Analysis of a Clutch Damper Using a Discrete Model

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It is important to have a precise model for the clutch damper in order to simulate the entire powertrain of a vehicle and predict the responses of the system. In this research, we developed a new model in which the spring used in the clutch damper is divided into a finite number of elements. The model takes many unique properties of arc-shaped springs into consideration and is anticipated to be more precise than conventional simple models. With the model, two meaningful results were presented which can be utilized afterwards. One is a simulation concerning the peak torque transmitted via the clutch damper. The other is a simulation that shows the hysteretic characteristics of the clutch damper.

Key Words : Vehicle Powertrain, Clutch Damper, Multi-D.O.F. Discrete Model, Transient Torque, Hysteresis

1. Introduction

The springs placed in the clutch damper are arc-shaped. Thus, power can be delivered through the springs in both radial and tangential directions. There exists friction between the springs and the guide of the clutch, for there are as many contact points as the number of coil turns. The friction can be an additional route of power transmission and the stiffness effect of the spring and the rotational speed of the clutch can be coupled due to this friction. That is, as the engine speed increases, the centrifugal force and the frictional force increase as well. Therefore, the stiffness effect of the total system changes with the engine speed.

The friction causes the force at one contact point to differ from the forces at the other points resulting in a different compression at each point. For example, the terminal coil turn may not be compressed at all or become inactivated.

Despite all these complex and nonlinear properties of the arc-shaped spring, there haven't been many attempts to take all of them into consideration.

When modeling the clutch in a powertrain model, conventional studies (Ohnuma, 1985 ; Keeney, 1992 ; Virkler, 1986) considered it as a combination of stiffness, viscous damping and structural damping. However, it is not appropriate to use those conventional methods when we need to achieve a higher accuracy.

In this paper, we developed a model which shows all the properties of the clutch damper mentioned above. To replace the conventional structural equation, $T_{td}=f(\theta_t-\theta_e, \dot{\theta}_t-\dot{\theta}_e)$, additional parameters were taken into account. Stiffness of the spring and dynamic equations with centrifugal force and frictional force were considered altogether to explore the dynamic responses of the entire powertrain system.

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2. Discrete Models Designed

2.1 Roles of a clutch damper

The clutch damper refers to a set of dampening springs placed in the clutch. Some of the clutch dampers developed today use only linear springs as dampening springs while other products use additional arc-shaped springs to get the additional effects. Diverse forms of clutch dampers have been developed and their shapes and functions depend on the know-how of each manufacturer. However, every type of clutch dampers aims at some common purposes. By placing a relatively soft element among the powertrain, followings can be achieved.

First, the damper is supposed to protect other components of the powertrain by absorbing the intense impacts from the engine or the driving wheels. If all the inertias are coupled only by hard stiffnesses such as simple shafts, those impacts would be delivered without any time delay and dissipation, and there might be huge problems with the stability of the system.

Second, the clutch damper can act as a mechanical low-pass filter. Since the reciprocating motion of the engine is converted into the rotational motion, torsional irregularities can occur and cause vibrations of very high frequency. They have bad influences on ride comfort, drivability and the durability of the components. These vibrations also induce unpleasant noises such as gear rattles, body boom. With the clutch damper, these vibrations of high frequencies can be filtered.

Third, the clutch damper can play a tuning role in adjusting the natural frequencies of the system. Every stiffness and inertia constituting the vehicle powertrain is designed for its best performance and has a limited range of structural modification. Furthermore, each component is individually designed from a respective production line and it is not easy to consider the problem which arises when the components are assembled altogether. That is, the natural frequencies of the system are determined only after all parts are assembled and before that, the exact values of the natural frequencies are hardly predictable. This problem can be solved using the wide flexibility of the clutch damper design. It is because there are not many constraints in designing the damper and it is possible to design it considering other components and the desired natural frequencies. This will guarantee the good performances in vibration isolation.

