Analysis of friction-induced vibration leading to “EEK” noise in a dry friction clutch

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This article deals with the “EEK” sound that may occur during engagement of a dry friction clutch in vehicles with manual transmissions. Experimental data showed that near the full engagement point of the clutch, the pressure plate suddenly starts to vibrate in the rigid-body wobbling mode with a frequency close to the first natural frequency of the pressure plate and clutch disk cushion. The self-excited vibration exhibits typical signs of a dynamic instability associated with a constant friction coefficient. A linearized lumped-parameter model of the clutch was developed to explain the physical phenomenon. The effect of key parameters on system stability was examined by calculating the complex eigensolutions. Results of the analytical study were in agreement with experimental observations. For instance, it was observed that the instability of the rigid-body wobbling mode was controlled by friction forces. This mode may, however, be also affected by the first bending mode of the pressure plate. A stiffer plate could lead to an improved design with a reduced tendency to produce the “EEK” sound. © 2005 Institute of Noise Control Engineering.

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1 INTRODUCTION

A squealing sound may be produced during the last part of the engagement of a dry friction clutch in vehicles with manual transmissions. The strongly annoying sound of the transient event is known as “EEK” noise, which may dominate an overall perception of vehicle sound quality. Some consumers might even prematurely change the clutch in an effort to eliminate or reduce the disturbing noise.

Figure 1 illustrates a typical arrangement of a pressure plate and a clutch disk used to transmit torque from the engine to the transmission and ultimately to the vehicle’s driven wheels. Depressing the clutch pedal (not shown) moves a throw-out bearing (not shown) toward a diaphragm and, in turn, causes the pressure plate to move away from the clutch disk into the disengaged position where the engine is uncoupled from the transmission. Letting up on the clutch pedal allows the pressure plate to squeeze the clutch disk against the flywheel mounted on the engine’s crankshaft in the engaged position. A critical position is achieved at the point of “liftoff” where, in reaction to the force applied by the throw-out bearing, the pressure plate is just disengaged from the clutch disk.

Despite many experimental studies, this friction-induced “EEK” sound (like many other friction problems [1]) remained poorly understood. One hypothesis was that the “EEK” sound was triggered by a dynamic instability of the subsystem composed of the clutch pressure plate and the clutch disk cushion. It had been observed experimentally [2] that, during the “EEK” event, the pressure plate vibrated in a rigid-body wobbling (out-of-plane) motion corresponding to the natural frequency of the pressure-plate/disk-cushion system. Experimental studies showed that the modal properties of the pressure plate, as a result of the fabrication process, seemed to play a significant role in the mechanism(s) that cause the “EEK” sound. To clarify some of these issues, the analytical model described in this article was developed and used to identify key parameters affecting the stability of a dry-friction-clutch system.

2 EXPERIMENTAL MEASUREMENTS AND PROBLEM FORMULATION

Two conditions are typically necessary before the transient “EEK” sound (Figure 2) can be induced.

First, the engine speed should be within the 1500 to 2500 rev/min range. The transmission speed does not seem to have a significant effect. Since the slip speed is relatively high, a constant coefficient of friction $\mu$ at the threshold of “EEK” may be assumed. These conditions suggest that the “EEK” sound may not follow the symptoms of classical stick-slip phenomenon [1].
Second, the clutch pedal motion (or throw-out bearing travel) has to be such that the clutch is close to full engagement at the onset of the “EEK” sound. Figure 2 shows this particular position (where the “EEK” sound begins to occur) where the pressure plate and clutch disk cushion are already in contact, but not yet fully engaged. The two vertical dashed lines in Figure 2 indicate the times \( t_1 \) and \( t_2 \) at the onset and cessation of the strong “EEK” sound as noted by the rapid increase and decrease of the sound pressure.

Measurements of the spectra of the “EEK” sound exhibit a dominant frequency \( f_w \), related to the bearing travel position, of approximately 450 Hz to 500 Hz, as shown at various relative times in Figure 3(a). This frequency has been correlated to

**Fig. 2**—“EEK” sound as measured in the interior of a vehicle as a clutch is being engaged and then disengaged as correlated with the variation of engine speed and the position of the clutch throw-out bearing.

