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RESEARCH

Experimental Method for Analyzing Friction Phenomenon Related to Drum Brake Squeal

Automobile brakes have been intensively developed during past few decades, but the maximum motor's power, that should amortized in vehicle brakes, has been significantly increased also. Most of the kinetic energy of the moving vehicles is transforming into heat through friction. But the small part of kinetic energy transforms into sound pressure and makes noise. Low frequency squeal of drum brakes is very intense and can lead to customers' complain. The interaction between the brake system and the vehicle framework and suspension is often very substantial during occurrence of brake noise. Unfortunately, to solve this type of squeal problem is also difficult because of the large number of components involved. The other cause is attributed to self-excited vibration that is induced when the friction material has a negative slope in relation to the relative velocity. This paper illustrates an approach to experimental studies of drum brakes in road conditions in order to monitor changes in the coefficient of friction that can generate drum brake squeal at low frequencies.

Keywords: *drum brakes, noise, friction, experiment, road testing.*

1. INTRODUCTION

Until recently, drum brakes were used in many types of vehicles, especially heavy vehicles such as trucks and busses. Now, disc brakes are emerging also on those vehicles. Drum brakes will, however, be in use for many years to come. Disc brakes are more common on passenger cars, but on small and non-prestige cars, drum brakes are common on the rear axles. Drum brakes are well-proven, simple and quite reliable. However, like most types of mechanical brakes, they suffer from squeal. The squealing noise occurs as a very clear harmonic sound with frequencies between a few hundred Hz to a few kHz on heavy drum brakes, and some kHz for heavy duty disc brakes. Passenger car brakes usually show higher frequencies than heavy vehicles. Among the heavy vehicles, noise from city busses is the most disturbing, due to frequent stops in residential areas. The noise level of brake squeal can be greater than 100 dBA. The driver of a squealy vehicle might feel that he does not want to disturb, and therefore, he does not want to brake. Another point of view is the customer perspective. As the customer has spent a lot of money on his vehicle, he does rot want any disorders. As there is a trend towards quieter vehicles, great emphasis is put on the elimination of brake squeal. This is done despite the fact that this squealing noise does not affect brake performance and safety [1].

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There are two general causes of brake squeal. One cause is attributed to self-excited vibration that is induced when the friction material has a negative slope in relation to the relative velocity. The other cause is ascribed to self excited vibration due to the coupling of the two eigenvalues that are present in parts experiencing friction. It is difficult to conceive that brake squeal, which occurs at high frequencies of several kHz, would be caused by either one of these factors operating alone. Rather, it is presumed that brake squeal is caused by the compound effect of both factors. One way to reduce the first type of squeal is to improve the friction characteristics of the friction material. This generally involves reducing the velocity dependence of the friction coefficient of the friction material as much as possible. The second type of squeal has been reduced by optimizing the vibration characteristics of the brake assembly that cause self-excited vibration. This was accomplished by conducting a complex eigenvalue analysis of a brake assembly model in which the friction coefficient was set to a certain constant value [2].

The description of the characteristics of squealing drum brakes can be used as a base for the search for solutions to the problem. The research strategy is the development of methodology for determination of the change in a coefficient of friction on the drum brakes in terms of road research. In this way, it is possible to effectively monitor changes that can lead to squeal phenomenon.

2. CATEGORIZATION OF FRICTION INDUCED VIBRATIONS IN BRAKES

The development of models concerning brake squeal has historically undergone some major changes. The changes lie in the friction models proposed for generating squeal as well as in the mathematical treatment and modeling of the brake.

There are many properties of a friction material in combination with a structure that has the potential of creating squeal. Figure 1 summarizes these different types of friction characteristics and excitation types.

