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Brake Vibration

Formal studies of brake vibration appear to have been first reported in 1935 by Lamarque and Williams, who were concerned with brake squeak [1]. Subsequent theoretical studies provided a mathematical formulation of the problem and further experimental data. Holography and finite element analyses have provided a unified description of the behavior of the brake assembly (drum, backplate, shoes, and lining in the case of drum brakes and disk and caliper in the case of disk brakes) as it vibrates and have shown that brake vibration is the result of an interplay between the variation of the coefficient of friction as a function of the relative velocity between the brake pad the friction surface (disk or drum) and the masses, equivalent springs, and dampers that comprise the associated mechanical system.

I. BRIEF HISTORICAL OUTLINE

Lamarque and Williams [1] appear to have been the first to suggest that brake vibration was due to a stick-slip frictional phenomenon dependent on the coefficient of friction decreasing as the relative velocity between the friction surfaces increased. Although not explained in detail, the implication was that the brake would engage and the associated mechanical system would deform slightly under the applied load to the point where the shoe and drum configuration would change enough for the elastic forces to cause the shoe and brake to disengage momentarily. Once disengaged, the elasticity of the mechanical system would cause the shoe and drum to snap back to their undistorted configuration fast enough to lower the friction between contact-ing surfaces sufficiently for the components to nearly reassume their original configuration and the process to repeat. In the years that followed, many investigations were conducted to better understand the role of the factors that affected brake and clutch vibration. Only a few of the many contributors to the literature of brake vibration will be discussed explicitly in this section. This partial listing is sufficient, however, to portray the general trends in the years after 1935.

An experimental investigation by Hollmann [2], reported in 1954, indicated that the tendency for frictional vibration increased with the contact pressure and that it also increased with the temperature of the friction materials up to 100°C but then decreased rapidly as the temperature rose above 100°C. After studies of brake squeal on railway vehicles, Broadbent [3] also concluded that brake vibration was dependent on the mechanical linkage used and was associated with a friction coefficient that decreased with increasing velocity between shoe and brake. He also observed that no chatter was found when wooden brakes whose friction coefficient increased with increased slip speed were used. Similar results were found by Sinclair [4], who also reported that the frequency of oscillation was strongly dependent on the equivalent mass and spring constant of the mechanical system that held the brake lining in place in the slider block configuration used in the laboratory model.

Spurr [5] rejected the notion that it was necessary for the friction coefficient to decrease with increasing slip speed in order to have frictional vibration. His experiments with railway block brakes on a railway wheel indicated that vibration was independent of the change in the friction coefficient with slip speed but that it was more likely if the friction coefficient was large. The stick-slip theory of friction appeared to hold as the driving means, however, and Spurr also observed that the frequency of oscillation depended on the associated mechanical system used to force the brake block against the wheel.

In the discussion of Spurr's paper, F. R. Murray introduced what has been called the sprag theory in British literature, involving a sprag (a cantilevered beam pressed against a moving surface), as shown in Figure 1. A somewhat similar configuration was used by Jarvis and Mills [6] in their theoretical and experimental simulation of a caliper disk brake. Their analysis was based on the classical analysis of the normal deflection of a circular disk in terms of Bessel functions and an assumed deflection of the sprag with coefficients chosen to match experimentally observed values. Substitution of these deflections and their time derivatives into the Lagrangian equations of motion resulted in a set of nonlinear partial differential equations whose solution was approximated by what they termed the "slowly varying amplitude and phase" method described in Ref. 7. The solution found in this

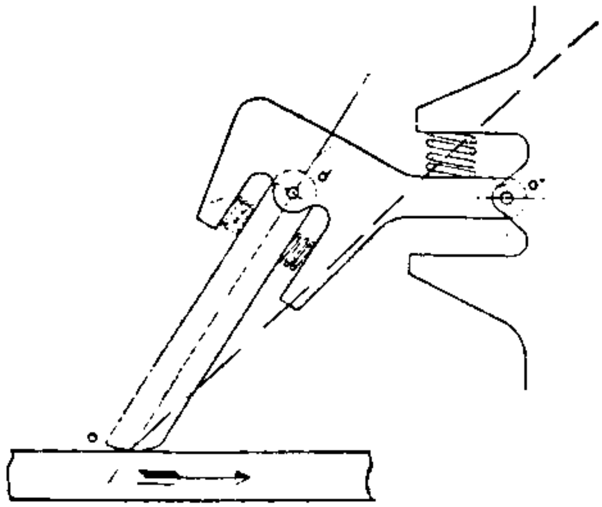


FIGURE 1 The sprag mechanism suggested by Murray in Ref. 5.

manner contained an exponential term which became infinite for certain lateral displacement distributions along the sprag and certain forces between the disk and the sprag. These combinations of sprag deflections and disk forces defined a region of instability which was interpreted to signify large sprag and disk vibration. As formulated, the solution depended only on the mass and elastic properties of the disc and sprag and was not influenced by variation of the coefficient of friction with slip velocity.

Although the solution so obtained agreed with the measured regions of instability in terms of the slope of the sprag relative to the plane of the disk, the shape of the calculated boundary curve for the region of instability differed markedly from the experimentally measured curve. The measured curve was concave upward but the theoretical curve was concave downward.

Based on these results, Jarvis and Mills concluded that brake vibration could be avoided by careful design of the disk and caliper without specifying a particular variation for the coefficient of friction. In the published discussion Spurr agreed that brake vibration could be controlled by careful design of the associated mechanical system.

In spite of the comments by Broadbent, Sinclair, Spurr, and Jarvis and Mills on the effect of the associated mechanical system in determining the vibration resulting from the friction excitation, various authors, such as North [8], have considered the papers by Spurr and by Jarvis and Mills as expounding a theory different from those of Broadbent, Sinclair, and others, who discussed the effect of a negative slope for the curve of the friction coefficient as a function of the relative speed between friction surfaces, the slip speed. A more careful reading of their papers, however, shows that while they may have devoted more space to a discussion of friction characteristics, they were definitely aware of the importance of the response characteristics of the brake's activation system in determining the nature of the resulting oscillations. There has, therefore, been general agreement as to the nature of the problem even though the aspects emphasized have changed with time.

Analysis of the mechanical system studied by Spurr, by Jarvis and Mills, and by North [9] was extended by Millner [10] to consider the effect of a single pad at the contact region of the sprag and the disk. This analysis, which was based on a highly simplified lumped-parameter model, implied that nonlinear pad compression may cause the brake to vibrate within discrete bands of actuating pressure.

