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RESEARCH

An Experimental Study of Forced Vibration Influence on Disc Brake Drag Torque in Heavy Commercial Road Vehicles

Pontus Fyhr[a], Rikard Hjelm[b], Jens Wahlström[b,*]

[a]*Haldex Brake Products AB, Landskrona, Sweden,*
[b]*Lund University, Department of Mechanical Engineering, Lund, Sweden.*

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* Corresponding author:

Jens Wahlström
E-mail: jens.wahlstrom@mel.lth.se

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A B S T R A C T

Drag torque in a disc brake is the residual torque when the brake is not activated. This torque can adversely influence the total energy consumption, life of the brake components, and level of airborne wear particles from vehicles. Drag torque is of special interest for long haul trucking since it could have an effect of the total energy consumption. Drag torque performance of a disc brake is usually evaluated with conventional inertia dynamometers. Vibrations induced by the wheel can help to retract the pads from the disc and decrease the drag torque. It is hard to study this phenomenon with conventional inertia dynamometers since there is usually no system to induce a forced vibration on the brake. Therefore, the aim of the paper is to investigate the effect of vibrations on the drag torque for a commercial vehicle disc brake with a novel test setup. This is done by comparing test run with and without pads, forced vibrations, and pad retraction springs. The results indicate that the forced vibrations strongly affect the level of measured drag torque. In addition, the introduction of a retraction spring can reduce drag torque with more than 50% when vibrations are induced.

1. INTRODUCTION

Disc brakes are frequently used to safely slow down vehicles. During braking, a force is applied inside the caliper, to push the pads into contact with the disc, and the resulting sliding friction between the pads and disc slows the vehicle down. After braking, the pads could still be in contact with the disc since there is often no active mechanism that controls the retraction of the pads. This residual torque, which is a result of the pads being in contact with the disc between brakings, is often called drag torque.

Drag torque can have an effect on the total energy consumption, life of the brake components and level of airborne wear

particles from vehicles. Drag torque is of special interest for long distance driving at relatively constant vehicle speed, such as long-haul trucking, since it can have an impact on of the fuel consumption [1] and uneven pad wear [2].

An important consequence of drag torque is the generation of airborne wear emissions [3-6] which have negative health impact [7]. It was estimated by Farwick zum Hagen et al. [8] that brake drag contributes with about 34% to the total airborne particle mass emission from a passenger car. However, Hascoët and Adamczak [9] report that drag torque could be responsible for around 14% of the total particle mass emission.

A large number of factors influence the drag torque [2,6,10]. Some disc brakes use springs that connect the pads with the carrier and/or the caliper to retract the pads to reduce the drag torque. Piston sealing roll-back utilized on hydraulic disc brakes are normally not applicable on heavy road vehicles. Moreover, since there are tolerances of the thickness of disc, so called disc thickness variation, and axial run-out, there is also a knock-back by the disc on the pads, which could help to decrease the drag torque. In addition, structural and thermal phenomena of the components such as stiffness of caliper, pads, and disc, thermal expansion, and coning of the disc, all have an influence on the drag torque. It should be mentioned that drag torque is important from a safety perspective since it helps to clean the contacting surface from rust [11] and dirt, and guarantees a good friction performance when the brakes need to be used.

Drag torque performance of disc brakes is generally evaluated on component level with brake dynamometers e.g. [2,4,6,12] using standardized drive cycles. The impact of different drive cycles has been investigated by Woo et al. [13] .

The influence of the vehicle dynamics (e.g. cornering and road surface height variation) on drag torque is generally not studied with this kind of test setup. Haag et al. [14] used a dynamometer setup equipped with a floating caliper to investigate the influence of wheel load and side forces mainly due to cornering on drag torque. They concluded that side forces considerably affect the level of drag torque.

The motivation for performing tests on only an axle, with disc and caliper, is that isolating the brake drag torque losses from tyre rolling losses, bearing losses as well as aerodynamic drag losses from the rotating wheel will be very difficult. By utilizing only the minimum amount of components and performing tests with the same assembly, allows for clear differentiation of results when the drag torque reduction device is used from when it is not. With other losses present in the system, separation of the drag torque influence becomes more difficult.