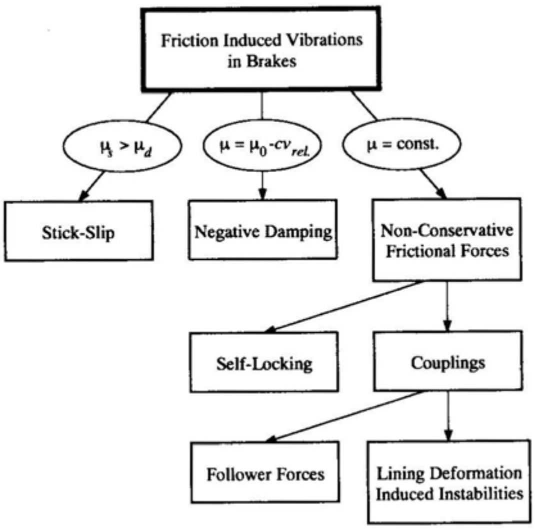

Figure 1. Categorization tree for friction induced vibrations in brakes

In the figure 1, the ovals indicate friction models while the boxes indicate excitation types. The stick-slip phenomenon, occurs when the static coefficient of friction, μ_s, is larger than the dynamic one, μ_d. Here, the friction model corresponds to one excitation type.

The earliest studies focused on a particular variation of the coefficient of friction sliding speed. Before 1950, the studies were either experimental in character or consisted from a compilation of experiences gained by manufacturers and operators. The suggested cause was stick-slip, i.e. motion caused by a static coefficient of friction that is larger than the dynamic one, as illustrated in figure 2.

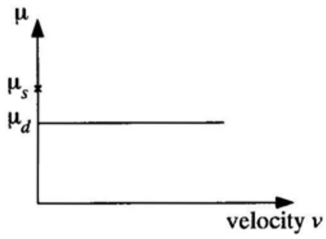

Figure 2. Friction characteristic when stick-slip is possible

The second type of friction model has a negative slope in the μ-velocity characteristic, figure 3. Since this slope gives a relation between friction force and the relative velocity, it can be seen as damping. If the slope is negative, the system will be subjected to "negative damping", and if this negative damping is larger than the "ordinary" damping of the system, it will be unstable.

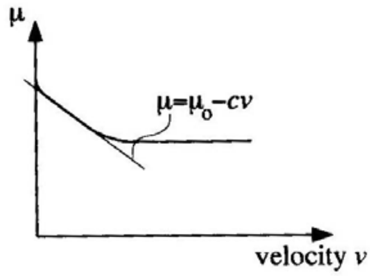

Figure 3. Friction characteristics when "negative damping" is possible

A series of one-dimensional models covering this slope in the coefficient of friction were developed. All the models are more or less like the one shown in figure 4.

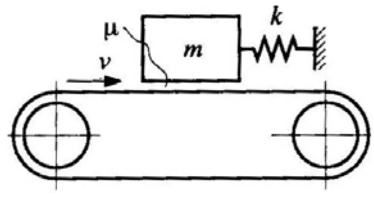

Figure 4. Negative μ-velocity slope model

The models were simple and the results from them were verified by experiments on equally simple rigs. However, the models did not come near the complexity of a brake. Nevertheless, the negative μ-velocity slope excitation type might still be responsible for brake squeal since many lining materials have this characteristic.

In the case when the coefficient of friction is assumed to be constant, non-conservative friction forces occur, i.e. energy is put into the system if it is displaced along certain paths. Such forces, together with certain geometries, can generate instabilities. Two major groups of instabilities can here be observed:

- self-locking, e.g. pin-disc models and sprag-slip models.
- couplings of modes.

There are two types of couplings:

- The first type is the follower forces. In this case, there is a coupling between rotational degrees of freedom and radial and tangential force components.
- The second type is lining deformation induced instabilities. When the normal force at the drum-lining interface varies, the friction force will vary and a coupling between radial degrees of freedom and tangential friction forces occurs.

The friction excitation types in figure 1 can be grouped according to the vibration amplitude at which they are active, figure 5.

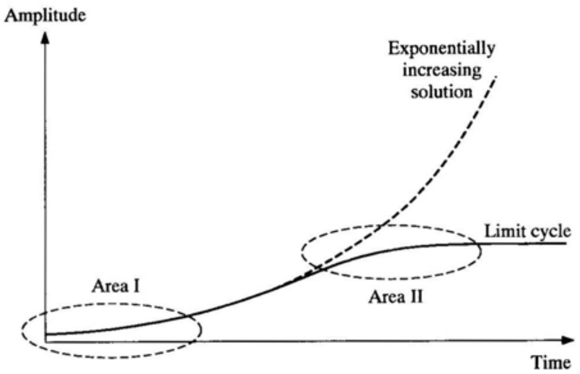

Figure 5. Schematic envelope of growing unstable vibration

Mathematical models of friction induced instabilities can be linearized for some of the excitation types. Such linearized models can be used for studies where the tendency of small vibrations to grow is to be investigated. On the other hand, there are instability types that are only active above a certain threshold value in the amplitude. In such cases, linearized models are not appropriate. Hence, the excitation types are grouped as follows:

- Area I (excitation types active for infinitely small disturbances);
  - lining deformation induced instabilities,
  - follower force induced instabilities,
  - μ-velocity slope induced instabilities,
  - self-locking induced instabilities (beginning of stick phase).
  - It is only the beginning of the stick phase, which is the unstable part of the path that can be analyzed in a linearized model.
- Area II (excitation types not active for infinitely small disturbances):
  - stick-slip induced instabilities,
  - self-locking induced instabilities (end of stick phase and slip phase).

The lining deformations tend to couple modes and the closer are the frequencies, the larger are the instabilities. If modes occur at the same or nearly the same frequencies, it is very likely that they couple together, and when they do, the system becomes unstable, see figure 6.

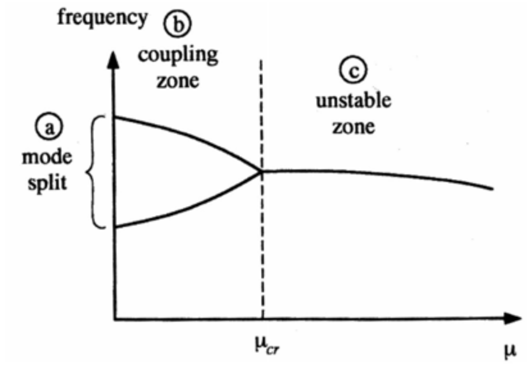

Figure 6. Coupling of modes in actuated brake as function of coefficient of friction

Figure 6 shows three different zones for the solution. The zones can be characterized as follows:

a) The applied brake has no rotational symmetry (a rotor of a non-applied brake is symmetrical). Therefore, an applied brake has in general distinct eigenvalues. In other words, the modes associated with the double modes of a free rotor are split up. This is called "mode split" and gives the brake a resistance to instabilities.

b) The non-conservative friction forces tend to couple split modes.

c) When the modes are coupled together, the solution becomes unstable. The more the coefficient of friction is increased, the more unstable the system becomes. The mode shapes for these unstable modes are wave motions [3].

3. TESTING OF BRAKE PAD FRICTION CHARACTERISTICS

Testing of material friction characteristics demands delicate procedure, because their features can not be estimated based on their chemical structure, configuration or on other data used for estimation of metals and alloys, but exclusively based on experimental methods. Friction properties of brake pads depend on several parameters of operating conditions, first of all on temperature, surface pressure and sliding speed at friction surfaces.

Data required for the accurate characterization of friction materials is acquired under controlled conditions in the laboratory and on the vehicle. In order to effectively interpret the acquired data, engineers must be familiar with both the procedures and equipment utilized. Many variables, such as timing, cost, sample availability, and data obtained from a particular test, are considered when selecting a testing methodology appropriate to the engineer's scope of work.

Vehicle brake simulations are conducted on an inertial dynamometer (figure 7). To simulate the kinetic energy of the vehicle mass moving, the dynamometer utilizes mechanical mass fixed in increments to a rotating shaft. An electric motor is responsible for bringing the rotating mass up to a speed set point. Once the set point is reached, the motor releases control, and the braking system is responsible for bringing the rotating mass to a stop. The energy dissipated during braking can be equated to the energy dissipated during braking in a vehicle. To accommodate multiple vehicle platforms, dynamometers utilize a stepped shaft that accepts fixed increments of inertia. To prepare a vehicle for testing, the following equipment is required: transducers (torque, pressure, temperature, speed, acceleration, noise, etc), signal conditioning equipment and data acquisition equipment.

