction couples under consideration.

Figure 5 shows the results of two tests under steady-state conditions. Both tests had a duration of 600s under equal sliding speeds of 8m/s. The same friction couple was used for both tests. In the first test (Figure 5a), the normal force was controlled (held constant at 40N) and the friction force was a measured parameter resulting from the response of the friction pair. In the second test (Figure 5b), the friction force was controlled (held constant at 20N) and the normal force was the measured parameter. This example shows how the two fundamentally different control strategies can lead to different conditions in the sliding interface even if the trend of the friction coefficient is very similar in both types of tests.

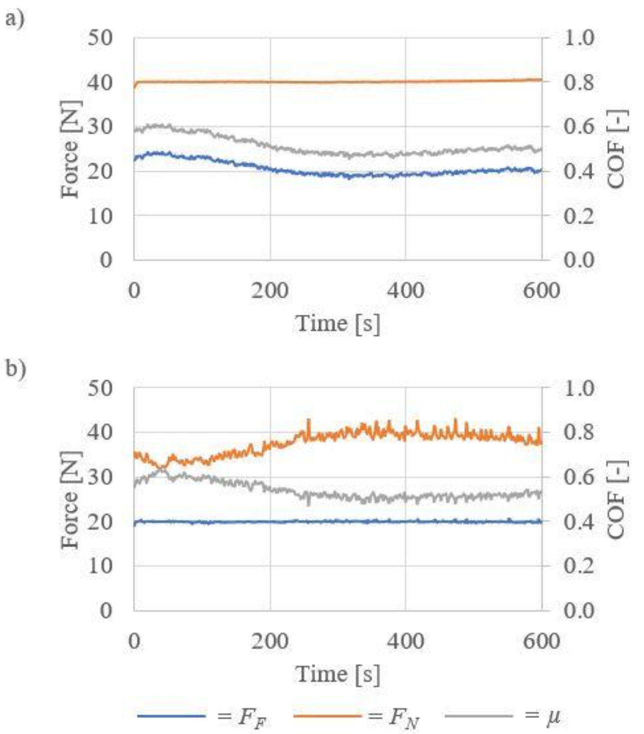

Figure 5: Steady-state tests controlling a) the normal force F_N and b) the friction force F_F.

Being able to control the friction power also enables the option to replicate more realistic braking events as given by the WLTP-based brake emission test cycle [6]. The test cycle defines a driving velocity profile of a vehicle that includes a series of braking events. The braking events are defined by the braking time s, as well as the start and end velocity, v, of the vehicle. The braking torque T_B (Nm) for one front disc brake can be calculated through:

$$T_B = \frac{a\, m\, R_{dyn}\, \phi}{2}$$

Where a (m/s²) is the mean linear deceleration given through the change in velocity by the braking time, m (kg) is the vehicle mass, R_{dyn} (m) is the dynamic rolling radius of the tire and ϕ the dimensionless brake force distribution for the front axle. The scaled setup aims to replicate one pad/disc friction interface of a front axle disc brake. The resulting friction force, F_F, for the scaled test bench can be calculated with the effective rotor radius, r_{eff}(m), of the full-sized brake system and the scaling factor, f, *using*:

$$F_F = \frac{\left(T_B/r_{eff}\right)}{2} * \frac{1}{f^3}$$

The WLTP-based brake emission test cycle is a time-controlled cycle meaning that the braking events take place at fixed time intervals. Since the scaled test bench will have different cooling rates than the full-sized system, the adapted cycle is set to execute each subsequent braking step once a desired initial disc surface temperature is reached which will also have the positive side effect of reducing the cycle time.