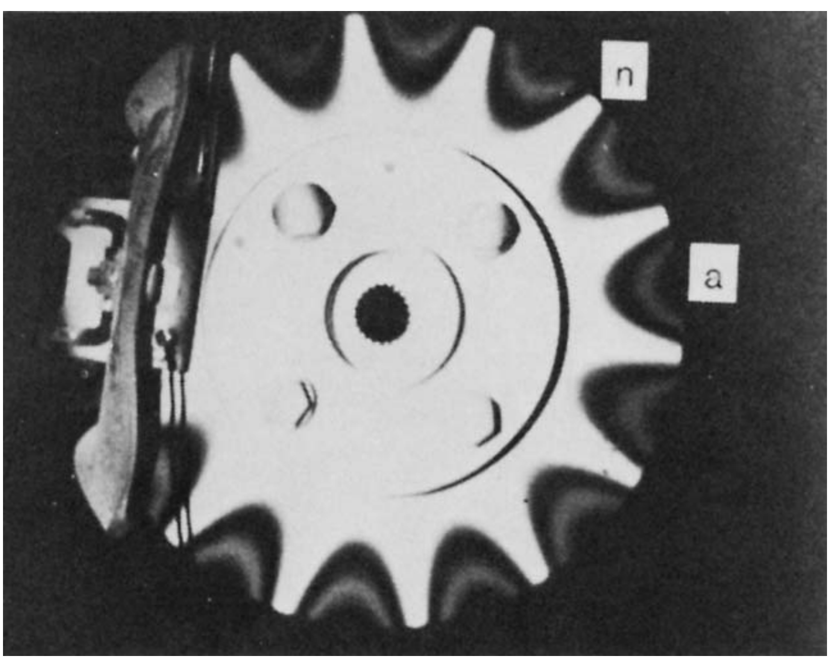


Figure 2 Vibration mode of a yoke-type disk brake at 10 kHz. (Reprinted with permission; © 1984 Society of Automotive Engineers, Inc.)

II. RECENT EXPERIMENTAL DATA

Using experimental techniques not available in the 1960s for the examination of both disc and drum brake vibration, Felske, Hoppe, and Matthäi employed holographic interferometry to demonstrate conclusively that it is the caliper vibration that is the major contributor to brake noise from disk brakes [11] and the backplate vibration that is the major contributor from drum brakes [12]. Typical standing wave shapes, or the nodal patterns, for the disk are shown in Figures 2 and 3. Vibration of the caliper is shown in Figures 4 and 5. The alternating black and white line boundaries represent contour lines, or elevation lines, on the caliper and disk and consequently measure the deflection of the disk and caliper in a direction perpendicular to the plane of the photograph, as indicated in Figure 6 for an antinode on the disk.

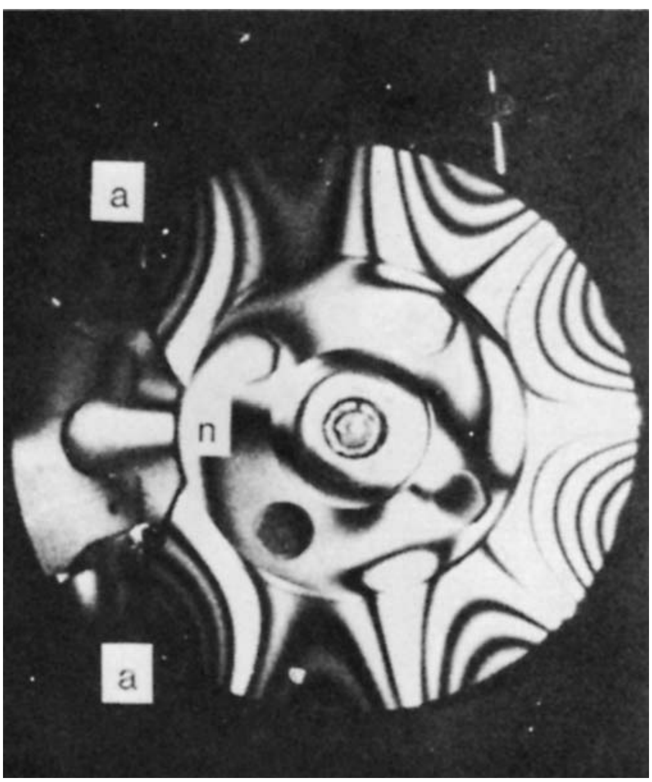


FIGURE 3 Vibration of a first-type disk brake at kHz. (Reprinted with permission; © 1984 Society of Automotive Engineers, Inc.)