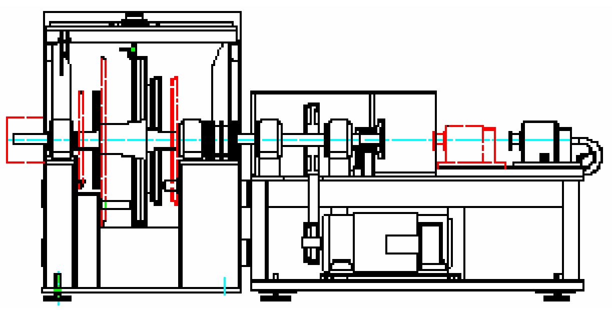

Figure 7. Brake inertial dynamometer

Inertial dynamometers are used for testing of: friction characteristics, durability, influence of friction pads on metal element's life time, dimensional stability, behavior in conditions of extreme heat and cold, noise (including the sounds inaudible to human ear).
A slightly different laboratory test based on the use of real vehicle brake is presented in figure 8. Electric engine, directly connected to the brake through gear box, controls deceleration independently from braking force. This technique is more convenient for noise tests (squealing), but it is limited to low speeds. Described equipment presents a compromise between the flexibility of simplified laboratory testing and relevance of the polygon tests. Brake installation pressure, brake torque and disc temperature are measured during tests. A transducer mounted on drive shaft measures brake torque. The friction coefficient can be calculated.

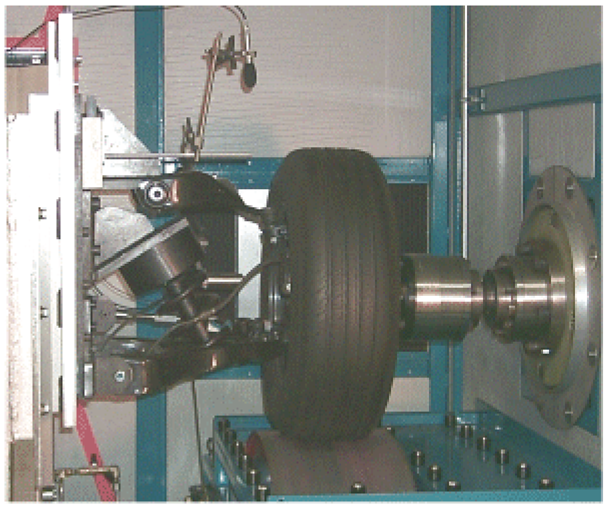

Figure 8. Brake inertial dynamometer

It is important to conduct further testing to understand how the lining material behaves when introduced to the environmental conditions. The dynamometer test should be used again and the linings should be burnished first and then cooled down to the room temperature. Torque on the drive axle was monitored and calculated to obtain the coefficient of friction. The dotted line in figure 9(a) shows that the coefficient of friction stayed relatively constant at value of 0.43 during the six minute test, even thought the temperature was increased slightly (figure 9b).

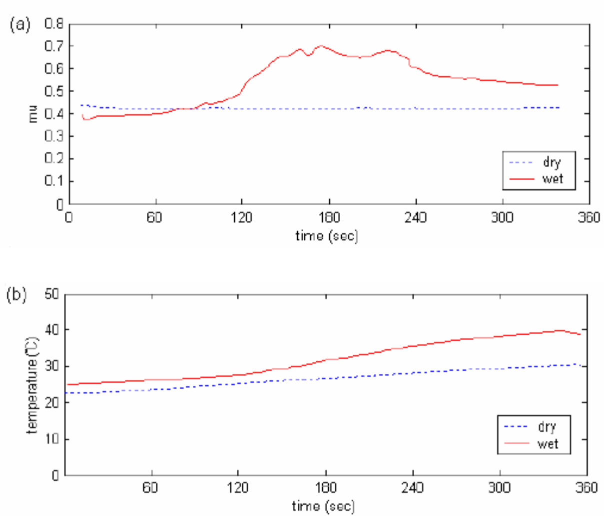

Figure 9. Results of the lining friction test at wet and dry conditions: (a) coefficient of friction, (b) temperature

In the second test, water was sprayed on the same linings as an easy way to introduce the moisture condition. The procedure was repeated and the lining friction coefficient and temperature were plotted in solid lines in figure 9 (a) and (b). This test showed that the lining material was sensitive to humidity. With moisture introduced to the lining, the coefficient of friction increased dramatically. Furthermore, the self-energizing effect of the leading shoe caused brake torque to rise about four times higher than the baseline case. The resulting forcing function was the root cause of the harsh vibration and noise [4].

