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DESIGN OPTIMIZATION OF AUTOMOTIVE LOCK-UP CLUTCHES WITH DAMPER SPRINGS USING SIMULATED ANNEALING, FEM, AND B-SPLINE CURVES

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ABSTRACT–An efficient optimum design process has been developed and applied to systematically design a lock-up clutch system for a torque converter used in an automatic transmission. A simulated annealing algorithm was applied to determine the parameters of the compressive helical damper springs in the clutch. The determination of the number, location, a number of turns, and deflection of damper springs plays an important role in reducing vibration and noise in the lock-up system. Next, FE-based shape optimization was coded to find the shape of the clutch disk that would satisfy the strength, noise and vibration requirements. Using the optimum code, parametric studies were performed to see how spring diameters and frequencies of clutch systems changed as the damper spring traveling angles and the torques were varied. Based on the optimum results, five different designs for clutches with different springs were fabricated and vibration analyses and tests were conducted to validate the accuracy of the proposed method. Results from the two methods show a good correlation.

KEY WORDS : Shape optimization, Lock-up clutch, Damper spring, FEM

1. INTRODUCTION

It is important to reduce the weight of an automobile in order to improve power transfer and fuel efficiency. One of the heaviest parts in an automobile is the engine and power transmission, which account for about 20–25% of the total weight. In an effort to reduce weight, design optimization was applied to a specific part of the automotive transmission (Kim *et al.*, 2006). The lock-up clutch in the torque converter of an automatic transmission connects or disconnects engine power to a propeller shaft using hydraulic pressure. At startup or at low speed in an automatic T/M car, a fluid coupling (*i.e.*, torque converter) is engaged. When the car speed increases the mechanical lock-up clutch is engaged to transmit engine power directly, without using the fluid coupling, in order to prevent power loss. At the moment of direct engagement, large torsional impact and vibration occur and the damper springs of the lock-up clutch must absorb such disturbances without excessive vibration and noise. Therefore, an optimum design for damper springs and clutch disk shapes is an important issue for minimizing noise and vibration and maximizing the strength. A genetic algorithm for spring optimization and an FEM-based algorithm for optimizing disk shape were utilized in this paper.

The first FEM-based shape optimization with isoparametric elements was suggested by Zienkiewicz and Campbell (1973). During shape optimization, the shape continues to change by repositioning boundary nodes. Therefore, this method requires a feature that will maintain a good quality finite element mesh because mesh quality affects the accuracy of finite element solutions. Until the 1980s, most researchers used all nodes along the boundary surface as design variables. Such a scheme, however, sometimes causes discontinuous or unstable boundary surfaces, which, in turn, may cause distorted meshes. To cope with this problem, Fleury and Braibant (1986) adopted the control points of B-spline curves as design variables instead of the surface nodes, which can smooth a design boundary surface in a continuous fashion. Bennett and Botkin (1985) suggested an automatic mesh generation method, coupled with an adaptive mesh refinement scheme.

Simulated annealing provides robust optimization for a global solution and is applied for higher order functions. An optimum solution for helical springs (Mancini and Piziali, 1976) was obtained using geometric programming. The research objective here is to develop an optimization procedure for a lock-up clutch system and to validate the

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designs by vibration tests. Simulated annealing code was developed to optimize the wire and coil diameters of the damper springs. For clutch disk shape optimization the finite element nodes on B-spline curves (Rogers and Adams, 1990; Anand, 1993) constituting the structural boundary were allowed to move in a proposed way using adaptive remeshing. This design procedure was validated by experiments.

2. MATHEMATICAL FORMULATION FOR OPTIMIZATION

2.1. Spring Optimization by S.A.

The determination of the location, number of turns, deflection, and number of damper springs plays an important role in reducing vibration and noise in a lock-up system. The presence of high powers of wire and coil diameters in the model, e.g., terms involving d^4, etc., results in difficulty when applying gradient methods and thus the simulated annealing method is especially attractive for spring design. Since damper springs can fail by fatigue or yielding, our objective is to maximize the smaller of the two safety factors for fatigue and yielding so that wire and coil diameters are calculated by satisfying alternating shear stresses, mean shear stresses, endurance limits, strength, and torsional natural frequencies, as follows,

$$\text{Maximize } \frac{1}{sf_1}=\frac{1}{\tau_a/S_e+\tau_m/S_u} \tag{1}$$

$$sf_2=\frac{S_y}{\tau_a+\tau_m} \tag{2}$$

$$\text{Subjected to } L+2d\leq\frac{2\pi r}{n}-l_r \tag{3}$$

$$D+d\leq D_{out} \tag{4}$$

where the design variables D and d are coil diameter and wire diameter, respectively. The variable sf_1 is a fatigue safety factor and sf_2 is a static safety factor. L is a free spring length, r is a radial distance from a center to a location of a coil center at the clutch, n is the number of springs, l_r is the length of spring holders, and D_{out} is the diameter of an outer spring.