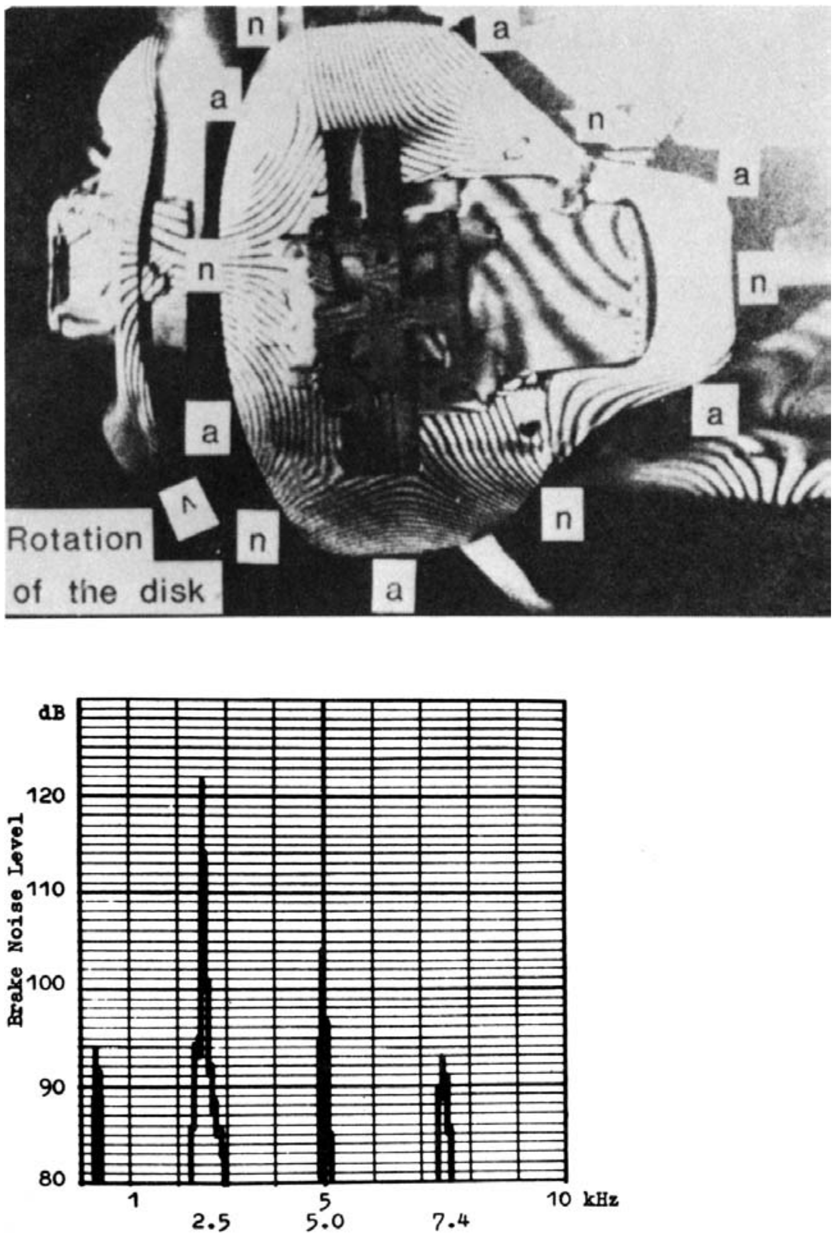


FIGURE 4 Reconstruction of a double-pulsed hologram of a squealing yoke-type caliper exposed at a noise level of 120 dB at 2.5 kHz along with its frequency spectrum. (Reprinted with permission; © 1984 Society of Automotive Engineers, Inc.)

Typical modes of backplate vibration in the first mode are shown in Figure 7 and its nodal lines, indicated by dashed curves in Figure 7, are shown alone in Figure 8. Figures 9 and 10 show a backplate before and after a raised portion (Figure 11) was added to reduce the frequency of the noise generated.

III. FINITE ELEMENT ANALYSIS

Murakami, Tsudada, and Kitamura [13] reported on a finite element analysis of automotive disk brakes to compare the calculated resonance frequencies with previous measurements of brake squeal on a chassis dynamometer and to associate them with calculated deformation of the brake components. Secondary low-frequency squeal from 2 to 3 kHz and primary high-frequency squeal from about 5.5 to 10.5 kHz, as shown in the histogram in Figure 12, correlated well with the clustering of frequencies found for the brake disk, cylinder, pad, and torque member, shown in Figure 13. Calculated disk and caliper frequency modes where verified by holographic interferometry and by accelerometer measurements in the case of the caliper, or cylinder.

Although the cluster of frequencies correlated with the primary and secondary squeal regions, several component resonant frequencies between 3 and 5.5 kHz did not result in brake squeal in this frequency range. This implied that the driving force, the stick-slip phenomenon, did not excite these frequencies and that they were not excited by structural coupling. Linearized, seven-degree-of-freedom analysis of the system using the lumped-parameter model, as shown in Figure 14, gave solutions of the form

$$\theta = Ae^{\alpha t} \sin(\omega t + \phi) \tag{3-1}$$

for each of the seven variables in which exponent α, which determined the regions of instability in the Jarvis and Mills analysis, was termed the *squeal index*. This analysis indicated that the squeal index was related to the negative gradient of the friction coefficient, as illustrated in Figure 15, which also shows test results for two test brake lining pads. These results also correlate with Spurr's finding that the squeal probability increased as the friction co-efficient increased. The low squeal index between 3.0 and 5.5 kHz and its slope also correlate with the histogram shown in Figure 12, which implies that negative slope of the friction coefficient versus velocity curve is one of several significant parameters in the generation of brake squeal. Analysis of the influence of the torque member, shown in Figure 16, indicated that its influence was also minimum in the vicinity of 3 kHz, as shown in Figure 17, which also accounts for the low squeal amplitude between 3.0 and 5.5. kHz.

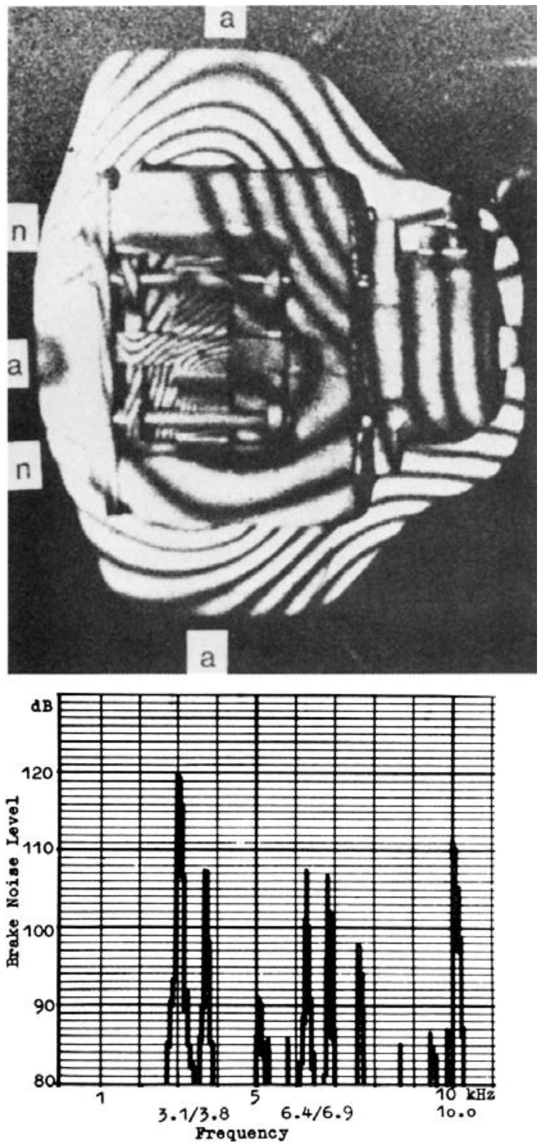


Figure 5 Vibration mode of a squealing yoke-type caliper exposed at 120 dB at 3.1 kHz along with its frequency spectrum. Reconstruction of a double-pulsed hologram. (Reprinted with permission; © 1984 Society of Automotive Engineers, Inc.)