There is a lack of studies in the scientific literature about how vibrations induced by the wheel could affect the drag torque for heavy commercial vehicles. It is hard to study this with conventional inertia dynamometers since there is typically no system to induce a forced vibration on the disc brake.

Therefore, the aim of the paper is to investigate the effect of vibrations on the drag torque for a heavy commercial vehicle disc brake with a novel test setup. This is done by comparing test runs with and without pads, forced vibrations, and pad retraction spring.

2. EXPERIMENTAL SETUP

2.1 Disc brake system

A standard SAF-Holland trailer axle with a Haldex ModulT disc brake in standard configuration is used in the present setup. In Fig. 1, an overview (upper) and a cross-section view (lower) of the disc brake is presented. A sliding caliper is supported on guide pins in a carrier fixed to the axle, and an actuation device comprising a pneumatic actuator is mounted on the caliper and acting on a lever connected to a mechanical clearance adjuster in the caliper housing. The pads are slidingly supported in the carrier and forced against guide surfaces at the bottom of it by pad springs supported by a pad retainer at the top of the caliper.

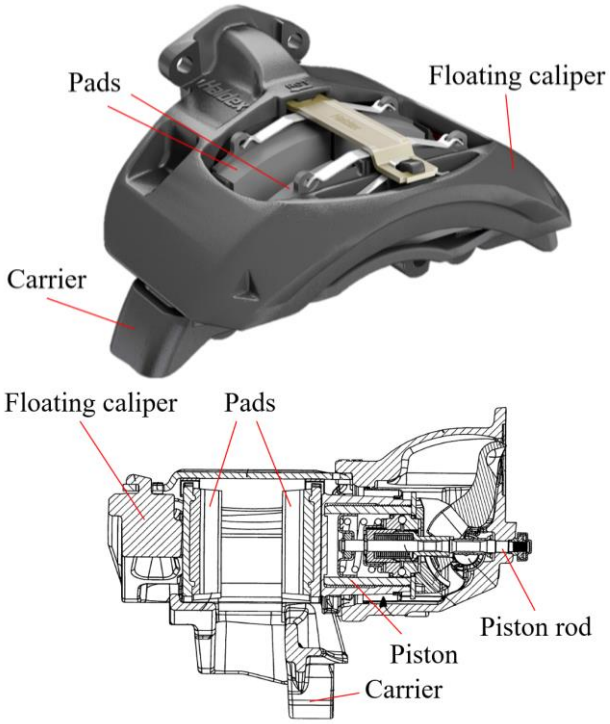


Fig. 1. Upper: Disc brake with its floating caliper, knuckle and pads. Lower: Cross-section view of the disc brake with piston and piston road.

Some parameters of the brake assembly to be tested are essential to determine, e.g. moment of inertia of the disc and parts rotating with it, rolling circumference of wheel assembly (for relation of speed and rpm), and expected driving speeds and scenarios for the actual vehicle. The values of the most important parameters for the disc brake system used in the present study are presented in table 1.

Table 1. Data for the disc brake system.

Parameter	Value
Weight without disc [kg]	<32
Pad thickness [mm]	30
Pad surface area [cm^2]	159
Disc weight [kg]	27
External disc diameter [mm]	430
Disc thickness [mm]	45
Wheel radius [mm]	478
Max brake torque [kNm]	20
Moment of inertia [kgm^2]	0.86

2.2 Test rig setup

The test rig is designed for evaluation of brake drag torque for heavy commercial vehicles. Photographs of the test setup (left), disc spinner (middle), and vibration rig (right) can be seen in Fig. 2. Various disc brake assemblies can be tested in test rig to evaluate different discs, calipers, pad springs, and pad retraction devices. The wheel axle end is mounted on a test stand. A spinner is used to accelerate the disc to the initial rotational speed of the test. The disc rotational speed is measured during its deceleration (roll-out).

The test stand is installed in an electro-hydraulic vibration rig, which can induce controlled vibrations both horizontally and vertically. A sketch of the test setup is presented in Fig. 3. The vibration rig can induce vibrations with both constant and varying amplitude and frequency. It is thus capable of reproducing vibrations recorded on vehicles in traffic.