**Fig. 3**—Spectra of the “EEK” sound; (a) Principal frequency components as the clutch is engaged; (b) Magnitude of the spectral components of “EEK” sound as measured by a spectrum analyzer before the full engagement of the clutch at relative time \( t_1 \), where the component at the fundamental frequency \( f_b \) of the bending mode was dominant at approximately 950 Hz; (c) at time \( t_2 \), where the component at the fundamental frequency \( f_w \) of the wobbling mode was dominant at approximately 500 Hz. [Vertical lines indicate the fundamental, second harmonic, and third harmonic of the wobbling-mode frequency in (b) and (c).]
the wobbling rigid-body motions of the pressure plate (in both out-of-plane rotations) from dynamometer tests [2] and from a single-degree-of-freedom (SDOF) model that included the inertia of the pressure plate and the stiffness of the clutch diaphragm coupled with the stiffness of the clutch cushion. The pressure plate and clutch disk cushion were assumed to be the main components that control the “EEK” phenomenon. This assumption was reinforced by many empirical experiments that showed that elements external to the clutch (gears, release system, engine pulsations, etc.) as well as internal to the clutch (diaphragm and cover) do not significantly influence the tendency to produce the “EEK” sound [2].

Further, a change in the pressure plate material from gray-iron (Pressure Plate G) to vermicular iron (Pressure Plate V) seemed to reduce the occurrence of “EEK”. Such a change of material could alter the modal properties of the pressure plate, as found in some brake-squeal studies [3,4]. Experimental modal studies showed that a pressure plate made from Plate V material is essentially stiffer than a pressure plate made from Plate G material. Thus, the influence of pressure-plate elastic deformations needed to be investigated.

From experimental modal analyses, the second frequency \( f_y \), between 900 Hz and 1000 Hz as shown in Figure 3(a), could be linked to the first bending mode of the plate. Figure 3(b) shows that the prominence of this bending mode (at frequency \( f_y \)) at relative time \( t_y \) is quickly replaced by the dominance of the wobbling mode (at frequency \( f_w \) and harmonics) 0.3 s later at relative time \( t_z \) in Figure 3(c).

This article develops a preliminary simulation model for the clutch assembly. At the threshold of the “EEK” sound, vibration amplitudes were considered sufficiently small to build a linearized model that would reproduce the trends of the observed dynamic instabilities. An analytical lumped parameter approach was preferred over the structural finite-element method. The analytical model examines a self-excited system without any external torque excitations. The pressure plate was assigned an initial angle in order to examine the growth of vibration amplitude with time and demonstrate the existence of dynamic instability.

3 ANALYTICAL MODEL

The lumped parameter model of Figure 4 incorporates the inertia of a deformable pressure plate along with the clutch cushion stiffness \( k_z \) that was divided into four springs of stiffnesses \( k_{xf}, k_{yr}, k_{xf}, \) and \( k_{yr} \), where subscripts \( x \) and \( y \) indicate coordinate axes and \( f \) and \( r \) indicate the direction of the motion toward the forward and rear of the power-transmission axis, respectively.

Figure 4(a) shows the model for the fixed-base flywheel and the clutch disk. The +z-axis points toward the engine and is aligned with the axis of the crankshaft. The clutch disk, Figure 4(b), was modeled as four masses, \( m_f \), at segments A, B, C, and D and connected by four springs of stiffness \( k_f \).

In order to consider the nonlinear characteristic of the clutch-disk cushion, the stiffnesses \( k_{xf} \) and \( k_{yr} \) were respectively divided and multiplied by a non-dimensional amplitude ratio \( \lambda \), as shown in Figure 5 and in the following, where the stiffness of an individual spring was assumed to equal one fourth of the total clutch stiffness \( k_z \).