Vehicle testing procedures vary as widely as dynamometer procedures. Examples of vehicle procedure test objectives include: certification testing, durability testing and product development. Performing vehicle testing is more complicated and expensive than dynamometer testing. In order to prepare for a vehicle test, the following must be considered:

- timing – vehicle testing typically takes longer than dynamometer testing (the dynamometer can be run unattended, the vehicle cannot),
- staffing–scheduling test drivers with appropriate skill levels in a manner that maximizes efficiency in vehicle run-time is difficult,
- facilities – facilities for the installation of test hardware, installation of transducers, signal conditioning and data acquisition are extremely specialized and require the coordination of multiple disciplines,
- inspection – inspections typically occur at mileage intervals and frequently must be performed outside of “normally” scheduled shifts to maximize vehicle run-time.

A direct correlation between dynamometer and vehicle test data is not something that just “occurs” without a considerable amount of planning. In most situations, successful vehicle testing is moved to the laboratory to decrease cost and increase testing efficiency. To develop a dynamometer procedure that correlates with acquired vehicle data, the following must be considered:

- What exactly is being correlated (wear, noise, performance)
- Which brake applies or sequences from the vehicle test are candidates for inclusion in a dynamometer control program
- What is the acceptance criteria for correlation
- How will correlation be reported (correlation coefficient, Iso-Plot, etc.)
- Will each test on the dynamometer be compared to a baseline set of vehicle data, or will relative comparisons be made between the dynamometer data sets.

Common properties successfully correlated between vehicle and dynamometer tests include: noise, wear, performance, judder, torque variation. While it is possible to improve correlation between data obtained on a test vehicle and data obtained on a dynamometer, it is extremely unlikely that dynamometer testing, or any other modeling or simulation testing, will ever fully replace vehicle testing.

4. EXPERIMENTAL METHOD FOR ANALYZING FRICTION PHENOMENON RELATED TO DRUM BRAKE SQUEAL

Cold tests of brakes are performed during polygon tests. Variations of characteristic values during cold brake tests are shown in figure 10.

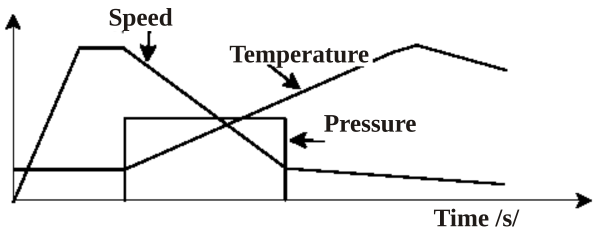

Figure 10. Variations of characteristic values during cold brake tests

A part of the applied measurement system for determination of the drum brake's operation characteristic, shown in figure 10, includes: brake cylinder pressure transducer, brake torque transducer and angular velocity transducer. During braking, activating forces, F_s, move apart the brake pedals, which, leaning on supports, come into a contact with the drum. At the same time, reactive brake torque acts and transmits itself to brake pedals, trying to rotate them in the direction of rotation. Reactive torque mentioned, identical by intensity to the brake torque, is transmitted through pedal supports and brake cylinders to pedal carrier made from sheet metal.

Design solution of brake torque transducers specially adapted for drum brakes is shown in figure 11. According to the solution in figure 11, reactive brake torque is measured indirectly by force transducer subjected to tension during braking. Total brake torque is transferred from pedal support, through force transducer, to screw bolt that connects wheel sleeve with shock absorber and A-arm.

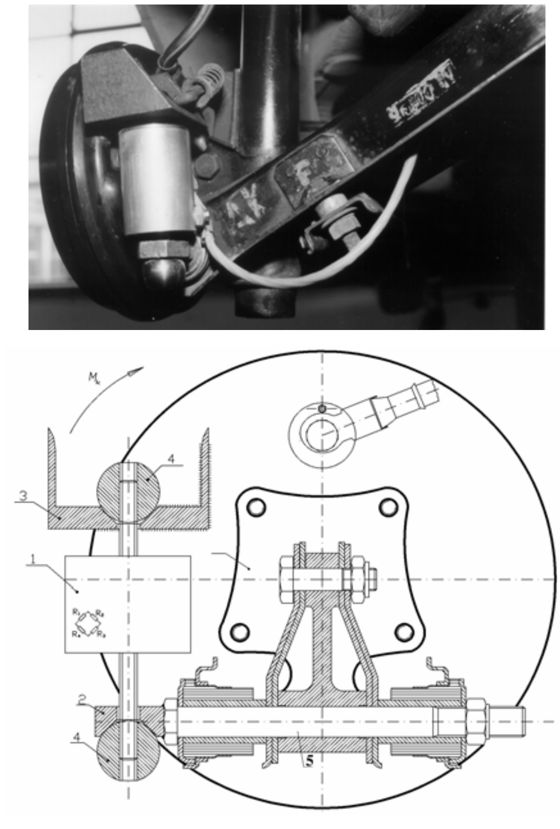

Figure 11. Design solutions of brake torque transducers indirectly, through force transducer