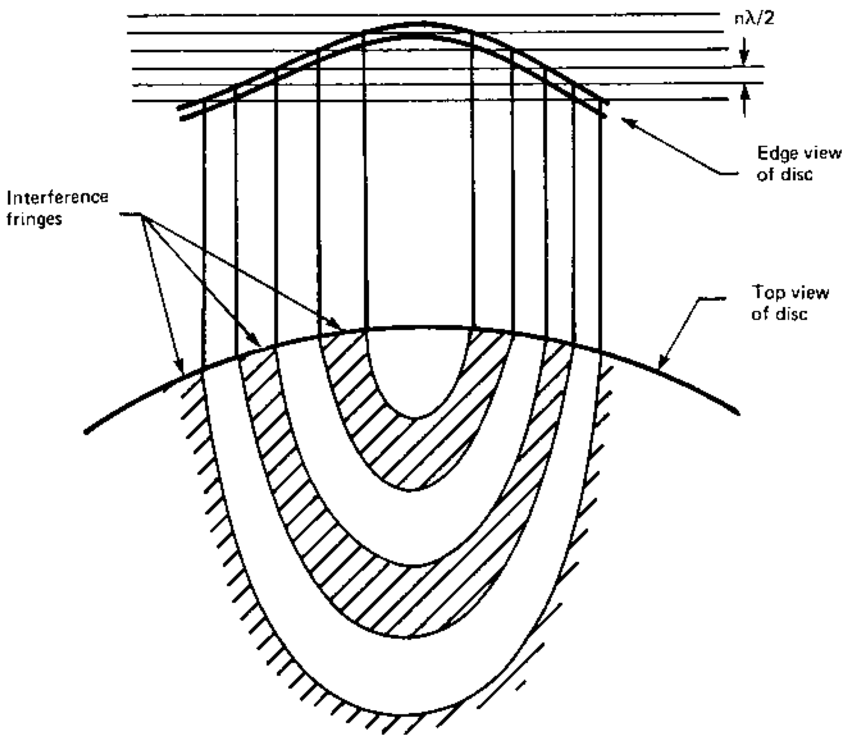


FIGURE 6 Relation between contour lines in an interference hologram and the displacement perpendicular to the plane of the interference pattern.

Calculated mode shapes for the disk, pad, cylinder, and torque member are displayed in Figures 18 through 21.

IV. CALIPER BRAKE NOISE REDUCTION

An example of caliper brake redesign in an attempt to reduce brake noise is that shown in Figure 22. According to the manufacturer, the one-piece sound insulator and backplate shown in that figure are central to its noise reduction. Details are not given because the design is patented and proprietary. Examination of the cross section shown in Figure 22, however, suggests that the physical dimensions and the material properties of the different regions in

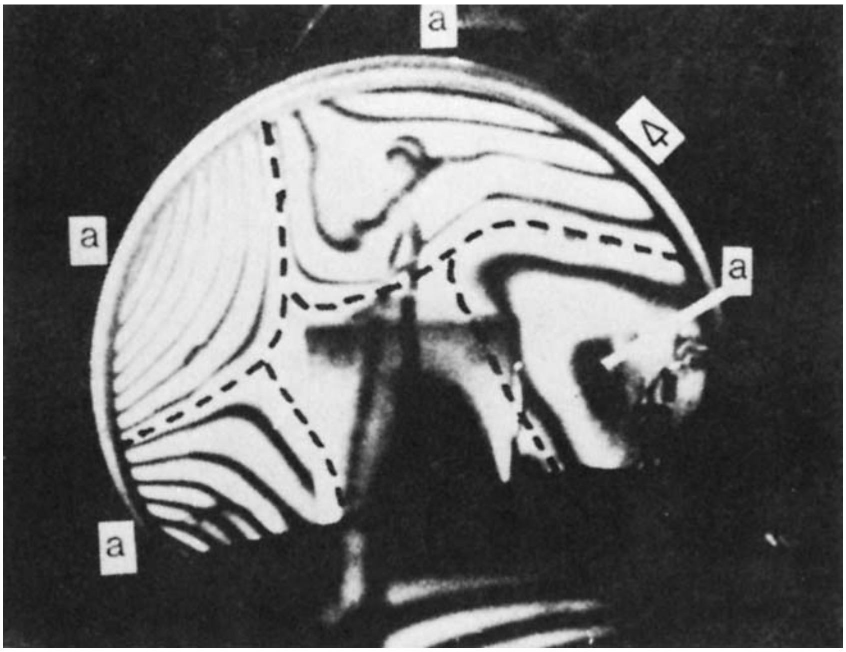


FIGURE 7 First vibration mode of a backplate on 200 × 400 mm drum brake at 1.1 kHz. Broken nodal lines superimposed on reconstructed double-pulsed hologram. (Reprinted with permission; © 1984 Society of Automotive Engineers, Inc.)

the insulator strip above the friction material may serve to dampen and suppress high-frequency vibration, and the pad material, central plate, and grooves may aid to suppress low-frequency vibration.

Grooves in brake pads of caliper brakes also aid in removing water when the brake operates in wet conditions.

Longer pad life is said to be another advantage of this design, because the insulator and backplate tend to absorb and dissipate the heat generated during stopping over a greater surface than in caliper brake pads that are not of this design. This improved heat dissipation is said to be an advantage because the heat generated causes the pad material to deteriorate.

According to U.S. patent 5,433,194, the proprietary lining material, which may also contribute to the noise reduction, is composed of organic, carbonaceous, metal, and mineral particles, rubber/resin curatives, and a corrosion inhibitor, in several different proportions.

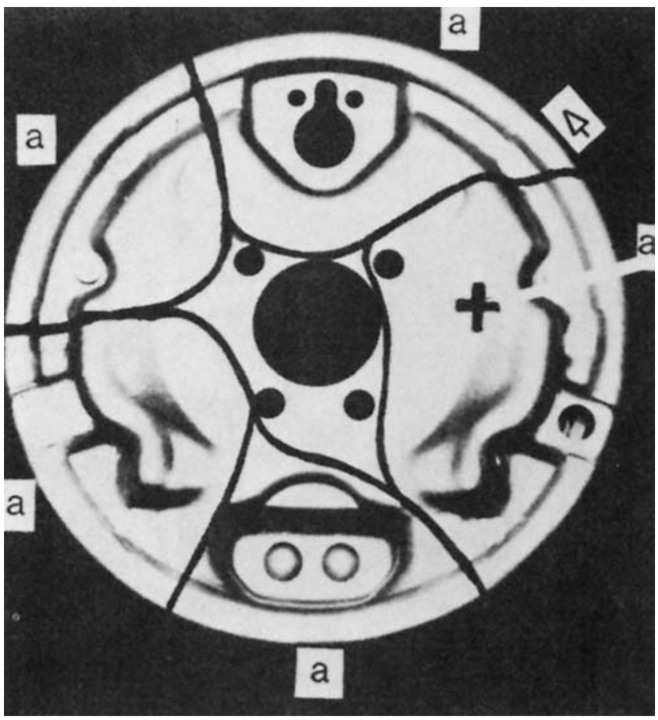


FIGURE 8 Nodal lines for the first vibration mode superimposed on a photograph of the entire back plate. (Reprinted with permission; © 1984 Society of Automotive Engineers, Inc.)