2.2 Structure of a clutch damper

There are several types of clutch dampers depending on the manufacturer. Each takes a unique figure in order to perform the intended properties. In this paper, a clutch damper with linear springs placed in the inner region of the plate and arc-shaped springs in the outer region is dealt with. The springs are called "Inner dampers" and "Outer dampers" respectively.

The outer damper is relatively long and connects both clutch plates regardless of the relative displacement between the two plates. It contacts the side guide of the clutch and the lubricant is filled between them.

The inner damper is inactive until the relative displacement of the two plates reaches a certain degree. After the angle passing beyond it, the inner damper gets activated. Then, it acts as additional stiffness and damping. Therefore, the characteristics of the clutch damper take a dual shape with a low stiffness region and a high stiffness region.

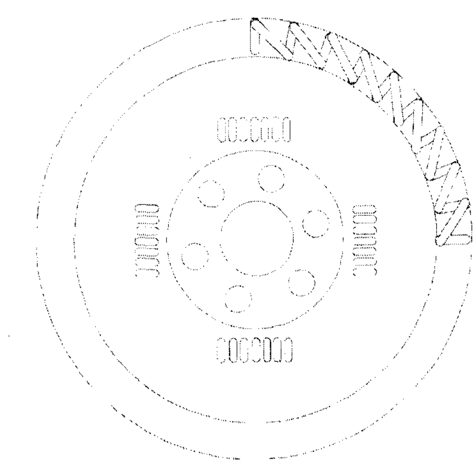

Fig. 1 Inner and outer dampers of the clutch damper

2.3 Discrete model of the arc-shaped spring

To analyze the arc-shaped spring, it is modeled to be a multi-d.o.f. discrete model with the d.o.f. of the number of the coil turns.

Figure 2. shows one coil turn of the spring and an element of the model.

As shown in the figure above, one coil turn is regarded as one element of the model and each element has specific stiffness, k and specific mass, m. It contacts the guide at either upper or lower contact points.

For the spring with J coil turns, J elements should be connected in series and the left terminal coil turn is fixed to a turbine and the right one to a pump.

The turbine and the pump are the terms for the vehicles with automatic transmissions. They can be replaced with two sides of the clutch plates in the case of vehicles with manual transmissions, and the primary and secondary sides in the case of vehicles with dual mass flywheels.

The configuration of the pump, turbine, elements and the guide of the clutch are shown in the figure below.

The structure above is depicted to be linear for the convenience but the actual shape is curved along the circumference. For this model, the governing equations containing the displacements, velocities and accelerations of J elements are constructed and to be analyzed in the following sections.

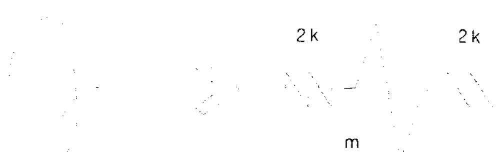

Fig. 2 An element of the arc-shaped spring.

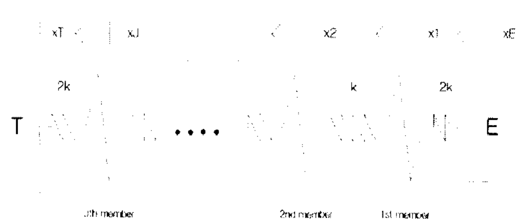

Fig. 3 Configuration of the spring composite

3. Dynamic Equations Formulated

3.1 Dynamic equations for a unit element

The unit element shown below is considered.

The displacement of each element, x_i is set to be a relative displacement from the undeformed state. Therefore, the absolute displacement, velocity and acceleration can be expressed as Eq. (1).

$$\begin{aligned} (x_i)_{abs} &= x_i + R\theta_t - R\beta\left(J + \frac{1}{2} - i\right) \\ (\dot{x}_i)_{abs} &= \dot{x}_i + R\omega_t \\ (\ddot{x}_i)_{abs} &= \ddot{x}_i + R\alpha_t \end{aligned} \tag{1}$$

where,

x : linear displacement of an element, m
R : radius of arcspring, m
θ : angular displacement, rad
ω : angular velocity, rad/s
α : angular acceleration, rad/s^2
β : deformed angle of an element, rad
γ : deformed angle of an element, rad
i : element number, subscript

With the equations, following dynamic equations can be developed.

