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AN AUTOMATED TEST FACILITY FOR EVALUATING FRICTION MATERIAL FOR AUTOMOTIVE-TYPE DISC BRAKES

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ABSTRACT. A constant torque dynamometer with associated instrumentation and control functions for the development of friction materials for automotive-type disc brakes, has been developed. Full scale disc pads are s·bject to a series of intermittent brake applications at a constant rotational speed of the brake disc and constant braking power. This paper gives a description of the dynamometer and an example of results obtained.

INTRODUCTION

The development of friction materials for automotive-type disc brakes requires a thorough investigation into the interfacial phenomena under simulated practical conditions on standardized test equipment. With the advancements of electronic equipment and control systems, programmed full scale "constant torque dynamometers" became available to friction material manufacturers having their own research and development divisions. The essence of this project was to design and construct such a dynamometer together with the required instrumentation and control.

On this dynamometer full scale disc pads are subject to a series of intermittent brake applications at a constant preselected rotational speed of the brake disc. Nominal system pressure is controlled by a feed-back system responding to a signal from a prony brake sensor, varying inversely proportional to variations in the instantaneous friction coefficient, maintaining a constant braking torque.

In sharp contrast to previously favoured test rigs exerting constant pressure on the pads regardless how the instantaneous friction coefficient may vary, a constant torque application in fact represents a constant power application, and the characteristic behaviour of various materials may thus be compared under specific conditions at preselected constant speeds and constant power ratings.

The above mentioned parametric analysis represents the quickest means for first stage determination of "order of magnitude" values in terms of relative wear rate, friction coefficient fade, the presence of exothermic reaction with entrained oxygen at the interface, metal pick-up and disc wear, relaxation oscillations and sonic emissions from the interface.

FUNCTIONAL DESCRIPTION

The motor vehicle disc brake dynamometer is powered by two variable speed hydraulic motors, situated at either end of the drive shaft. The shaft speed is adjustable from the control panel. The true shaft speed is measured by means of a magnetic sensor and relayed to the control panel, where it is displayed and recorded. The disc brake assembly is situated at the middle of the drive shaft. It comprises of a standard automotive brake disc mounted onto a load-arm which moves independently to the main system. An hydraulic power pack supplies the oil pressure in the callipers which actuates the force with which the brake pads are compressed against the brake disc. The pressure in the calliper is regulated by a flow control servo valve, which can be controlled by means of the electric current in its coils. This pressure is measured by means of a pressure transducer mounted in the supply line to the calliper. The braking force signal is obtained from a load cell mounted onto a load-arm as shown in Figure 1. The load-arm onto which the brake calliper is mounted exerts a downwards force onto the load cell during the braking process. The braking torque is calculated by multiplying the measured force value by the length of the load-arm. Both pressure and torque signals are displayed on the control panel and recorded together with the speed signal.

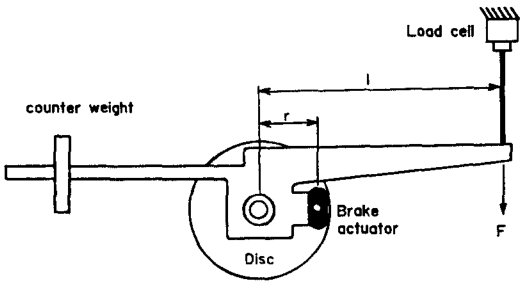

Figure I Load arm–load cell arrangement for braking torque measurement.

To determine the dynamic variation in the value of the friction coefficient a circuit was designed whereby the force signal is divided electronically by the pressure signal resulting in a signal proportional to the friction coefficient. This signal multiplied by a constant produces a time varying signal indicating the value of the friction coefficient. This latter signal is also recorded.

The series of intermittent brake applications is generated by means of a digital sequencer, which can be programmed to apply the brakes at regular intervals of which the on and off times can be dialled into the instrument. The number of brake applications can also be preset. The design makes provision for an automatic and manual mode. In the latter mode the operator switches the "ON" and "OFF" conditions manually and has therefore also direct control over the number of brake applications.