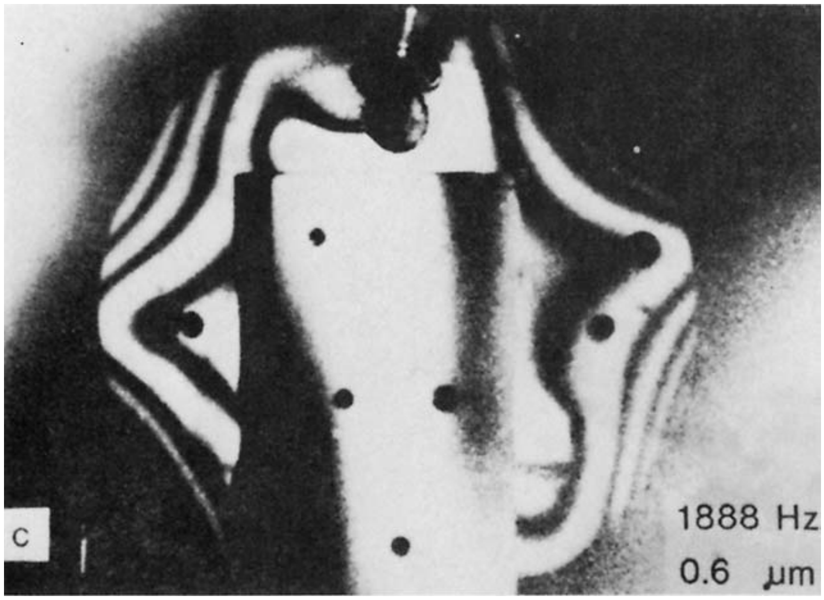


Figure 9 Backplate modes at 1888 Hz for a 180 × 30 mm drum brake. Noisy configuration. (Reprinted with permission, © 1984 Society of Automotive Engineers, Inc.)

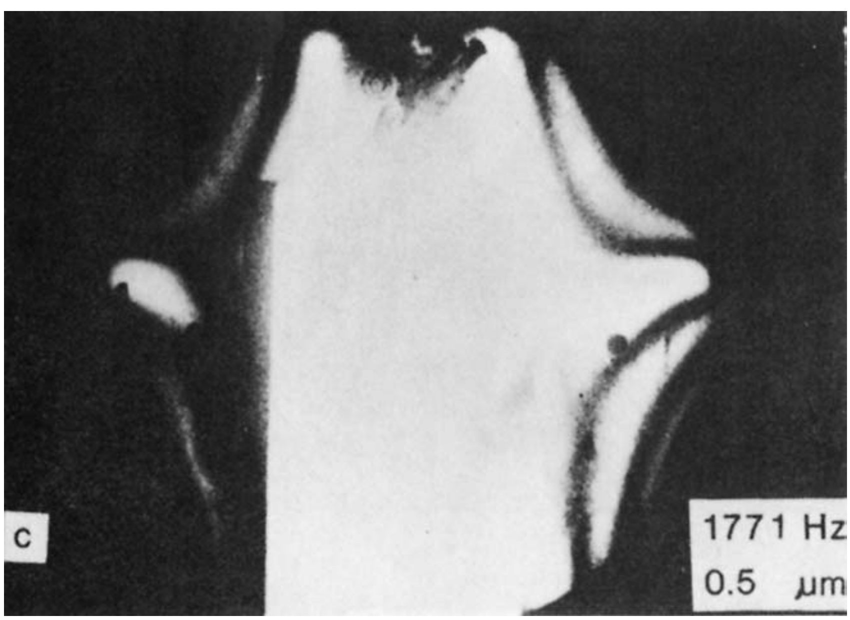


Figure 10 Backplate modes at 1771 Hz after backplate modification for a 180 × 30 mm drum brake. Quiet configuration. (Reprinted with permission; © 1984 Society of Automotive Engineers, Inc.)

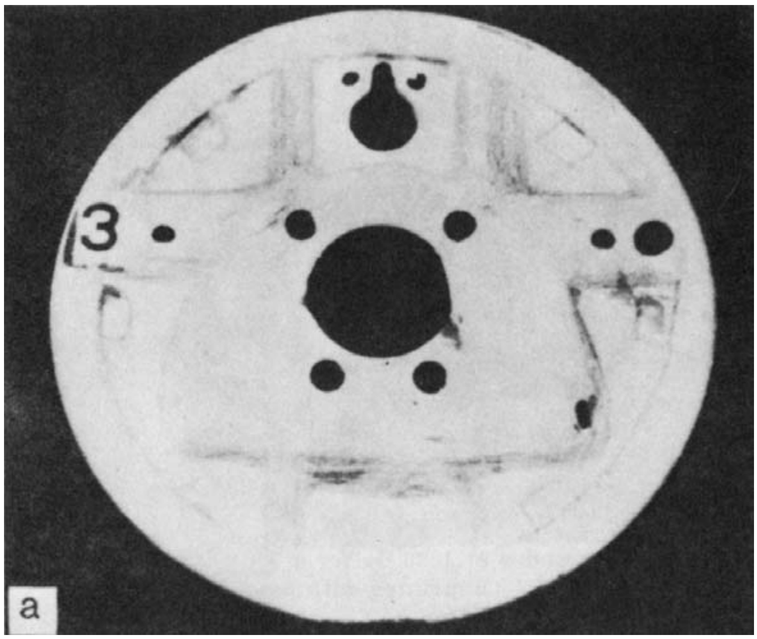


Figure 11 Backplate modified by adding stiffening ridges and a central raised portion, which reduced squeal by 6.5 dB at low frequency (820–990 Hz) and 4.8 dB at high frequency (1681–1888 Hz).

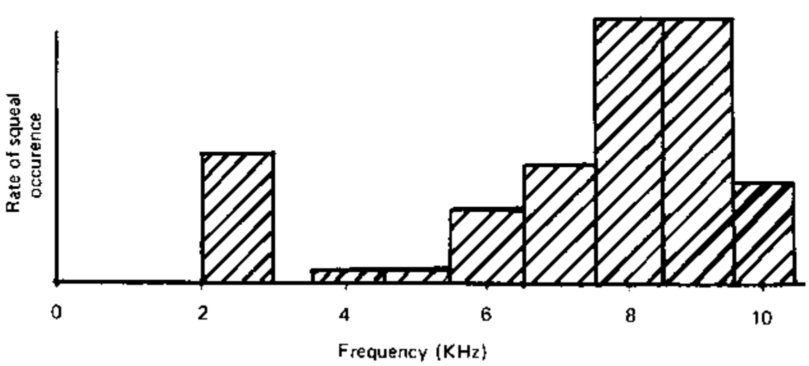


Figure 12 Squeal histogram for a vehicle test on a chassis dynomometer.