4.EMISSIONS TEST SYSTEM

To reliably measure and sample particulate matter originating from the brake friction couple, it is necessary for the brake system to be enclosed from the environment. This is to ensure that clean air can be supplied to the system so any particulate matter originating from other sources can be eliminated. It is also necessary for samples to be measured under isokinetic conditions. The Dekati ELPI+ impactor, which is designed for measuring airborne particulate matter, forms the basis of emission characterisation in this set-up. This device works with a sampling flow rate of 0.6m³/h through a sampling nozzle from a full-sized dynamometer. The nozzle geometry leads to a sampling airspeed of 6.87m/s. More details can be found in [7]. Some of the criteria that were considered for the design of the brake system housing and ducting can be summarised as:

- Small volume-to-flow rate ratio
- Minimal number of bends
- Smooth transitions between geometry changes

After the first housing design for the brake system was finished, the choice was made to use an inner diameter, d_i of 50mm for the supply and exhaust ducting. The housing design was adjusted so that the cross-sectional area stayed constant throughout the enclosure, thereby minimising the negative impact of blockages.

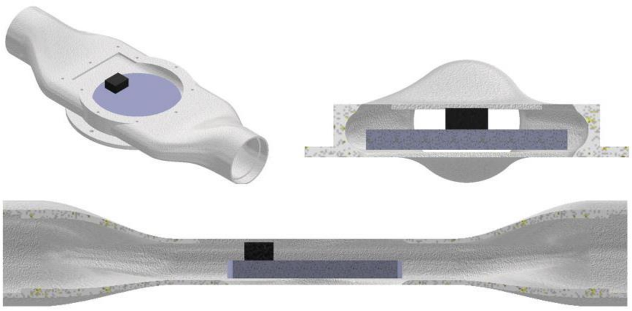

Figure 6: Brake system housing.

The final design of the housing is shown in Figure 6 together with the brake pad and disc. The housing will mount onto the outer ring of the rotary drive and seal from the environment with a lid (not shown in Figure 6) which includes a rectangular opening slightly larger than the pad holder. During operation, the gap between the pad holder and the lid will be sealed with adhesive tape to enclose the system from the environment yet still allow vertical displacement of the pad. Temperature measurement will take place using a K-type rubbing thermocouple (Therma MST-20347) that attaches to the housing lid and is in permanent contact with the disc during operation.

Following the EPA Method 1A [8] guidelines, the length of the straight sampling section of the exhaust duct was chosen to be 1000mm with the sampling plane being 500mm (10 d_i) downstream and upstream of any disturbance. With the ducting diameter of 50mm and a sampling airspeed of 6.87m/s, a volumetric flow rate of 48.5m³/h is required to meet isokinetic sampling conditions. Clean air will be supplied through an Allentown EcoFlo EFS supply blower. These units are usually used to supply air for individually ventilated animal cage systems. The blower comes equipped with a gravimetric pre-filter and a HEPA H14 filter and allows for monitoring of air temperature in the range of 5°C – 50°C (+/-0.13°C) as well as humidity in the range of 10% - 90% RH (+/-2%). The supply blower can be set to provide a constant flow rate in the range of 20m³/h to 100 m³/h. The exhaust duct will be connected to the fume extraction system of the laboratory making it a single pass system.

In order to guide the design process, Computational Fluid Dynamics (CFD) simulations, using Ansys Fluent 2020 R2, were conducted. The geometries for the simulations were extracted as the inner (air) volume from the CAD assembly which includes the housing, disc, pad and the pad holder. Figure 7 shows an example of the extracted volume (Figure 7b) for the final housing design (Figure 7a).