Tangential direction :

$$\begin{aligned} m\ddot{x}_i &= k(x_{i-1} - x_i)\cos\frac{\gamma_i}{2} \\ &\quad + k(x_{i+1} - x_i)\cos\frac{\gamma_i}{2} + f_i \end{aligned} \tag{2}$$

Radial direction :

$$\begin{aligned} 0 &= N_i + k(x_{i-1} - x_i)\sin\frac{\gamma_i}{2} \\ &\quad - k(x_{i+1} - x_i)\sin\frac{\gamma_i}{2} + \frac{m(R\omega_i + \dot{x}_i)^2}{R} \end{aligned} \tag{3}$$

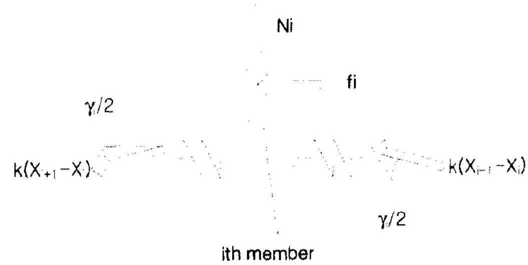

Fig. 4 Free body diagram of an element

where,

m : mass of an element, kg
k : stiffness of an element, N/m
f : frictional force, N
N : normal force, N

The deformed angle per unit element is obtained by Eq. (4).

$$
\begin{aligned}
\gamma_i&=\beta+\left(\frac{x_i+x_{i+1}}{2}\;\frac{x_{i-1}+x_i}{2}\right)/R\\
&=\beta+\frac{x_{i+1}-x_{i-1}}{2R}\ \text{for } i=2,3,\cdots,j-1\\
\gamma_1&=\beta+\left(\frac{x_1+x_2}{2}-R(\theta_e-\theta_t)\right)/R\\
&=\beta+\frac{x_1+x_2}{2R}-\theta_e+\theta_t\\
\gamma_J&=\beta+\left(0\,\frac{x_{J-1}+x_J}{2}\right)/R=\beta\,\frac{x_{J-1}+x_J}{2R}
\end{aligned}
\tag{4}
$$

where,

e, p : engine, subscript
t : turbine, subscript

3.2 The extended Dahl's friction model

The frictional force in the model has a great importance. Energy dissipation due to the frictional force results in damping effects. The normal force acting on the guide of the clutch plate is also due to the frictional force. It plays a significant role in the dynamic equations making the engine speed to be an important factor which characterizes the dynamic responses of the powertrain.

In this paper, the extended Dahl's friction model is used for representing the frictional force in the system. The model was suggested by Dahl and is considered to make up the week points of the conventional slip/stick friction model.

This model introduces a new state variable, z, and the variable is used to calculate the frictional forces when mixed with other variables in dynamic equations. The model also introduces a function, g(v), to represent the frictional force in a steady state.

The clutch damper discussed in this paper is a system where much attention should be paid on its responses in the transient state and most of its motions are sliding motions. Therefore, the use of this model is considered to be quite appropriate in describing the motions and the responses more precisely.

The equations related to the frictional forces and the friction model introduced above are written as Eq. (5).

$$
\begin{aligned}
&f_i=N_i(X_s z_i+K_d\dot{z}_i)\\
&\dot{z}_i=v_i\left(1-\operatorname{sign}(v_i)\frac{z_i}{g(v_i)}\right)\\
&g(v_i)=\frac{1}{K_s}\left\{(\mu_s-\mu_k)\exp\left(-\frac{|v_i|}{V_e}\right)+\mu_k\right\}
\end{aligned}
\tag{5}
$$

where,

f : frictional force, N
N : normal force, N
g : frictional force in a steady state, N
K_s : friction stiffness coefficient
K_d : friction damping coefficient
z : equivalent friction state variable
v : relative velocity, m/s
μ_s : stationary friction coefficient
μ_k : kinetic friction coefficient
V_c : velocity constant, m/s

4. Powertrain Model Analyzed

4.1 Formulation of dynamic equations

To model the powertrain of the vehicle, a 6-d.o.f. model like the figure below was developed.