Based on acquired signals, the influence of brake cylinder's pressure variation and vehicle speed on friction coefficient during tests is analyzed and illustrated in figures 13 and 14. The increase of friction coefficient with the increase of pressure is obvious for initial speed of *80[km/h]* while braking. The influence of the vehicle speed is less obvious and declination trend with the increase in speed is present in all cases.

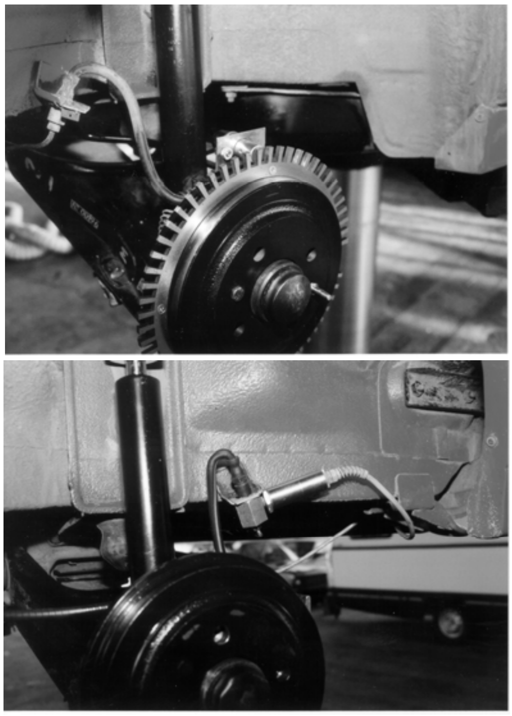

Figure 12. Measuring wheel speed and pressure in the rear brake cylinder

Figure 15 presents another way of analysis of friction coefficient variation with pressure for initial vehicle speed $v_0 \approx 40$ *[km/h]* and for unloaded and fully loaded vehicle.

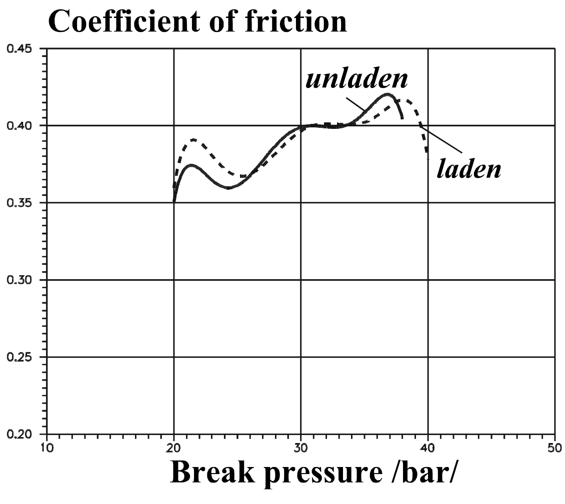

Figure 13. Variation of friction coefficient with the variation of pressure for unloaded and fully loaded vehicle at initial speed of 80 [km/h]

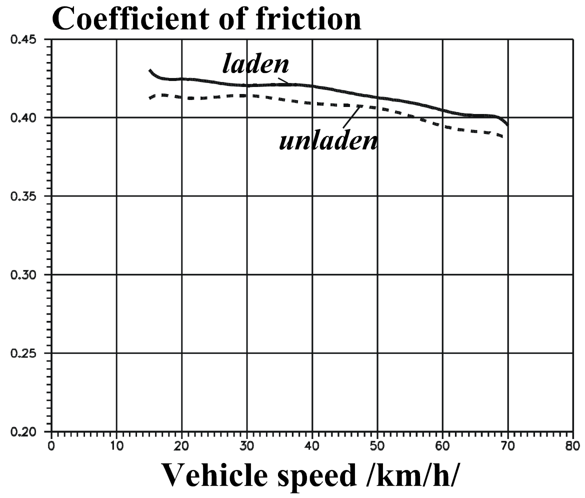

Figure 14. Variation of friction coefficient with variation of speed for unloaded and fully loaded vehicle for speeds up to 80 [km/h]

Measured brake torque is compared to analytically determined brake torque, with the assumption of cosine distribution of pressure along brake pad, for different values of friction coefficients (from *0.1* to *0.5*). It is obvious, especially for the case of fully loaded vehicle, which values of friction coefficient at pressures above *20* [*bar*] are in good agreement with analytically obtained curves.