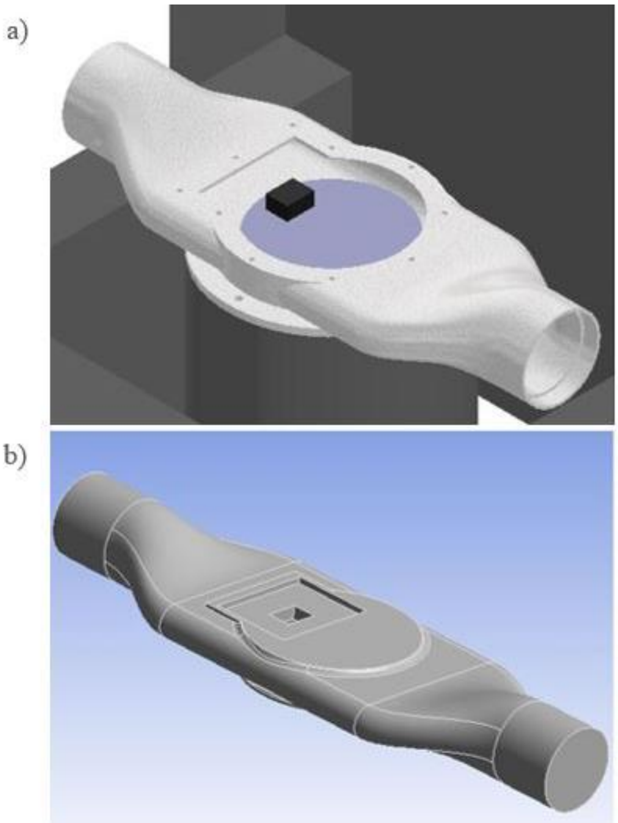

Figure 7: a) CAD assembly (not showing the pad holder) and b) the extruded volume for the CFD simulation.

With the fluid volume defined, it was discretised into elements using a meshing process. The mesh is comprised of unstructured tetrahedral elements with a size of 0.9mm applied to all wall surfaces. Mesh inflation was added to improve the boundary layer resolution consisting of 15 layers with a growth rate of 1.1 and a smooth transition to the bulk mesh. An air velocity of 6.87m/s was prescribed using a velocity inlet boundary condition, positioned at the entry of the housing. A pressure outlet boundary condition was specified at the outlet with a static pressure set to atmospheric pressure. All wall surfaces were set to be smooth with the no-slip condition.

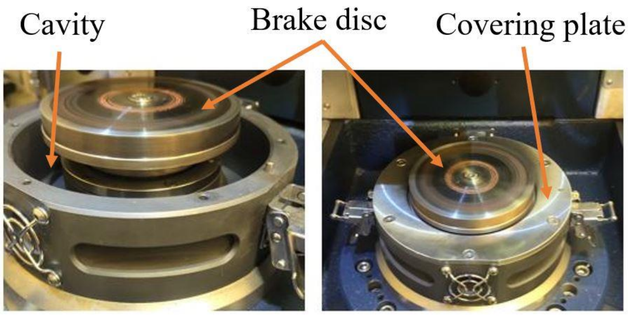

Figure 8: Rotating drive with the cavity (left) and with the covering plate (right).

The UMT allows for the rotating drive to operate with or without a plate that covers a cavity below the rotating disc (Figure 8). Two housing designs were considered. The first one works together with the rotating drive without the covering plate. The housing for this variant extends underneath the disc to allow for the airflow to pass underneath (Figure 9a). The second design works in conjunction with the covering plate that caps the housing off at the bottom of the disc (Figure 9b).

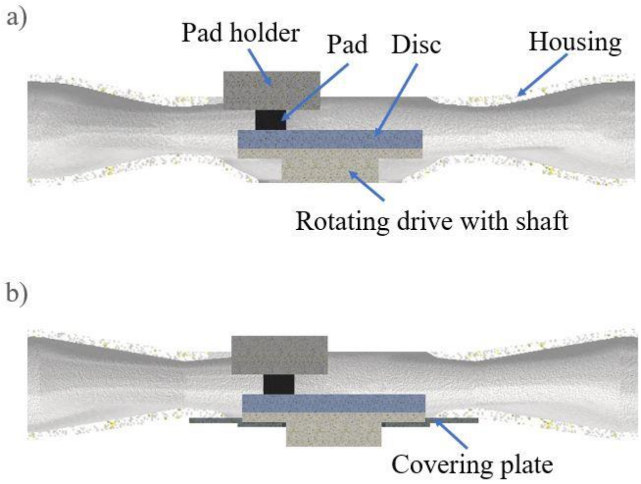

Figure 9: Half section views of the housing designs a) with the cavity and b) with the covering plate.

Figure 10 shows the airflow using pathlines coloured by velocity magnitude throughout the housing for the two designs. Based on these results, the decision was made to go forward with the housing design using the covering plate to increase the air velocity around the friction interface to allow for a fast evacuation of the airborne particulate matter.