Following equations can be obtained with the model.

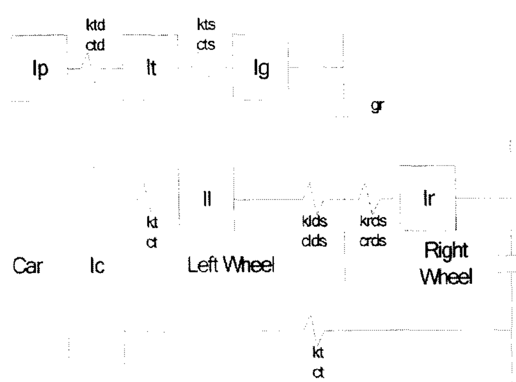

Fig. 5 6-D.O.F. vehicle powertrain model

$$
\begin{aligned}
&I_p\ddot{\theta}_p = T_e - (T_{td})_{pump} \\
&I_t\ddot{\theta}_t = (T_{td})_{turbine} - T_{ts} \\
&I_g\ddot{\theta}_g = T_{ts} - T_{lds}/gr - T_{rds}/gr \\
&I_l\ddot{\theta}_l = T_{tl} - T_{lds} \\
&I_r\ddot{\theta}_r = T_{tr} - T_{rds} \\
&I_c\ddot{\theta}_c = T_r - T_{tl} - T_{tr}
\end{aligned} \tag{6}
$$

$$
\begin{aligned}
T_{td} &= (T_{td})_{pump} = (T_{td})_{turbine} \\
&= k_{td}(\theta_p - \theta_t) + c_{td}(\dot{\theta}_p - \dot{\theta}_t) \\
T_{ts} &= k_{ts}(\theta_t - \theta_g) + c_{ts}(\dot{\theta}_t - \dot{\theta}_g) \\
T_{lds} &= k_{lds}(\theta_g/gr + \theta_l) + c_{lds}(\dot{\theta}_g/gr + \dot{\theta}_l) \\
T_{rds} &= k_{rds}(\theta_g/gr + \theta_r) + c_{rds}(\dot{\theta}_g/gr + \dot{\theta}_r) \\
T_{tl} &= k_t(\theta_c - \theta_l) + c_t(\dot{\theta}_c - \dot{\theta}_l) \\
T_{tr} &= k_t(\theta_c - \theta_r) + c_t(\theta_c - \theta_r)
\end{aligned} \tag{7}
$$

where,

td : clutch damper, subscript
ts : turbine shaft, subscript
lds, rds : left/right drive shaft, subscript
tl, tr : left/right tyre, subscript
g : gear box, subscript
c : vehicle sprung mass, subscript
I : mass moment of inertia, kgm^2
c : damping coefficient of coupling, Ns/m
gr : gear ratio
T_f : turbine driving torque via frictional force, Nm
T_e : engine torque, Nm
T_r : resistance torque, Nm

In the equations above, it is assumed that all the stiffnesses between the inertias including the clutch damper are linear and all dampings are viscous. However, the stiffness and damping of the clutch damper can be replaced with elements realized by following equations for further exploration. These equations are made using the discrete model developed in the former sections.

$$
\begin{aligned}
&(T_{td})_{pump} = R \times 2k\{R(\theta_p - \theta_t) - x_1\} \\
&(T_{td})_{turbine} = R \times 2kx_J + \sum T_f
\end{aligned} \tag{8}
$$

4.2 Overshoot simulation

When the maximum engine torque is delivered to the powertrain abruptly, a transient state occurs. In designing the stiffness of the clutch damper, the overshoot due to the transient state of the power transmission is important in the aspect of the deterioration in ride comfort and the protection of the components.