Assume that the bearing travel is such that the “EEK” threshold point (P in Figure 5) is reached. This instant would correspond to a constant plate liftoff \( v = v_{EEK} \). Liftoff corresponds to the displacement of the plate from its engaged position to a disengaged position. Now suppose that the plate is already vibrating about the x-axis with amplitude \( \theta_{y0} \). This
vibration will change the normal spring forces at locations A and B as shown in Figure 5. Consequently, the stiffnesses \( k_x \) and \( k_y \) were obtained by assuming a constant stiffness slope around point P. The different spring forces are used in Equations (1) and (2).

\[
k_{xy} = \frac{k_x}{4\lambda}
\]

(1)

\[
k_{yx} = \frac{\lambda}{4} k_x
\]

(2)

The ratio \( \lambda \) could be used to incorporate an asymmetry into the resulting stiffness matrix. For instance, \( \lambda = 0.9 \) in Equations (1) and (2).

Friction forces from the clutch disk cushion were also included by assuming a constant coefficient of friction \( \mu \) due to a high slip speed (>700 rev/min). For the six degree-of-freedom (DOF) model of Figure 4, two DOFs describe the rigid-body "out-of-plane" rotations (wobbling mode) and four additional translational DOFs (normal to the pressure plate) depict the first two nodal-diameter bending modes of the pressure plate. By choosing an appropriate lumped bending stiffness \( k_x \), this bending mode was created in our model with a frequency close to the observed frequency \( f_y \). Setting one of the modal masses, \( m_i \) in Figure 4(b) to correspond to a quarter of the total plate mass was believed to be a reasonable approximation of the modal mass.

Next, the free-body diagrams are drawn as shown in Figure 6. Here, even though it is represented as a rigid body for the sake of clarity, the pressure plate can also elastically deform as explained before. The normal spring forces \( (N) \) and the corresponding friction forces \( (F) \) are therefore expressed as follows along the x and y axes and in the forward and rearward directions:

\[
N_{xf} = k_{xf}(z_{xf} - r\theta_x)
\]

(3)

\[
N_{xt} = k_{xt}(z_{xt} + r\theta_y)
\]

(4)

\[
N_{yf} = k_{yf}(z_{yf} + r\theta_y)
\]

(5)

\[
N_{yt} = k_{yt}(z_{yt} - r\theta_x)
\]

(6)

\[
F_{xf} = \mu N_{xf}
\]

(7)

\[
F_{xt} = \mu N_{xt}
\]

(8)

\[
F_{yf} = \mu N_{yf}
\]

(9)

\[
F_{yt} = \mu N_{yt}
\]

(10)

where \( r \) is the radius of the pressure plate, \( \theta \) is a rotation about the x or y axis, and \( \mu \) is the coefficient of friction.

The resulting governing equations for a linear, undamped and unforced system were obtained in a matrix form in terms of the mass \( [M] \) and stiffness \( [K] \) matrices and the generalized coordinate vector \( \{X\} \).

\[
[M]\{\dot{X}\} + [K]\{X\} = \{0\}
\]

(11)

where

\[
X^T = [\theta_x \ \theta_y \ z_{xf} \ z_{xt} \ z_{yf} \ z_{yt}]
\]

(12)

\[
M = \text{diag}(I_x \ I_y \ m_f \ m_f \ m_f \ m_f)
\]

(13)

and

\[
K = \begin{bmatrix}
0 & -\Gamma_x & -\Gamma_x & -\Gamma_x & -\Gamma_x & -\Gamma_x \\
-\Gamma_x & 0 & -\Gamma_y & -\Gamma_y & -\Gamma_y & -\Gamma_y \\
-\Gamma_x & -\Gamma_y & 0 & -\Gamma_x & -\Gamma_x & -\Gamma_x \\
-\Gamma_x & -\Gamma_x & -\Gamma_x & 0 & -\Gamma_x & -\Gamma_x \\
-\Gamma_x & -\Gamma_x & -\Gamma_x & -\Gamma_x & 0 & -\Gamma_x \\
-\Gamma_x & -\Gamma_x & -\Gamma_x & -\Gamma_x & -\Gamma_x & 0
\end{bmatrix}
\]

(14)

with

\[
\Gamma_{xf} = 2rk_x + \mu(2k_x + k_{xf})
\]

(15)

\[
\Gamma_{xt} = 2rk_y + \mu(2k_y + k_{xt})
\]