Usually, the damper springs of a lock-up clutch (Figure 1) consist of inner and outer springs, i.e., dual springs, as in Figure 2. The strengths, S_e, S_u, and S_y are the endurance limit, the ultimate strength, and the yield strength, respectively, and $S_e=6790.9d^{-0.16}N^{-0.2136}$, $S_u=1721.4d^{-0.16}$, $S_y=933.6d^{-0.16}$ and N is an infinite fatigue life (10^6 cycle).

The free length L is determined by a spring rate k and an active turn N_a. as

$$N_a=Gd^4/8kD^3,\ L=N_ad(1+p) \tag{5}$$

where G is a shear modulus and p is a spring pitch. The

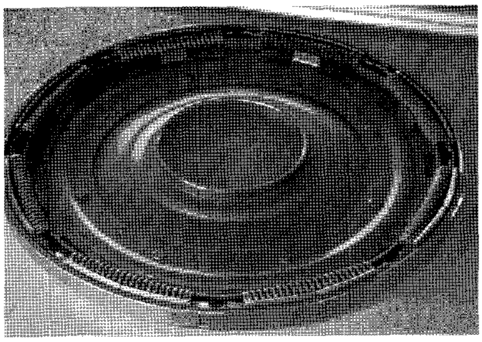


Figure 1. A lock-up clutch with eight damper springs.

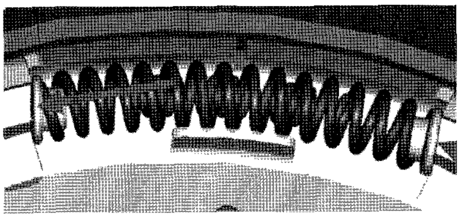


Figure 2. Inner and outer damper springs.

alternating stress τ_a and the mean stress τ_m are expressed as

$$\tau_a=8F_aDK/(\pi d^3),\ \tau_m=8F_mDK/(\pi d^3) \tag{6}$$

where F_a and F_m are an alternating force and a mean force and K is a Wahl factor such as

$$K=\frac{4(D/d)-1}{4(D/d)+4}+0.615\frac{d}{D} \tag{7}$$

The results of damper spring optimization are listed in Table 1. These results are the optimum designs that satisfy optimization equations (1)-(7), depending on the spring constants. The subscript, *out*, designates an outer spring and *in* an inner spring. For three different spring rates (k) in a six-springs case the optimization was performed under a constant 100200 N-mm torque and $\theta=18°$, 19°, and 20° in the clutch disk circumference. These

Table 1. Optimum results of damper springs.

No of springs	k (N/mm)	d_{out} (mm)	D_{out} (mm)	d_{in} (mm)	D_{in} (mm)	Max. $S.F.$
6	6	3.03	21.97	1.76	8.21	1.08
	5.68	3.00	22.00	1.64	11.58	1.06
	5.40	2.97	22.03	2.23	15.71	1.03
8	6.23	2.84	22.16	1.61	10.91	1.18
	5.60	2.79	22.21	2.10	15.23	1.23
	4.98	2.73	22.27	1.95	10.94	1.30
	4.36	2.66	22.34	1.97	9.85	1.38

Table 2. Variation of natural frequency by changing traveling angle (θ).

No. of springs	L_{max} (mm)	θ (deg)	δ (mm)	k (N/mm)	f_n (Hz)	f_n' (rpm)
6		18.00	32.57	6.00	27.86	1672
(T_1=	81.49	19.00	34.22	5.68	27.11	1627
100200)		20.00	35.87	5.40	26.44	1586

Table 3. Variation of natural frequency by changing torque (T_1).

No. of springs	L_{max} (mm)	T_1 (N-mm)	δ (mm)	k (N/mm)	f_n (Hz)	f_n' (rpm)
		100200	24.20	6.23	29.80	1788
8	57.36	90200	25.03	5.60	28.81	1729
(θ=13)		80200	25.93	4.98	28.18	1691
		70200	26.74	4.36	27.45	1647

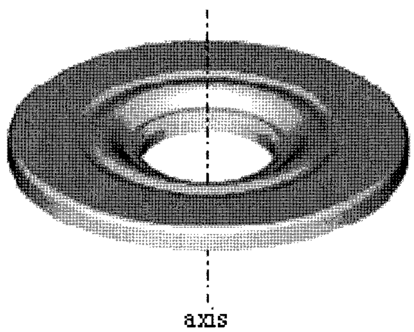
three spring rates were calculated based on the data shown in Table 2. These springs are designed for long-travel (*i.e.*, a large contracting angle, θ) to absorb a high torque during engagement at low engine rpm. This means that earlier (*i.e.*, lower rpm) direct engagement between the clutch and engine is possible and the rpm range for clutch engagement can be expanded.