The clutch damper needs to be designed to deliver the maximum torque within the limited range of angular displacement and to have a stiffness which is as low as possible at the same time. Therefore, it is important in the design to have the information of the peak torque transmitted via the clutch damper. With the information of the peak torque, the maximum torque as a design parameter can be lowered and the stiffness of the damper can also be lowered simultaneously. For this purpose the following simulation is carried out.

The input torque of the engine is set to be a step signal. Initially the vehicle runs at the speed of 40 km/hr with the 3rd gear ratio (6.654 ; considering the final drive gear) and there is no change of gears. To simplify the system, 0 Nm of engine torque is assumed before the vehicle starts. In 0.05 seconds the input engine torque of 100 Nm is applied.

By solving the governing equations developed above with respect to time, plots shown in Fig. 6 can be obtained.

The graphs show the angular velocities of the inertia — engine/pump, turbine, gear box, left wheel, right wheel and vehicle unsprung mass respectively.

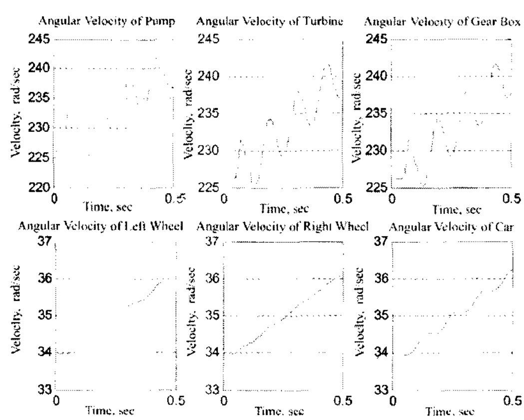

Fig. 6 Angular velocities of the equivalent inertias

The torque transmitted via the clutch damper and the displacements of each arc-shaped spring elements are shown in Figs. 7 and 8.

As shown in Fig. 6, there are differences in phase and it is considered to be due to the relatively soft stiffness of the clutch damper. It can be also found that the transient torque is absorbed in the response of the vehicle unsprung mass.

In Fig. 7 the maximum torque of 145 Nm can be expected to be delivered when the maximum engine torque is applied. It is due to the overshoot phenomenon occurring when the impact torque is applied. Therefore, the clutch damper needs to be designed to have a maximum torque capacity of over 145 Nm and have enough stiffness accordingly.

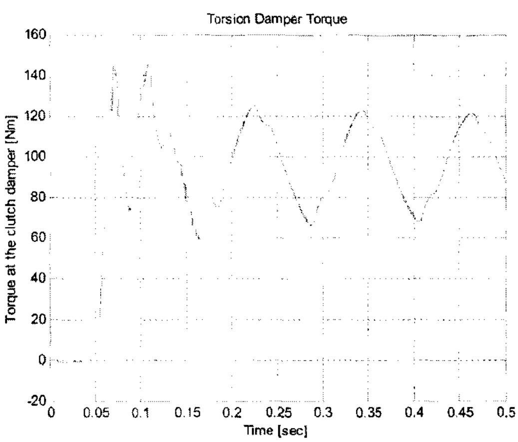

Fig. 7 Transmitted torque via the clutch damper

Figure 8 shows the displacements of the elements. 10 elements were used in the simulation. As can be expected, we verified that the displacements are not linear because of the friction between the springs and the guide. The terminal coil turn is obtuse to the input, which brings partial inactiveness of the spring.

4.3 Hysteresis plot

The energy dissipation due to the friction results in the nonlinearity of the clutch damper. For this reason, every local point of the damper exhibits different stiffness.

To display the characteristics mentioned above, the relative angular displacement and the corresponding torque were plotted together and the hysteresis loop could be obtained.

The locus has a varying thickness along the wind-up angle. The thickness varies due to the friction and other circumstances.

Initially the spring moves from the undeformed position with harder stiffness than the stiffness which was originally designed. The locus is formed above the straight line from the origin with a certain distance. When the wind-up angle decreases, the stiffness becomes larger and the locus shows a steep downturn. Thus, the locus with decreasing wind-up angles is formed below the straight line passing the origin.