(16)
\[
\begin{align*}
\Gamma_{rf} &= 2rk_i + \mu(2k_i + k_{rf}) \\
\Gamma_{rf} &= 2rk_i - \mu(2k_i + k_{rf})
\end{align*}
\]  

(17)  

(18)

\[\sigma > 0 \quad \text{stable} \quad \sigma < 0 \quad \text{unstable} \]

\[\begin{array}{c}
\text{Static equilibrium} \\
\text{Dynamic motion}
\end{array}\]

\[\begin{array}{c}
\text{Friction} \\
\text{forces}
\end{array}\]

\[\begin{array}{c}
\text{Plate G} \\
\text{Plate V}
\end{array}\]

4 STABILITY STUDIES AND DESIGN GUIDELINES

The asymmetric nature of \([K]\) is controlled by the coefficient of friction \(\mu\) and amplitude ratio \(\lambda\) and yields complex eigenvalues \(\lambda = \sigma \pm \omega_i\), where the imaginary part \(\omega_i\) is the damped natural frequency (in rad/s) of mode index \(s\) and the real part \(\sigma\) is an index of dynamic stability. When \(\sigma\) is positive, the amplitude of vibration grows exponentially in time. The effect of \(\mu\) on \(\lambda\) is illustrated in Figure 7. Two distinct wobbling modes with frequencies \(f_i\) and \(f_c\) (in Hz) can be seen, where the difference between the two frequencies was found to be related to the ratio \(\lambda\) for low values of \(\mu\). Nevertheless, as \(\mu\) is increased, the coupling between the two modes is enhanced. It results in two modes (one stable and one unstable) at \(\mu = 0.35\) (a typical value for clutch disk cushions or linings). Essentially, frequencies \(f_i\) and \(f_c\) merge to become the same frequency \(f_c\).

The effect of the pressure-plate structural stiffness \(k_i\) is investigated in Figure 8, which diagrams the natural frequencies and real parts of the wobbling modes. As \(k_i\) is increased, one passes from an unstable region (for Plate G) to a stable region (for Plate V). These results are in agreement with experimental observations [2]. Therefore, Young’s modulus does have a significant impact on the stability of a pressure-plate clutch-disk system and consequently on the generation of the “EEK” sound.

Figure 9 shows an animation of the complex wobbling modes that was developed to visualize the phase differences. As the pressure plate wobbles, a circle indicates the location along the edge that has the maximum positive amplitude (in the \(z\)-direction). At \(\mu = 0.35\), this circle actually moves along the edge of the plate in the direction of the friction forces for the stable mode (represented by the eigenvector \(\psi_i\)) and in the opposite direction for an unstable mode (denoted by eigenvector \(\psi_c\)). This result means that the rotation about the \(z\)-axis (denoted as dynamic motion in Figure 9) actually rotates as a function of time and in a different direction. Hence, the friction forces may enhance the energy of an unstable mode.

\[
\text{Fig. 7—Two eigenvalues of a pressure plate that could be coupled by varying the coefficient of friction } \mu. \quad \text{The index of dynamic stability } \sigma (\text{real part of the eigenvalues}) \text{ becomes negative for the stable wobbling mode (dashed line) and positive for the unstable wobbling mode (solid line).}
\]

\[
\text{Fig. 8—Components of the predicted eigenvalues for the wobbling modes of the pressure plate: (a) Index of dynamic stability } \sigma \text{ or real part of the eigenvalues for Plate G and Plate V, (b) natural frequency } \omega \text{ of the modes (in hertz), which exhibits two distinct values } \omega_a \text{ and } \omega_b \text{ for high pressure-plate stiffnesses } k_i \text{ and a single value } \omega_c \text{ for low pressure-plate stiffnesses.}
\]

\[
\text{Fig. 9—Visualization of the complex wobbling modes of the pressure plate for a coefficient of friction } \mu = 0.35. \quad \text{These modes occurred at the natural frequency } f_c \text{ of the clutch-disk cushion and illustrate the directions of the traveling waves induced by modes } \psi_i \text{ and } \psi_c \text{, moving with the axis of rotation indicated by the dynamic-motion line.} \quad \text{Mode } \psi_i \text{ is a stable mode associated with waves that move in the direction of plate rotation; mode } \psi_c \text{ is an unstable mode associated with waves that move in the direction of the friction forces.} \quad \text{The axes were normalized for this sketch.}
\]
conversely, they may dissipate the energy of a stable mode. For coefficients of friction less than 0.35, the axis of rotation remains stationary, showing the absence of any coupling between $\theta$ and $\theta'$. Stability diagrams are presented next to summarize the effects of pressure plate geometry (inner and outer radii $r_i$ and $r_o$, and thickness $2\ell$), pressure plate structural stiffness ($k_v$) and disk cushion characteristics ($\mu$ and $k$).