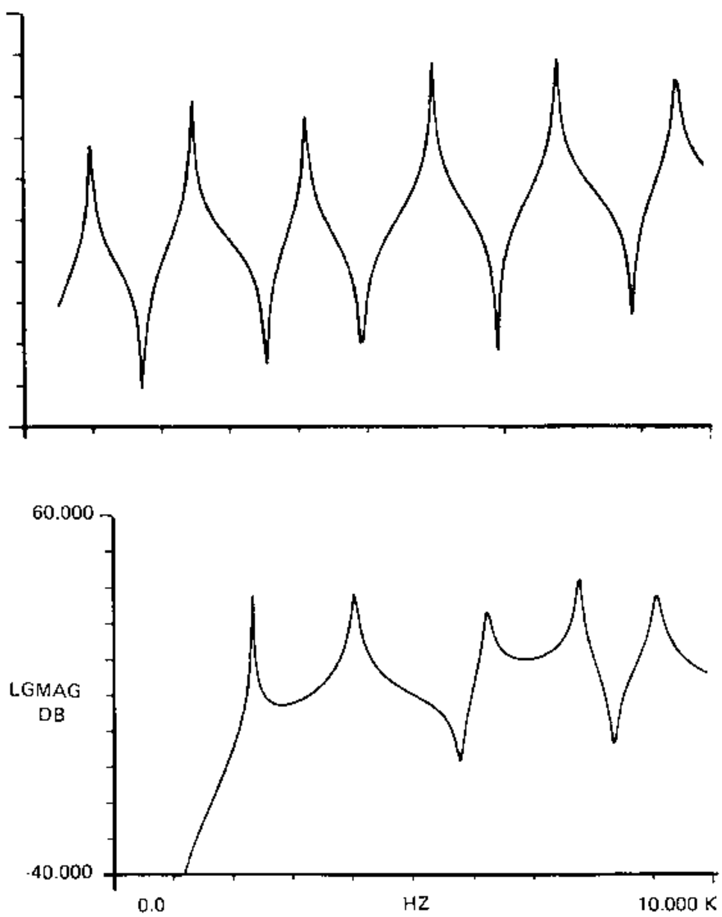


FIGURE 13 The natural, or resonant, frequencies of each component of the automotive disk brake tested.

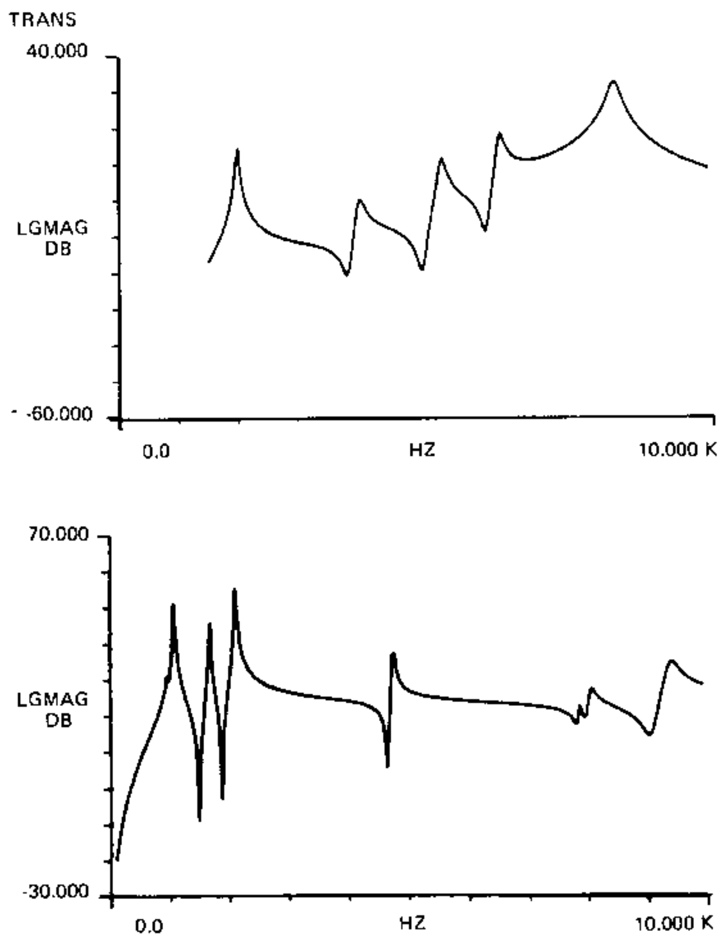


FIGURE 13 Continued.

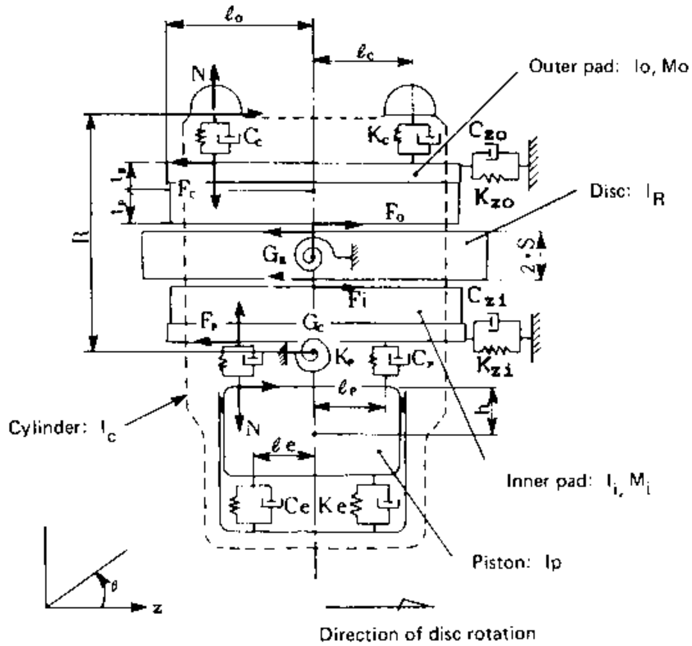


FIGURE 14 Sketch of a caliper and disk showing the spring and dashpot associated with each of the masses involved.