If the area of the locus can be calculated, the dissipated energy can be found and it can be used

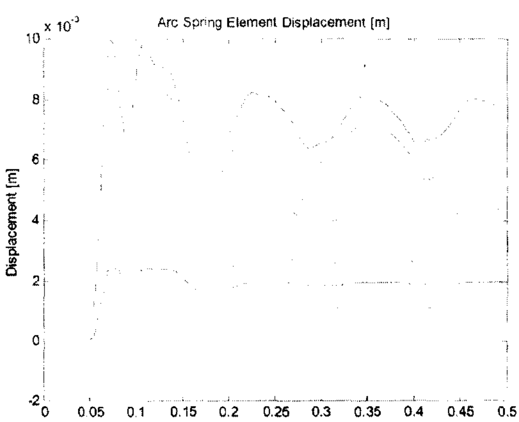

Fig. 8 Displacements of each element

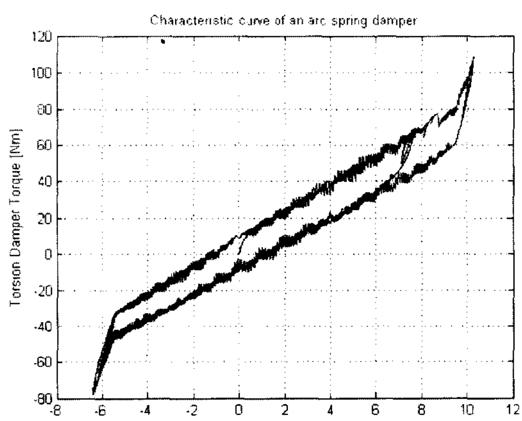

Fig. 9 Clutch damper hysteresis locus

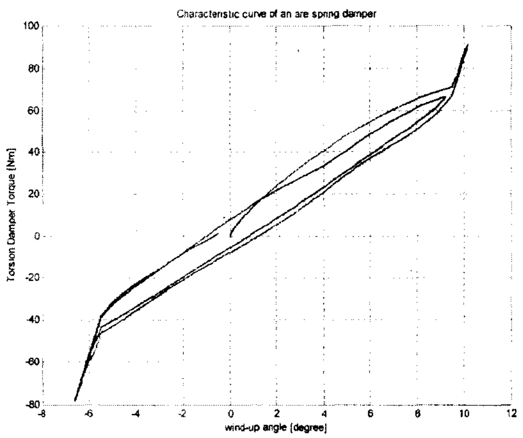

Fig. 10 Clutch damper hysteresis locus with a conventional model

to find the equivalent coefficients of the system. It should be noted that the thickness is not constant for all cases but varies with the operating circumstances. When the engine speed is high, the frictional forces are high as well and therefore the thickness goes thicker. On the other hand, it gets thinner when the engine speed is low and the frictional forces are low.

Figure 10 represents the hysteresis locus from the simulation with a conventional model. The locus takes an ellipsoidal shape while the locus in the Fig. 9 shows a parallelogramatic envelope. The parallelogramatic shape is closer to the curves from other experimental researches. (Albers, 1994 ; Ohnuma, 1985) By comparing the latter with the first, we could verify that the newly-developed model is better for predicting the fine movements of the system in detail. Also, we could arrive at the conclusion that we can obtain the higher accuracy by using the improved model.

5. Conclusion

In this paper, many factors were considered in order to develop a model for the clutch damper with which more precise simulations on the responses of the powertrain system can be carried out. The factors include frictional forces, centrifugal forces and the force transmission in radial and tangential directions, which characterize the various properties of the clutch damper. With this consideration, a new model was set up.

In the model, each coil turn of the arc-shaped spring was regarded as a unit element of the discrete model. Frictional, centrifugal forces acting on the elements were calculated with the discrete model in the form of ordinary differential equations, and the extended Dahl's friction model was introduced for the frictional forces.

With this model and simulations, the maximum torque capacity and the hysteresis loop of a clutch damper could be obtained. The results from the simulations showed that the model has the non-linear properties which can not be obtained with the conventional models.

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