The case of the eight-spring design is conventional and the four spring rates were calculated based on the data shown in Table 3. For these four different spring rates, optimum solutions were calculated with a constant θ=13°. It can be observed from Table 1 that the wire diameters for a six-spring case become larger, because each one of the six springs carries more torque than that of the eight spring case.

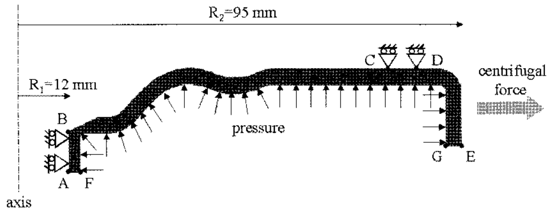
2.2. Shape Optimization of the Clutch Disk Cross-section

An axi-symmmetric shape optimization code was developed and an adaptive remeshing scheme was implemented, consisting of B-spline curves in order to represent the design boundary surfaces, and a mesh smoothing technique (Hyun and Lindgren, 1998, 2001) to improve interior mesh shapes (Hyun *et al.*, 2004). The volume of the clutch disk was taken as an objective function and the coordinates of the selected nodes on the boundary surfaces were taken as design variables. Maximum allowable stress and von Mises equivalent stress were set as constraints.

FEM-based shape optimization code was applied to obtain the shape of the clutch cross-section. Centrifugal force and hydraulic pressure were considered. The centrifugal force was calculated based on a rotational speed of 2000 rpm and a density of 7850 kg/m^3. The design stress under the combined load was limited to 50 MPa to 70

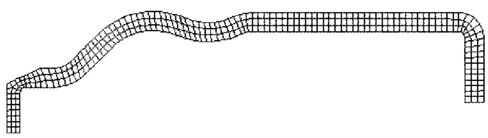


(a) Lock-up clutch piston

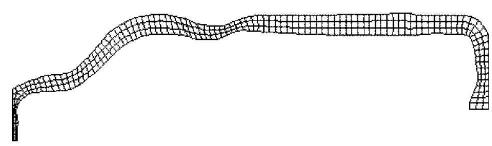


(b) Loading and boundary conditions

Figure 3. Loads and boundary conditions on a cross-section (P=0.784 MPa, E=209 GPa, ν=0.3, ρ=7850 kg/m, ω=2000 rpm).



(a) Before optimization



(b) After optimization

Figure 4. Finite element meshes before and after the shape optimization.

MPa, and the minimum thickness of the disk was limited to 0.6 mm. Figure 3 shows the 3-D shape of the clutch disk and cross-section with loads and boundary conditions. The FE meshes before and after optimization are shown in Figure 4. Most of the loads are supported by the disk area, and as a result, both vertical end parts are reduced in thickness due to small stresses in these areas. The curved region in the middle becomes thinner because centrifugal force tries to flatten the curve and, thus, relieve stress. The variable thickness clutch simply shows an example of numerical optimization. The clutch piston is normally manufactured by a press process so that the thickness is uniform.

2.3. Lock-up Clutch System

Lock-up clutch engagement at low engine rpm is desirable for fuel efficiency. However, it is achievable only if resonance is avoidable at the engagement rpm and NVH characteristics at that rpm are improved. Therefore, the 1st torsional natural frequency must be far lower than the engine rpm at engagement. Finite element frequency analysis was performed for a lock-up clutch system separated from the transmission. The disk and springs were modeled as spring elements and solid elements, respectively.

Several parametric studies, shown in Tables 2 and 3, have been performed using the optimum spring designs. Table 2 shows that the natural frequencies (f_n, f_n') of the lock-up clutches are reduced as the deflected angles (θ) (*i.e.*, traveled length (δ) to absorb impact by a sudden lock-up) increase. If a natural frequency at lock-up is harmonized with an engine lock-up rpm (2000 rpm in this size car), resonance takes place and NVH characteristics deteriorate. Theirfore, the results must be less than 2000 rpm so that earlier lock-up can be possible and it is better if frequencies are far from the specified 2000 rpm. The torque-carrying capacity of a spring reduces at a specified spring deflection (Table 3) and L_{max} is the max length of a single spring. This data shows that natural frequencies can be reduced as torque decreases.

3. EXPERIMENTS ON TORSIONAL VIBRATION

Following the design method developed, five different lock-up clutches were fabricated to measure their torsional frequencies. The natural frequencies of the entire powertrain system (combined engine, torque converter, and transmission) are usually available, but the isolated frequency data of the lockup clutch alone is rarely available. A design process that does not consider the clutch frequency itself is not efficient or accurate, and sometimes leads to unexpected noise and vibration.