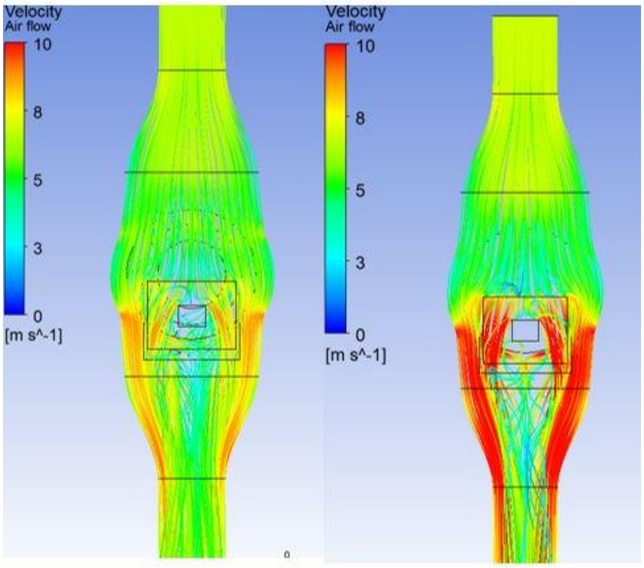

Figure 10: Top view of the airflow through the housing with the cavity (left) and with the covering plate (right). Flow direction is from top to bottom.

In the next design step, the height of the housing was reduced, and the width was increased to minimize the protrusion of the pad holder into the housing and to further reduce the enclosure volume to promote a fast evacuation of the particulate matter. As can be seen in Figure 7, the housing now keeps a consistent cross-section throughout the area of the brake assembly.

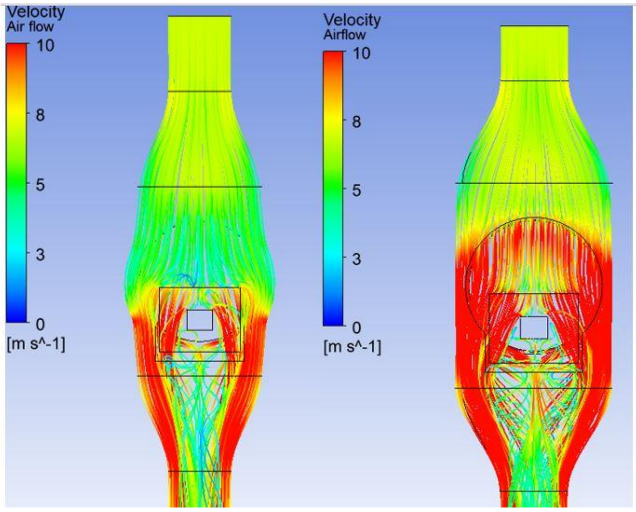

Figure 11: Top view of the airflow through the housing before (left) and after reducing the housing height (right). Flow direction is from top to bottom.

Figure 11 on the right shows how the air is increasingly accelerated over the brake disc surface for the enclosure design with the reduced height.

To see what influence the rotating brake disc can have on the airflow, the brake disc surface was set to a rotating speed of 2500 rpm (around 80 km/h vehicle driving speed) in the simulation.

Figure 12 shows that the rotating disc creates a spiral-shaped vortex structure towards the exit of the housing. In comparison with the static disc simulation results shown on the right side of Figure 11, the overall airflow inside the housing is still dominated by the incoming supply from the blower and impact of disc rotation on sampling particulate matter is expected to be minimal.

A noteworthy point is that the air around the brake assembly towards the wall of the housing and ducting keeps a steady direction towards the sampling point without diversion.

The approach of keeping the volume of the isokinetic sampling system as small as possible has led to a total volume of only 0.0016m³ (from the housing inlet to the sampling plane inside the ducting).

