Occurrence Mechanism and Suppression Method for Shudder in Automatic Transmission Powertrain

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ABSTRACT: Shudder occasionally occurs in the lock-up clutch of automatic transmissions when the frictional characteristics of the lock-up clutch have a negative slope with respect to the relative slip velocity. This study evaluates the effect of a dynamic absorber on reducing shudder in an automatic transmission powertrain. The automatic transmission is modeled as a linear multi-degree-of-freedom system using Lagrange’s equation of motion. The optimal dynamic absorber characteristics for suppression of shudder are calculated. The experimental results reveal the optimized dynamic absorber can completely suppress shudder. Engine forced vibration analysis shows that the dynamic absorber can reduce vibration without any detrimental effects.

KEY WORDS: vibration, powertrain, test and analysis technology, self-excited vibration, shudder, dynamic absorber [B3]

1. Introduction
In recent years, automatic transmissions have become widely used, and the number of cars that have automatic transmission systems continues to increase. However, automatic transmissions suffer from poorer fuel economy than manual transmissions. To overcome this problem, a lock-up clutch is used in the torque converter of automatic transmissions in order to achieve lower fuel consumption. Figure 1 shows a cross-sectional schematic of a torque converter containing a lock-up clutch. The lock-up clutch comprises a piston with a frictional material and dampers, the latter being a set of springs that support the piston. The engine directly rotates the pump impeller (input shaft), which causes oil to flow in the torque converter. The turbine runner (output shaft) rotates because of the resulting oil pressure. When the rotation speed of the turbine runner approaches that of the pump impeller, the piston with the frictional material is driven against the torque converter cover by the oil pressure, causing it to lock. The rotation speed of the turbine runner then becomes the same as that of the pump impeller. This process effectively transfers torque from the input shaft to the output shaft.

However, slip condition occurs during the lock-up process, and can lead to a frictional vibration referred to as shudder. Shudder is a type of self-excited frictional and torsional vibration in the powertrain, which can also be transmitted to the tires and car seats, thus leading to discomfort during driving.

Although several researchers have investigated shudder (1)-(7), the underlying mechanism has not yet been fully clarified. In order to suppress shudder, an improved automatic transmission fluid (ATF) (8)(9) and frictional material (10) have been developed. However, no research has so far been reported into methods of eliminating shudder by the use of an optimized dynamic absorber, nor has such an dynamic absorber been evaluated under engine forced vibration conditions.

In the present study, experiments were carried out using an actual vehicle to clarify the characteristics of shudder, with a test rig being set up to investigate the frictional characteristics. The results were compared to those of a numerical analysis based on
Lagrange’s equation of motion, in which the automatic transmission was modeled as a linear multi-degree-of-freedom system, as in an actual vehicle. In the analysis, a dynamic absorber was attached to the lock-up clutch, and its parameters were optimized to suppress shudder. Then, the selected optimum dynamic absorber performance is evaluated in forced vibration condition to make sure this dynamic absorber did not give any detrimental effect for engine forced vibration condition.

2. Characteristics of Shudder

A traveling test using an actual vehicle was carried out in order to investigate the occurrence of shudder. The characteristics of shudder can be summarized as follows:

(a) Figure 2 shows a frequency analysis of the variation in the propeller shaft rotation speed when shudder occurred. It can be seen that the vibration frequency associated with shudder is 32.7 Hz. This is found to be independent of the engine forced vibration frequency due to ignition of engine and the frequency is approximately constant even when the gear speed is changed. In addition, vibration with frequency of approximately 10 Hz is occasionally occurs.

(b) Shudder occurs when the difference in rotation speed between the pump and turbine runner is approximately 60 to 100 rpm, just after the lock-up process begins.

(c) Shudder is a torsional vibration that occurs throughout the drivetrain. The phase difference between the input, the output and the propeller shaft is in phase.

(d) As shown in the waveform in Fig. 3, the amplitude of the vibration of shudder increases exponentially during the early stages of shudder growth. In order to examine the growth rate of shudder, the envelope curve fitting of the waveform by the exponential function at that stage is applied, and the value that corresponds to the real part of the eigenvalue is calculated. The value is approximately 4.0 s⁻¹.

A hammering test was performed on a piston supported by springs (dampers) in the rotational direction under constant torque, and its natural frequency was found to be about the same as the shudder frequency. This result implies that the piston is a primary source of the shudder.