HYDRAULIC BRAKE SYSTEM

The hydraulic brake system for the dynamometer was designed to generate the desired pressure for operating the disc brake. A schematic diagram of the brake system is shown in Figure 2.

An induction motor drives the gear pump (a), which behaves as a constant flowrate source for the system. The oil is then pumped through a high pressure filter (b) and a non-return valve (c), which ensures that the load cannot back-drive fluid to the pump during transient operation.

The accumulator (d), connected in the supply line, smoothes out any short time fluctuations in supply pressure. A pressure relief valve (e) ensures a constant system supply pressure (P_s) to the two-stage flow control servo valve. The relief valve was manually adjusted to provide a supply pressure of 9 MPa.

The primary dynamic element in the system, is undoubtly the flow control servo valve (f). Control flow to the disc brake (Q_b) is regulated by the magnitude of the input current to the servo valve. However, due to the inclusion of an adjustable flow rate restriction (j) in the return brake line, this flow is maintained at a preset constant flow rate. Therefore an increase in the size of the control port orifice leads to an increase in the disc brake pressure (P_b). Exhaust oil is returned to the reservoir via the cooler (g) and low pressure filter (h).

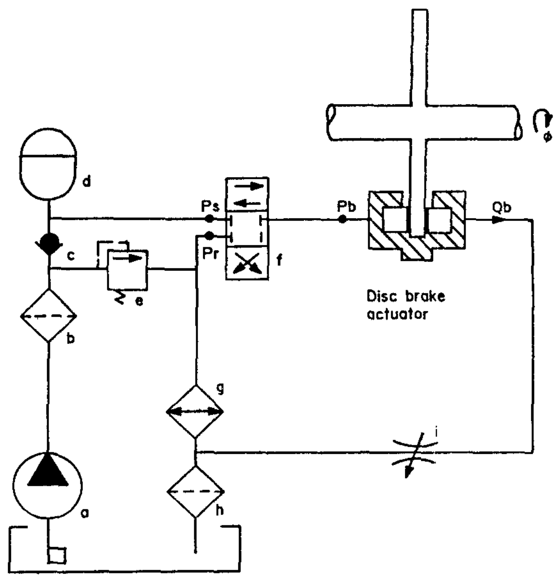

Figure 2 Schematic diagram of the brake system

CONTROL SYSTEM PHILOSOPHY

The complete control system for the dynamometer comprises of two feedback loops as indicated by the block diagram shown in Figure 3.

The inner closed-loop is for the pressure control. A pressure transducer connected in the supply line to the brake actuator provides the necessary feedback signal. The outer closed-loop forms the torque control. The torque feedback signal is generated from a load cell onto which a load arm is connected as illustrated in Figure 1.

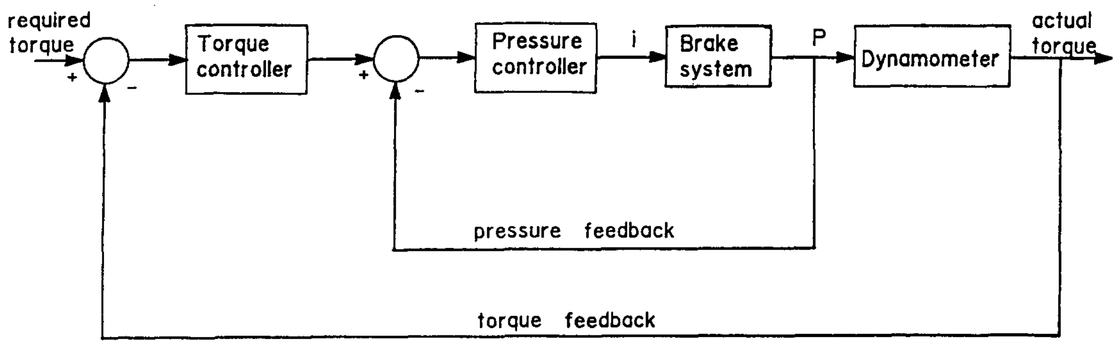

Figure 3 Block diagram of the complete control system

During the braking process, the load-arm exerts a downward acting force F onto the load cell. This force multiplied by the length of the load arm is equal to the braking torque. The braking torque however, is directly proportional to the product of the friction coefficient (μ) of the brake block/disc contact and the fluid pressure inside the brake actuator. Both the friction coefficient as well as the pressure inside the actuator will continually fluctuate and therefore the braking torque can be described as a function of two independent variables.