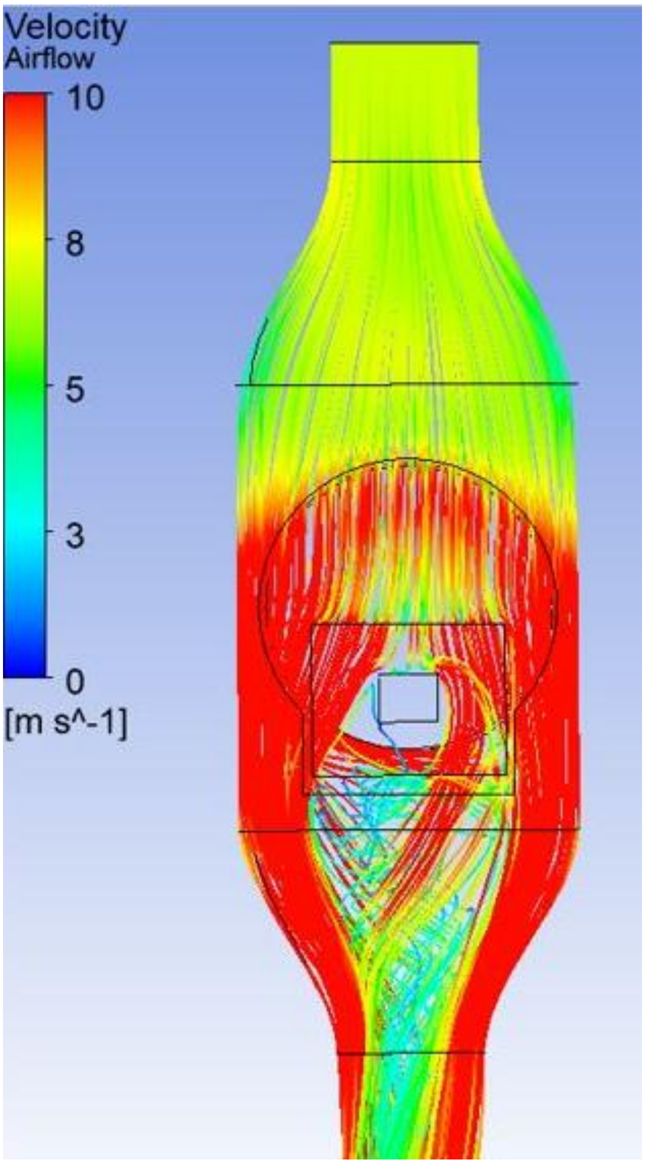

Figure 12: Top view of the airflow through the housing with a anti clockwise rotating speed of the disc of 2500rpm

The resulting air exchange rate will be 8.42 (number of air changes per second). This will help to keep the particle residence time in the system to a minimum, preventing settling adjacent to duct walls.

5. CONCLUSION & OUTLOOK

The motivation behind the presented small-scale system was to be able to reliably test brake friction couples under realistic braking conditions. An effort was made, through the scaling exercise, to match the sliding conditions as closely as possible to a common full-size brake system. The results showed that key geometric parameters such as the sliding radius differ slightly from what they would have been (recall Section 2) using a fixed scaling factor in order to help match the three geometric scaling parameters R_A, R_V and θ.

The two control strategies allow the test bench to be operated by input parameters such as pv-values or by the systems frictional output. The first strategy can be used to investigate and compare the friction couple's response to certain operating conditions while the second strategy is important to allow for direct comparability of different friction couples under identical power and energy output conditions.

The CVS system was designed to sample particulate matter as efficiently as possible, generating a consistent airflow around the sliding interface to direct any airborne particles towards the outlet of the housing. Unlike a full-sized dynamometer system, the housing volume could be kept much smaller and the airflow inside the housing is not dominated by flow through the cooling vanes of the rotor.

The proposed test bench will be used to screen different brake friction couples in an effort to investigate the underlying tribological processes that take place during braking. The housing of the small-scale brake system will be manufactured through selective laser melting (SLM) from aluminium alloy (AlSi10Mg) and manually polished to obtain a smooth surface finish. The ducting will be manufactured from 316 stainless steel and electrochemically polished.

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ACKNOWLEDGEMENT

This work has been funded by the Engineering and Physical Sciences Research Council (EPSRC) Grant no.: EPL01629X1 and TMD Friction Services GmbH.