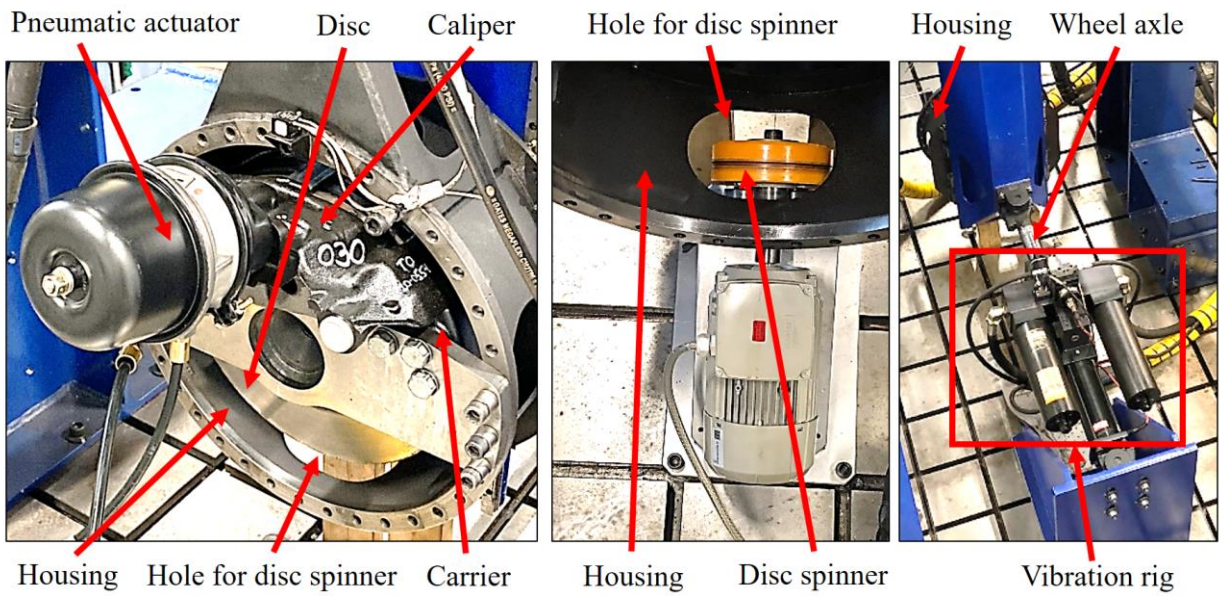


Fig. 2. Left: front-view of the test rig with the disc brake assembly mounted. Middle: top-view of the disc spinner seen from inside the housing. Right: vibration rig placed behind the disc brake assembly.

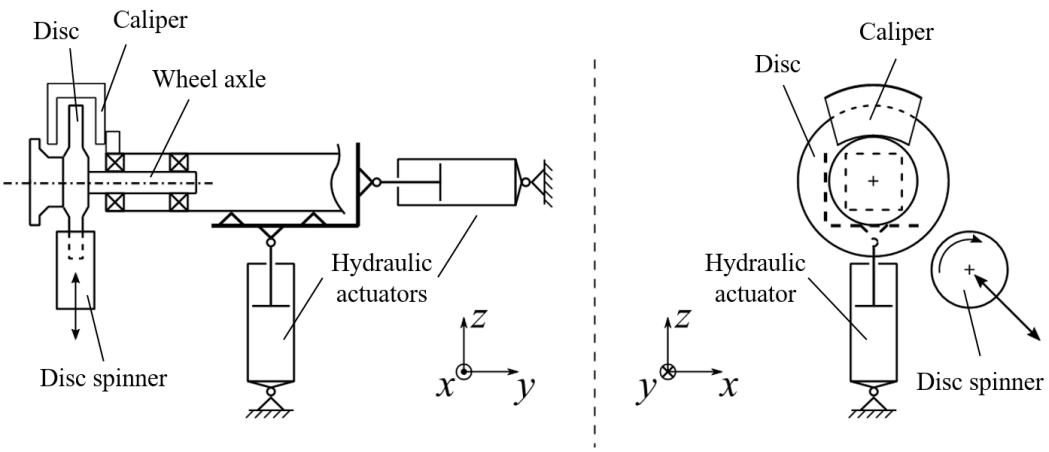


Fig. 3. A sketch of the test stand and the electro-hydraulic vibration rig.