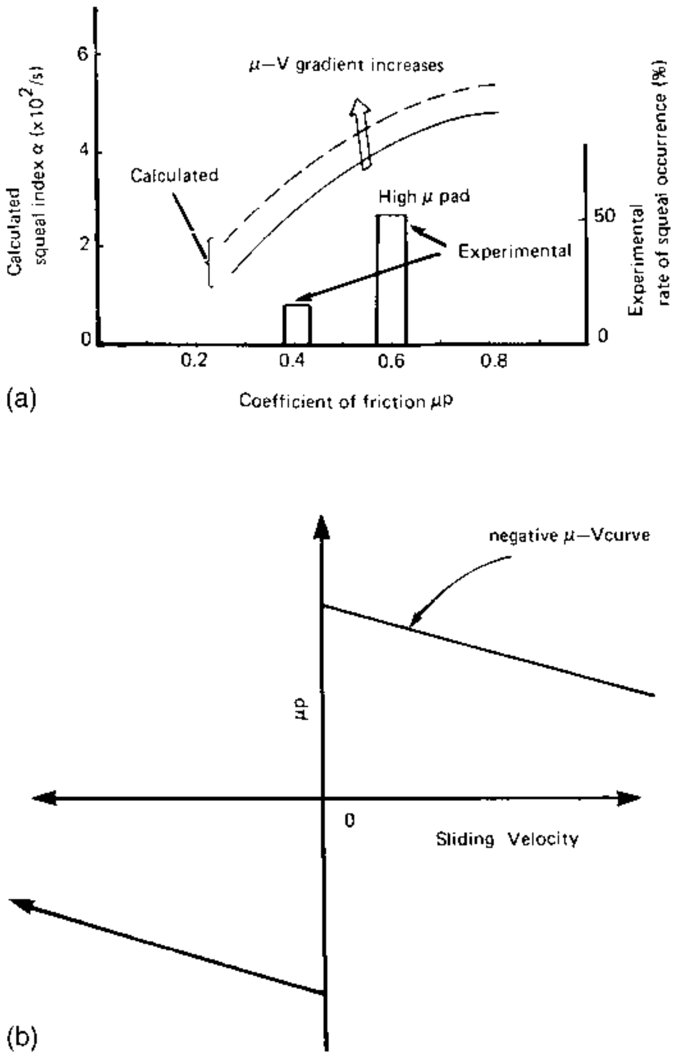


Figure 15 (a) Variation of squeal index α with variation in the friction coefficient; (b) assumed dependence of the friction coefficient μ on the relative, or slip, velocity.

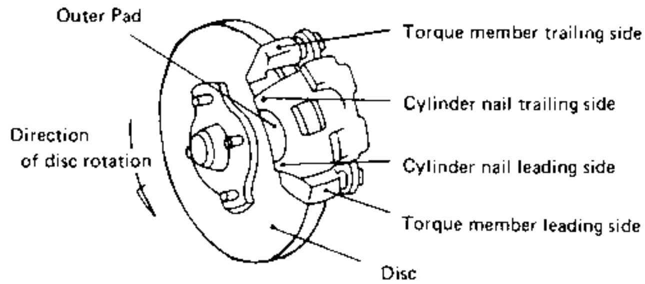


FIGURE 16 Sketch of the caliper and disk brake assembly.

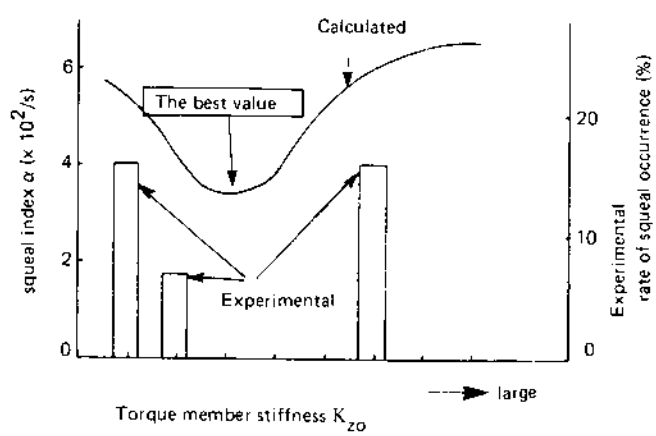
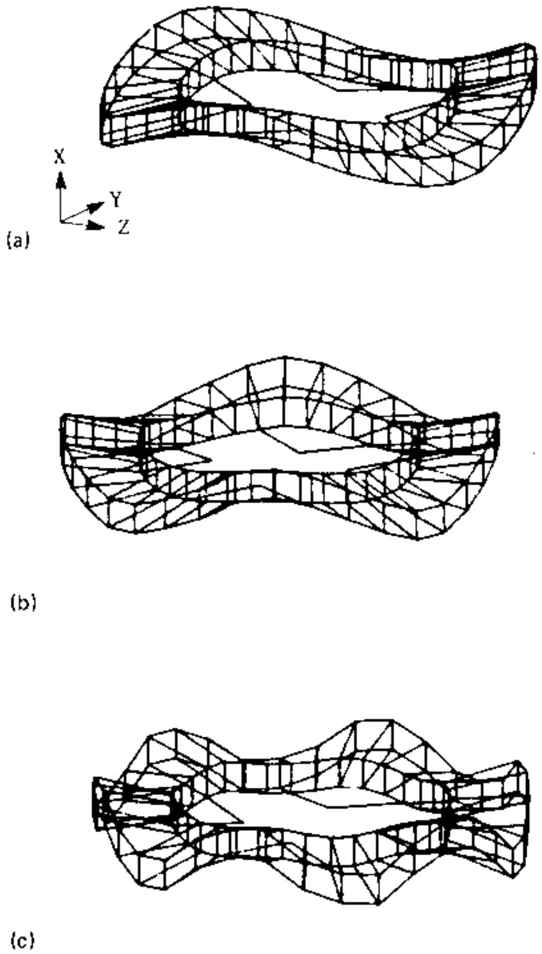
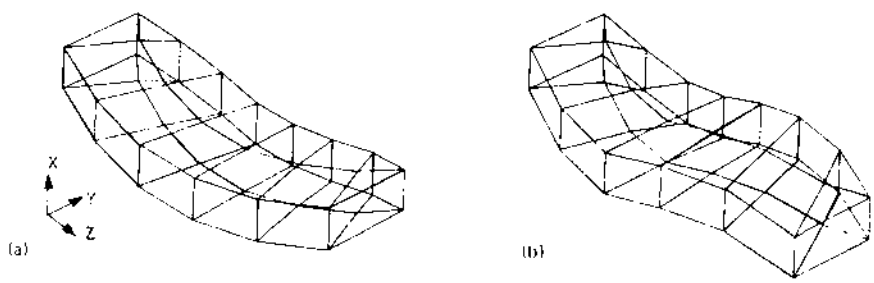


FIGURE 17 Dependence of the squeal index on the torque member stiffness.