In this paper, a new experimental method for testing natural frequencies of the lock-up clutch itself, independent of the powertrain system, has been devised. The experimental schematic is described in Figure 5. To match the test set-up with a real situation, a turbine and a driven plate that includes a torque converter system were assembled and connected to the lock-up clutch.

Two accelerometers were attached to the clutch to cancel out the effects of gravitational and the centrifugal force as shown in Figure 6. The accelerometers were located on the same line, i.e., 180° apart.

$$e_A=k_A(r\alpha-g\cos\theta+\ddot{u}\sin(\phi-\theta)) \tag{8}$$

$$e_B=k_B(r\alpha+g\cos\theta-\ddot{u}\sin(\phi-\theta)) \tag{9}$$

$$e_A+e_B=2r\alpha k \tag{10}$$

where e_A and e_B are signal intensity, α is angular acceleration, g is the gravitation, and $\ddot{u}$ is the linear acceleration of a center point. When the signal intensity of a linear accelerometer and an amplifier are adjusted to be $k=k_A=k_B$, only the tangential acceleration term is left,

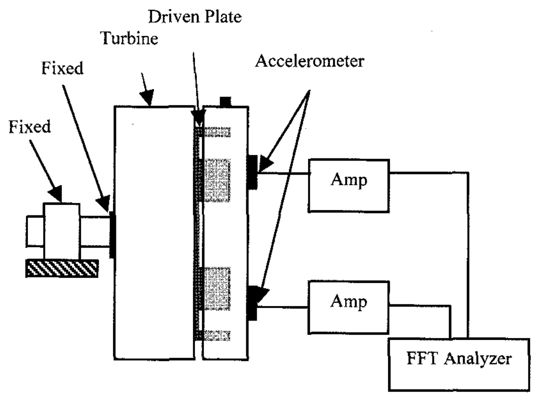
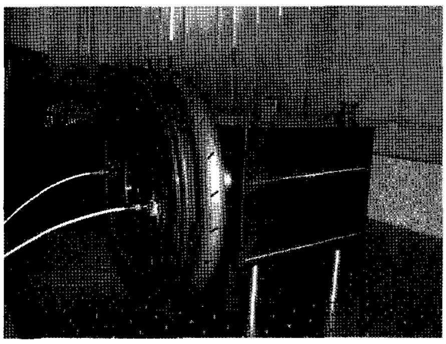
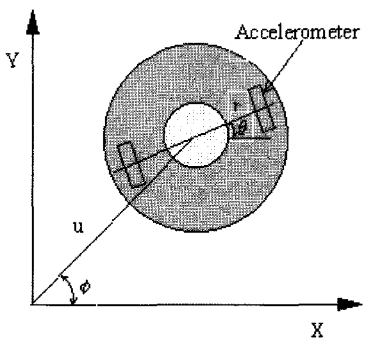


Figure 5. Experimental schematic diagram for a torsional vibration test.



(a) Experimental set-up



(b) Two accelerometers

Figure 6. Experimental set-up for a torsional vibration test.

Table 4. Torsional frequencies from analyses and tests for different springs and numbers.

No. of springs	Spring rate (N/mm)	Wire dia. (mm)	Coil dia. (mm)	Active Turns	FEM freq (Hz)	Tests freq (Hz)
4	7.04	2.3	11.2	27.0	22.8	21.0
	6.23	2.0	9.2	29.0	21.5	20.0
8	6.23	1.7	9.4	18.8	29.8	30.0
	6.23	2.0	12.2	15.7	29.8	28.5
	4.36	1.8	11.5	17.6	27.5	26.5

as in Equation (10). Using the system, the first torsional natural frequency of the lock-up clutch was detected and the results are summarized in Table 4. It can be observed that the measured values match well with finite element analysis results. Therefore, the present method is deemed efficient and accurate enough for predicting the frequency of the lock-up clutch system. It is quite natural that the clutches with longer springs had lower torsional natural frequencies, as noted in Table 4.

4. CONCLUSIONS

An efficient optimization method for a lock-up clutch system with dual damper springs was developed based on the simulated annealing and FEM. Torsional natural frequencies of the lock-up clutch were measured by a newly developed method and then calculated by analysis to demonstrate the accuracy of the developed design process. The shock absorbing characteristics of damper springs vary with the size of the lock-up clutch and the location and number of damper springs.

(1) As torque acting on the damper springs increases, the spring wire diameter becomes larger and the coil diameter decreases.
(2) The torsional natural frequencies of a long-travel damper spring are reduced markedly compared to those of a short-travel damper spring. This means a long-travel damper makes direct clutch engagement possible at low engine rpm.
(3) Optimum shapes of arbitrary complicated contours can be obtained in a relatively short computational time by the suggested method. The trial application for a clutch disk shows that the solution boundary was very smooth with no perturbations while weight savings of up to 13% were achieved.
(4) The design method suggested here can be reliably adopted to develop a new lock-up clutch system.

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