2.3 Test procedure

Six different cases are tested three times each. The cases are: 1) NVNP (No Vibration and No Pads), 2) VNP (Vibration and No Pads), 3) NVNS (No Vibration and No retraction Spring), 4) VNS (Vibration and No retraction Spring), 5) NVS (No Vibration and retraction Spring) and 6) VS (Vibration and retraction Spring).

All cases are run with the test procedure shown schematically in Fig. 4. The procedure is also explained in the numbered list below.

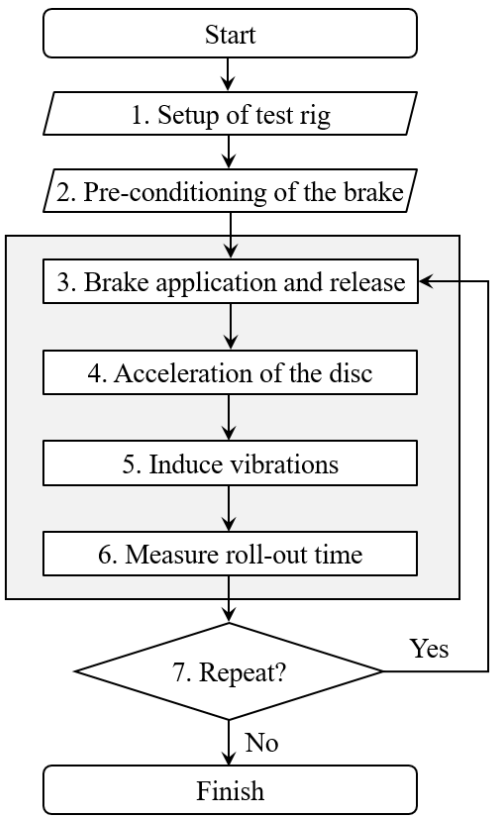


Fig. 4. A schematic overview of the test procedure.

1. **Setup of the test rig**. Preparation of the bearing assembly by removing seals, releasing bearing axial pretension, and removing almost all lubricant. Preparation of the pad clearance by setting the pads manually at approximately 1 mm clearance from the disc.
2. **Pre-conditioning of the brake**. This is done by making 100 brake applications at 1.5 kN actuator force without disc rotation. It brings the adjuster to a state of final and stable clearance.
3. **Brake application and release**. The brake is applied with an actuator force of 1.5 kN without disc rotation, and then released.
4. **Acceleration of the disc**. The spinner is set in contact with the disc and accelerates it up to 500 rpm, thereafter it is removed from the rotor. This rotational speed corresponds to a travel speed of 90 km/h at a wheel rolling circumference of 3000 mm (rolling radius 477.5 mm).
5. **Induce vibrations**. A vibration is induced on the brake at 1.0 g and 7 Hz along the *z*-axis, and 0.3 g and 6 Hz horizontally in *y* direction, continuously until the disc stops as a consequence of drag torque, bearing resistance, etc. Different frequencies were chosen for the axis to cause vibrations that are not in phase.
6. **Measure roll-out time**. The rotational speed of the rotor is measured with an analogue wheel speed sensor, with processed data output at 100 Hz.
7. Repeat point 3-6 three times.

2.4 Drag torque estimation

The total residual torque (M_{tot}) can be determined by

$$M_{\text{tot}} = J\frac{d\omega}{dt} \qquad (1)$$

where J is the moment of inertia for the rotating parts and $d\omega/dt$ is the angular deceleration of the wheel and disc. Equation (1) can be approximated as

$$M_{\text{tot}} \approx J\frac{\Delta\omega}{\Delta t} = J\frac{\Delta v}{\Delta t R} \qquad (2)$$

where $\Delta\omega$ is the change in disc angular speed during the time Δt, Δv is the corresponding vehicle speed difference, and R is the wheel radius. In this study, the speed difference Δv is chosen to ±10 km/h around the speed that drag torque is estimated for.

3. RESULTS

Fig. 5 to 7 present the vehicle velocity versus test time for all tested cases. A summary of the deceleration time between 90 and 20 km/h can be seen in Fig. 8.

An estimation of the drag torque is done by using equation (2) with the results presented in Fig. 5-7 and the data in Table 1. The estimated residual torques at corresponding vehicle speeds of 50 and 75 km/h are presented in Table 2.