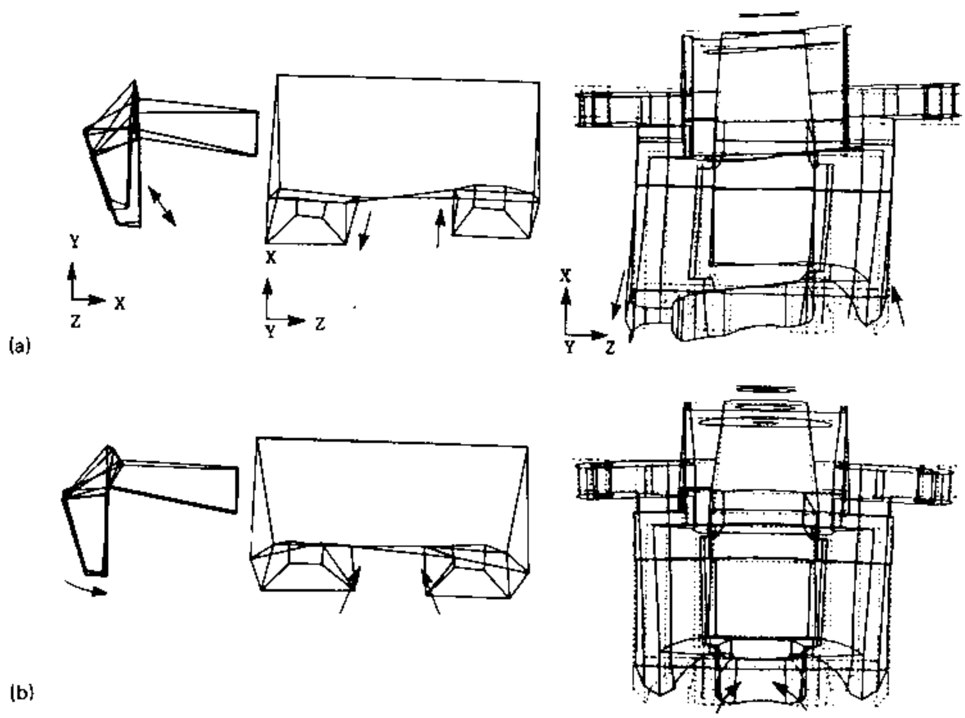
	Modal analysis	FEM	Error
(a)	2.47 kHz	2.47	0%
(b)	4.14	4.25	+2.7%
(c)	7.82	8.31	+6.3%

Figure 18 Examples of calculated mode shapes of the disk.



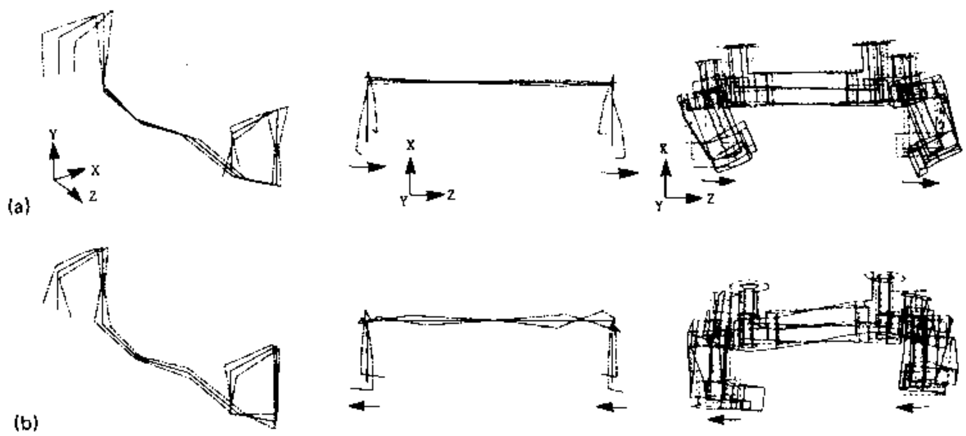
	Modal analysis	FEM	Error
(a)	4.06 kHz	3.82	-6.1%
(b)	8.36	9.06	+8.4%

FIGURE 19 Examples of mode shapes for the brake pad.



	Modal analysis	FEM	Error
(a)	2.32 kHz	2.45	+5.6%
(b)	3.97	4.00	+0.8%

FIGURE 20 Examples of the mode shapes for the cylinder portion of the caliper. (a) and (b) Modal analysis (left); FEM (right).



	Modal analysis	FEM	Error
(a)	2.14 kHz	2.14	0%
(b)	8.06	8.17	+1.3%

FIGURE 21 Examples of mode shapes for the torque member of the caliper. (a) and (b) Modal analysis (left); FEM (right).

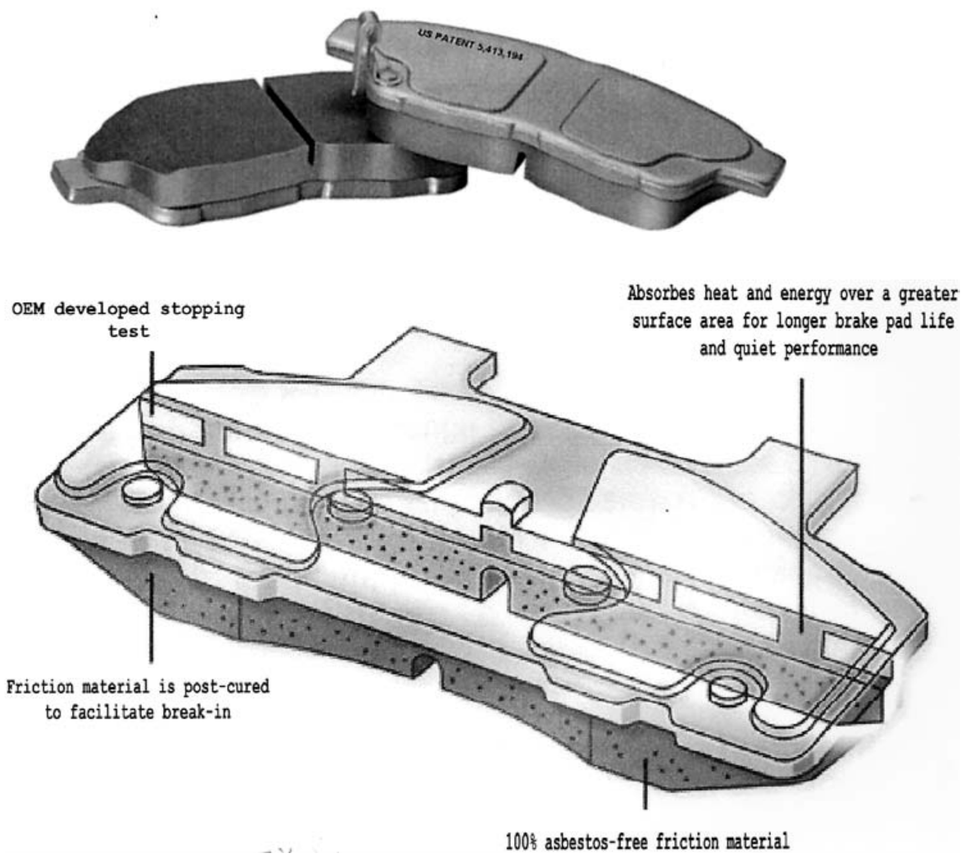


FIGURE 22 Photograph and cross section of a ThermoQuiet brake pad. (Courtesy Wagner Brake Products, Federal-Mogul Corp., Southfield, MI.)

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