First, a diagram in terms of $r_i$ and $r_o$ is displayed in Figure 10. Defining $\delta = 0.5(r_o - r_i)$, the radii were varied from their baseline value minus $\delta$ to their baseline value plus $\delta$. The increasing degrees of instability (described by the real parts $\sigma$ of the unstable eigenvalues) are represented graphically and grouped from a low to a medium to a high degree of instability. Two straight lines indicate the baseline radii. It was found that larger radii would result in increased stability. However, outer radius $r_o$ seemed to have a greater influence than the inner radius $r_i$. For practical purposes, it would be more convenient to maintain the same contact area between the pressure plate and the disk. A smaller surface is not recommended since it would produce more heat and reduce the life of the clutch disk. Moreover, the heat capacity of the clutch is also affected by the pressure plate mass. Based on these considerations, it seemed that $r_o$ should be increased, since a 15% increase would theoretically result in a more stable system. In addition, this change would increase the torque capacity of the clutch.

Second, the stability diagram of Figure 11 shows that a thicker pressure plate would result in reduced stability. This is conceivable since plate thickness $2\ell$ controls the torque generated by the friction forces. This torque is responsible for the wobbling mode, because it couples angular displacement $\theta$ with angular displacement $\theta'$. Interestingly, a 10% decrease in plate thickness would make the system stable. As with the previous design change proposed for the outer plate radius $r_o$, one needs to consider the consequences of a thinner plate in terms of clutch life and performance. For instance, a change in plate thickness would affect the stiffness $k_v$, which is related to the bending mode of the pressure plate.

Figure 12 compares the influence of the coefficient of friction $\mu$ on the stiffness of Plates G and V. The pressure plate of the plate V material was stable for $\mu < 0.45$. For the plate made from Plate G material, decreasing $\mu$ by 10% (from 0.35 to 0.31) would make the system stable. Once again, this change may cause problems in terms of clutch performance because the transmitted torque would be also reduced. To compensate for this effect, the clamp load (normal load at full engagement) would have to be increased accordingly.

Finally, the stability diagram of Figure 13 was obtained in terms of pressure-plate stiffness $k_v$ and clutch-disk cushion stiffness $k_c$. The latter stiffness was varied up to its maximum value at full engagement. In parallel, the eigenvalues were computed via an iterative process to determine the corresponding values for $k_v$ and $k_c$ at each point on the plot. Figure 10 shows the loss of stability as the clutch is being engaged. Here it occurs around $k_c \approx 26$ MN/m for plate G, whereas the instability region is avoided with plate V at the same cushion stiffness. Such predictions match empirical observations [2] fairly well.
5 CONCLUSIONS

The analytical study reported in this paper yielded much insight into the mechanism(s) of “EEK” noise in dry friction clutches. Specifically, the friction-induced vibration of the pressure-plate disk-cushion system was correlated with a loss of dynamic stability. The analytical model includes a six degree-of-freedom lumped-parameter system that describes rigid-body wobbling modes and the first elastic deformation mode of the pressure plate. The complex eigenvalue method was utilized to study the system stability in terms of the coefficient of friction between the pressure plate and the cushion or lining of the clutch disk, pressure-plate geometry, and the structural stiffness of the pressure plate. The latter was found to be a key parameter. By controlling the frequency of the first elastic deformation mode, the coupling with the wobbling mode could be altered. This instability mechanism and its effect seemed to correlate well with empirical observations. Future work should focus on developing an enhanced analytical basis (including nonlinear models) to better understand the “EEK” phenomena.

6 ACKNOWLEDGMENTS

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7 REFERENCES