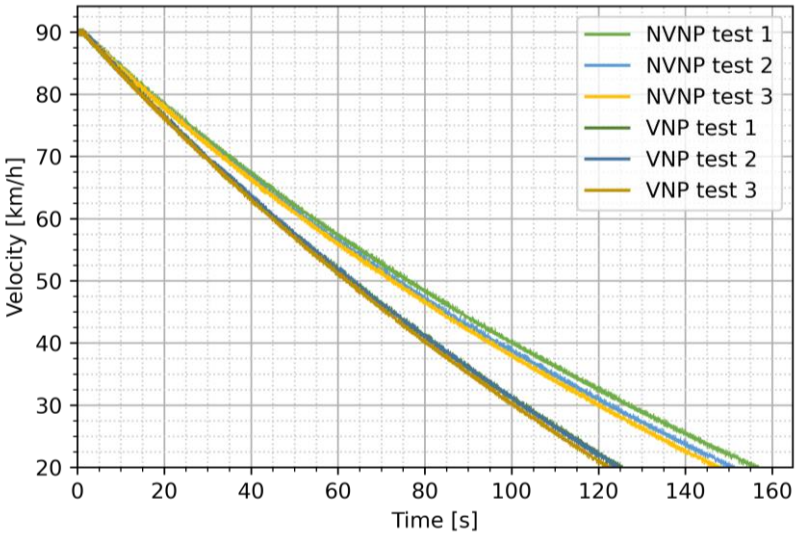


Fig. 5. Vehicle velocity versus time for the NVNP and VNP tests.

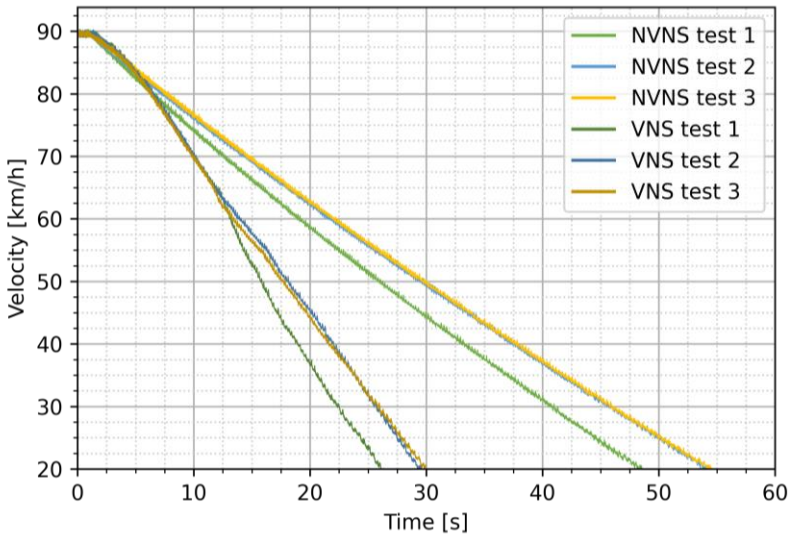


Fig. 6. Vehicle velocity versus time for the NVNS and VNS tests.