The control system philosophy is to maintain a constant braking torque. This is achieved by varying the pressure inside the actuator during braking to compensate for the natural changes in the value of the friction coefficient. When the friction coefficient decreases, the control system ensures that the pressure increases and likewise, should the friction coefficient increase, then the control system will ensure that the pressure inside the actuator decreases in order to maintain the desired constant braking torque.

Insight into the functioning of the system aided the design of the required control system. Figure 4 describes the process by means of a block diagram.

The following equations exist:

$$F_n = P_b \times A$$

$$F_f = \mu \times F_n$$

$$T_b = F_f \times r$$

F_n = Normal force

F_f = Friction force

P_b = Brake pressure

T_b = Brake torque

μ = Friction coefficient

r = Disk radius

A = Brake piston area

The brake pressure in the calliper acting on the brake piston of area A, results in a force F_n, with which the brake blocks are compressed against the rotating disc. The frictional force F_f thus generated is a function of the normal force. The friction coefficient forms the relationship between the applied normal force and the resultant frictional force. Therefore the system gain varies in accordance with the

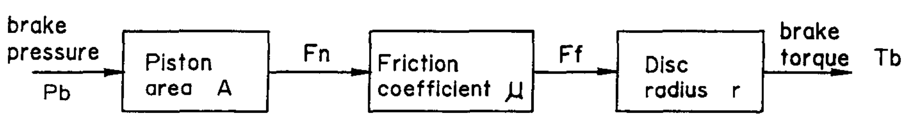

Figure 4 Block diagram representation of the torque - pressure relationship.

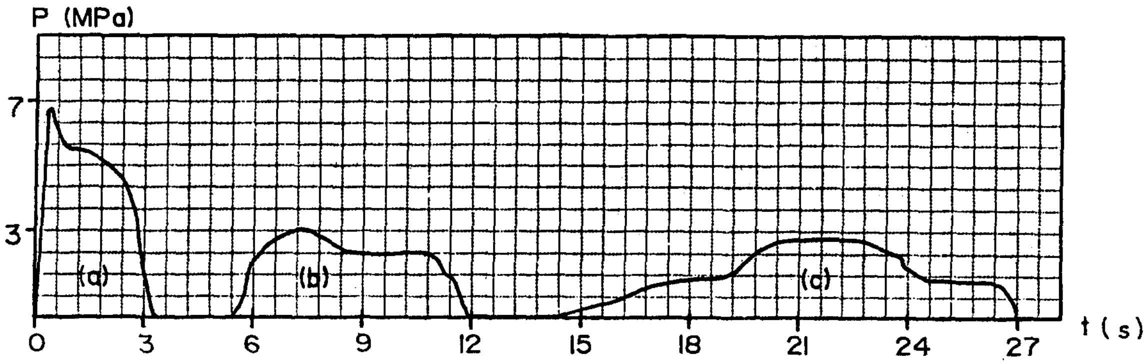

Figure 5 **Brake pressure responses measured in a motor car during (a) an emergency stop, (b) a quick stop and (c) a gradual stop.**

variation in the value of the friction coefficient. Results have shown that the system gain can fluctuate by a factor of up to 5. The product of the frictional force F_f and the disc radius r results in the applied braking torque.

To determine a realistic response time for the control system, a pressure transducer was connected to the hydraulic brake system of a standard motor car. The brake pressure responses for three different stops were recorded as illustrated in Figure 5.