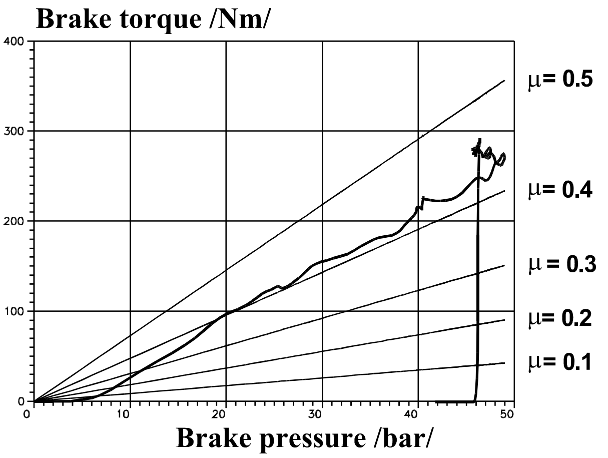

Figure 15. Comparative analysis of measured and analytically determined brake torque aimed at determination of friction coefficient fully loaded vehicle at $v_0 \approx 40$ [km/h]

Variation of friction coefficient during braking may be determined based on results obtained by experimental tests and on assumption of cosine distribution of surface pressure along brake pad. As an illustration of methodology applied, variation of friction coefficient during testing with initial speed of *40* [*km/h*] is presented in figure 16 [5].

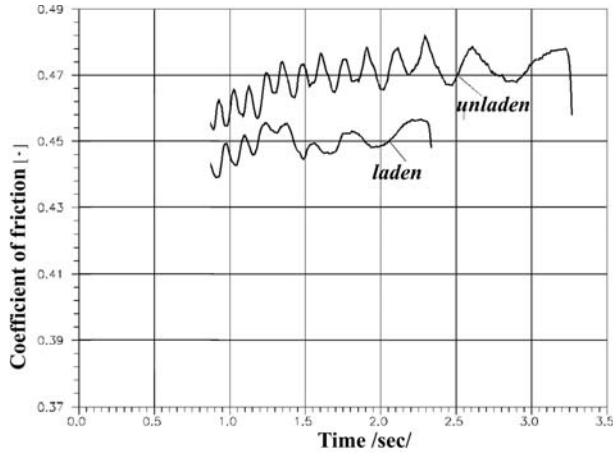

Figure 16. Variation of friction coefficient during brake test at $v_0 \approx 40$ [km/h]

5. CONCLUSIONS

The aim of presented experimental research is to provide a fundamental knowledge on the occurrence of squealing drum brakes and creating the basis for experimental verification of theoretical models of drum brakes. Solving the problem of squeal of drum brakes is difficult because of the large number of components involved in the process. Dynamometer tests are not able to replicate the appearance of noise due to the non-inclusion of suspension components. Even in cases where the noise played on the dynamometer, removing noise that has been successful in laboratory condition can be a failure on the vehicle.

Lessons learned from this study are described below. The friction coefficient seen in a brake system should be better understood. Potential materials should be tested at all possible environmental and operating conditions so a better selection of lining can be made. Second, the design process of the suspension system needs to include its related systems such as brakes for NVH consideration. The dynamics of the suspension links and the torsional compliance of the suspension system need to be studied for potential brake noise problems.

The noise occurs when the lining coefficient of friction becomes very high. Testing has confirmed that this occurs when the lining material is exposed to moisture. Since the coefficient of friction is dependent on temperature, pressure, velocity and humidity, it is important to understand all possible conditions when selecting a lining material. One potential solution is to change to a lining material that is less sensitive to environmental conditions.

However, making a lining change late in a program's schedule will require the rerunning of a series of durability tests. This delay could cost the program several months. Therefore, this solution is useful only at the early stage of the program or when more information of substitute materials is available.

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