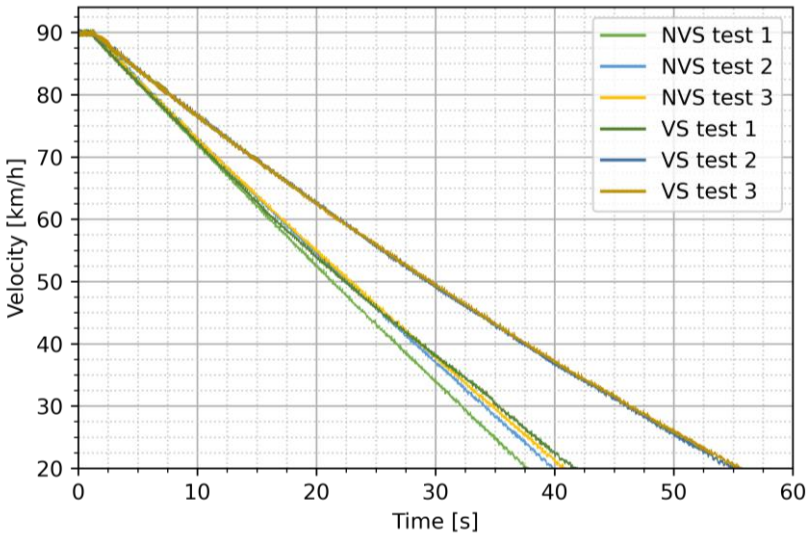
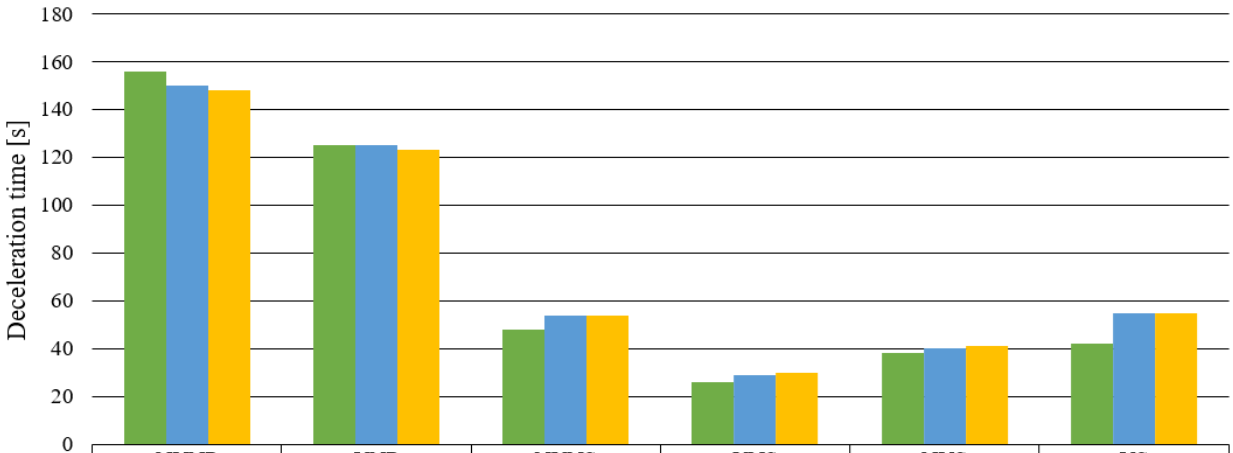


Fig. 7. Vehicle velocity versus time for the NVS and VS tests.



	NVNP	VNP	NVNS	VNS	NVS	VS
Test 1	156	125	48	26	38	42
Test 2	150	125	54	29	40	55
Test 3	148	123	54	30	41	55

Fig. 8. Deceleration time between 90 and 20 km/h for the all tested cases.

Table 2. Estimated residual torques at vehicle speeds of 50 and 75 km/h with vibrations.

Case	75 km/h	50 km/h
VNP	0.6 Nm	0.5 Nm
VNS	1.9 Nm	2.3 Nm
VS	0.9 Nm	0.8 Nm

4. DISCUSSION

The significance of this study lies in the novelty of the test setup. Commercial brake dyno rigs measure the brake drag torque either directly from friction force e.g. [14] or indirectly by e.g. measuring roll-out time e.g. [8]. These rigs are often expensive and complex, as well as large, especially for heavy commercial road vehicles. To replicate a heavy commercial road vehicle, a large inertia is needed. It is challenging to distinguish a relatively small drag torque from a large inertia and large friction torque in the flywheel bearings.

In the present study, the only inertia is that from the brake disc and the trailer axle. This inertia is considerably smaller, in the order of hundreds of times, than in a commercial inertia brake dyno. This implies decreased bearing torque, but does not affect drag torque. Thus, the residual torque comes practically only from brake drag torque. Also, it is possible to include forced vibrations on the brake in a controlled manner in the present setup.

4.1 Discussion of results

As can be seen in Fig. 8, the tests with forced vibrations show a different deceleration time compared to the tests without vibrations, for all tested cases. It can be seen in Fig. 5 to 7 that the repeatability for all tested cases is good for test 2 and 3. However, test 1 stands out, possibly due to adaption of the brake after the initial 100 brake applications (see test procedure, step 2). A possible explanation for this is that the pad is not yet knocked back by the disc at the start of test 1, but has found its intended pad gap for tests 2 and 3.