The quickest rise time recorded was for the emergency stop. Figure 5(a) indicates a rise time of 200 ms for this specific stop. Based on these results, the control system's response to a step input signal would have to be in this order. It was decided that a rise time of less than 200 ms would be acceptable for the dynamometer.

A critically damped transient response to a step input signal was required for both the torque and pressure control systems. However, to ensure swiftness of these responses a maximum overshoot of 0.5% would be tolerable.

It was decided that a maximum steady state error of 1% would be acceptable to ensure adequate torque control. It was further specified that the steady state error should not exceed this limitation when considering the wide variation in the value of the friction coefficient.

The above mentioned requirements for the control system have been met in the final design, which will be fully discussed in a paper to be published [1]. It will suffice to mention here that the servo-valve used is a highly complex device that exhibits a high order, non-linear response [2]. Notwithstanding this fact it has been observed in this project that simplified transfer functions may well be used as a basis for the design of the control system.

TYPICAL RESULTS

Figure 6 shows part of the results obtained during a particular brake block test. This specific test sequence comprises of a series of intermittent brake applications at a constant rotational speed of the brake disk. The "ON" time is 5 seconds and the "OFF" time 10 seconds. After each tenth brake application the temperature of the disk is allowed to return to below 40°C, before proceeding with the next ten applications.

Figure 6 shows results for two such series of 10 brake applications. Initially, the pressure in the system started off at 1.6 MPa for a friction coefficient of 3.1 and a preselected constant brake torque. Eventually the value of the friction coefficient fell to a minimum of approximately 0.65. This condition forced the pressure to rise to 8 MPa to maintain the brake torque, as is confirmed by the recordings. It is neccessary to mention that a particular bad friction material was chosen for this test to illustrate the performance of the system. Usually such a large variation in friction coefficient is not encountered.

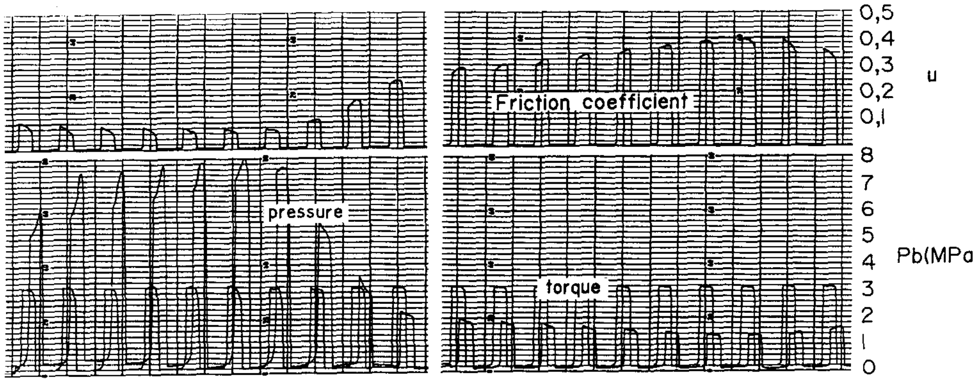

Figure6 Test results showing the change in pressure required to maintain a constant braking torque due to a changing friction coefficient

CONCLUSIONS

A programmable full scale constant torque dynamometer has been developed in conjunction with the research and development division of a leading friction material manufacturer. The evaluation of friction materials for automotive-type disc brake in terms of parameters like relative wear rate, friction coefficient fade, metal pick-up, disc wear, relaxation oscillations and sonic emissions from the interface was made possible. These tests can be conducted under specific controlled conditions at preselected constant speeds and constant power ratings. This system has been in operation for some time now and has played a considerable role in the development of new compositions of friction material for disc brakes. The results shown above are quite impressive when considering the large changes that occur in the value of the friction coefficient, especially when the brake pads are being bedded in.

The possibility also exists to control the retardation rate of the disc's rotational speed whilst simultaneously braking at constant torque. Although this will represent the practical stopping action more closely, such refinement will not really present valuable additional information to the development engineer. For final evaluation, the research and development division prefers to do a series of stop tests on the actual vehicle. Such a final test will then also give information on parameters like friction recovery towards zero speed.

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