The pad retraction spring reduces the drag torque significantly for the test case with vibrations (see Table 2). The vibrations in both in the horizontal and vertical direction can explain this. In the vertical direction, the vibrations push the pads against the mounting spring which results in a lower contact force between the carrier support surfaces and the backplate. The vibrations in the horizontal direction gives a contribution to the retraction force. Consequently, the pads are allowed to move more freely, and the retraction spring force is enough to retract the pads, which is not the case for the tests without vibrations. This indicates that the coefficient of friction between the carrier and pads is an important factor when selecting the stiffness of the retraction springs as also reported by e.g. [2] and [14].

With no vibrations, the retraction spring can decrease roll-out time, i.e. increase drag torque, see case NVS compared to NVNS (see Fig. 8). This could be explained by a wedging effect caused by the retraction spring. The vibrations counteracts the wedging by allowing the pad to move more freely.

4.2 Discussion of method

The experiment only includes the brake disc and wheel hub as rotating parts, no wheel rim or tyre, nor any normal load. The influence of these factors on a full vehicle travelling on a road are expected to be relatively small, making the obtained results relevant in a comparative context for quantifying the effects of drag torque. The values of obtained drag torque should not be used solely in longitudinal dynamics simulations, the bearing and seal friction should be quantified separately [2].

Before testing, it is recommended to measure the disc thickness variation and the disc axial run-out as they may provide pad knock-back influencing drag torque. Other parameters to observe are disc coning, slack and sliding resistance of caliper guiding [15], type of pneumatic actuator (parking brake actuator changes centre of gravity). Disc coning is influenced by temperature distribution at, and after, a braking operation. These effects vary between disc types and cannot be studied with the test procedure described here. Also, disc brakes may be installed on axles at various rotary positions (clocking [16]). Study of the possible effects of different clocking can to some extent be performed with the test procedure described here.

In the present test procedure, the vibrations have a constant amplitude and frequency. To better replicate real driving conditions, it would be advantageous to use a vibration spectrum from measurements conducted on-road. The disadvantage of this approach is that cases with different roll-out times will be affected differently by the vibration spectrum. This can be solved by repeating short sections of the spectrum several times during a roll-out. Alternatively, one entire spectrum can be used for all designs and the speeds at predetermined times are used to estimate the drag torque. This method will not handle the speed dependence of drag torque.

4.3 Discussion of environmental gain

As can be seen in Table 2, the difference in estimated drag torque between VNS and VS is from 1 to 1.5 Nm. This corresponds to a decrease of 53% and 65%, respectively. The gain of this is twofold. Firstly, reduction in drag torque can potentially decrease the airborne mass particle emissions considerably as discussed in [8] and [9]. Secondly, reduction in drag torque can reduce fuel consumption and exhaust emissions. A rough estimation is that every Newtonmeter of reduced drag torque per wheel at 75 km/h potentially saves 43 W, or 259 W for 6 wheels. If the fuel consumption is assumed to be 0.3 litres/kWh and the yearly driving distance is 100.000 km, the possible fuel saving is about 15 litres of diesel for a heavy commercial vehicle. With almost 700 000 heavy commercial vehicles in Sweden [17], this amounts to roughly 10 million litres of diesel saved yearly. Energy consumed by brakes is also discussed by e.g. Woo et al. [13] .

5. CONCLUSION

The effect of forced vibrations on drag torque for a commercial heavy vehicle disc brake is studied with a novel test setup. The setup consists of a trailer axle with a disc brake assembly together with an electro-hydraulic vibration rig. Roll-out time is used to indirectly estimate drag torque. The novel test setup does not use inertia flywheels as typically used in commercial brake test rigs. The study is done by comparing test run with and without pads, forced vibrations, and pad retraction springs. The following conclusions can be drawn based on the results of the study:

- The forced vibrations on the disc brake assembly strongly affect the drag torque for all tested cases.
- The drag torque decreases by 50-65% when a pad retraction spring is used together with forced vibrations.
- The possible reduction in fuel consumption due to the pad retraction spring is estimated to 15 litres of diesel yearly for a heavy commercial road vehicle.

It remains to compare the results of the present study with those of commercial brake test rigs.

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