The friction characteristics of the torque converter were next experimentally investigated. The frictional torque was measured while changing the slip velocity of the torque converter. Figure 4 shows the dependence of the coefficient of friction on the relative slip velocity. The region between the two dash-dotted lines represents the velocity range within which shudder occurred during the beginning of the lock-up operation. The experiments were performed using four different oil pressures. It can be seen that the coefficient of friction has a negative slope with respect to the relative slip velocity. In such a situation, the system has a negative resistance, which leads to the creation of negative damping that may generate shudder.

3. Modeling and Theoretical Analysis

Mathematical modeling was carried out to investigate shudder and the effect of forced engine vibration based on an actual automatic transmission powertrain. Since in the present study, shudder was found to occur in the fifth gear of a six-speed transmission, only the fifth gear was considered in the modeling process. The mathematical modelling to investigate shudder can be analyzed by a lower DOF to handle easier or to have the visibility in the early design stage of development. However, the modelling of automatic transmission powertrain in this paper is design as 13 DOF which is highly accurate as actual vehicle to obtain reliable result.

Figure 5 shows the analytical model used for the gear train in the fifth gear. There are three planetary gears in the gear train used in this research, referred to as the front (Fr), middle (Mid), and rear (Rr) planetary gears. Each planetary gear consists of a sun gear, a carrier, and a ring gear. When the rotation speed of two of these components are determined, the rotation speed of the other component is set. Considering the effect of this mechanism of the planetary gears on the natural frequencies and natural modes, the planetary gears were modeled exactly as actual structure in this theoretical analysis. The shafts in the gear train are modeled as rotational springs.

In Fig. 5, \( J_1, J_2, \ldots, J_m, J_n \) are the moments of inertia of the corresponding components, \( k_1, k_2, \ldots, k_n, k_m \) are the rotational spring coefficients, \( \theta_1, \theta_2, \ldots, \theta_n, \theta_m \) are the angular displacements.
for each element, $n_1,\ldots,n_6$ are values calculated from the planetary gear ratio, and $p,q,r$ are constants determined by the gear ratio between components. In this paper, the damping coefficients for the rotational springs are denoted by $c_{i1},c_{i2},\ldots,c_{i1},c_{i2}$.

The equation of motion is derived based on the torque converter characteristics and friction characteristics. The torque converter performance is expressed as:

$$e = \frac{N_p}{N_t},$$

$$t_c = \frac{T_c}{T_c},$$

where $e$ is the ratio of the turbine runner and pump impeller speeds, $N_p$ is the rotation speed of the turbine runner, $N_t$ is the rotation speed of the pump impeller, $t_c$ is the ratio of the turbine runner and pump impeller torques, $T_c$ is the torque acting on the turbine runner and $T_p$ is the torque from the pump impeller.

The capacity coefficient, $C$, for the torque converter is defined as:

$$C = \frac{T_c}{N_t^2},$$

Next, $T_c$ and $T_p$ are approximated using the following quantic equations from experimental data:

$$T_p = CN_p^2 = \sum_{n=0}^{N_p} b_n (\frac{N_p}{N_t})^n$$

$$T_c = t_c N_t = \sum_{n=0}^{N_p} d_n (\frac{N_t}{N_p})^n$$

where $C$ and $t_c$ are determined experimentally and $b_n$ and $d_n$ are constants derived from curve fitting of experimental data.

Then, $N_p$ and $N_t$ are described by the following equations:

$$N_p = N_{p0} + \Delta N_p,$$

$$N_t = N_{t0} + \Delta N_t,$$

where $N_{p0}$ is the rotation speed of the pump for static torque, $\Delta N_p$ is the rotation speed of the pump for dynamic torque, $N_{t0}$ is the rotation speed of the turbine for static torque, and $\Delta N_t$ is the rotation speed of the turbine for dynamic torque.

From the above equations, $t_c$ and $C$ are related to $N_p$ and $N_t$. Thus, $T_c$ and $T_p$ are also related to $N_p$ and $N_t$. From these relations, the following equations are derived:

$$T_p(N_p, N_t) = T_p(N_{p0}, N_{t0}) + a_1\Delta N_p + a_2\Delta N_t,$$

$$T_c(N_p, N_t) = T_c(N_{p0}, N_{t0}) + a_1\Delta N_p + a_2\Delta N_t,$$

where $T_p(N_{p0}, N_{t0})$ and $a_1\Delta N_p + a_2\Delta N_t$ are the static and dynamic torque components, respectively, generated by the pump impeller. Similarly, $T_c(N_{p0}, N_{t0})$ and $a_1\Delta N_p + a_2\Delta N_t$ are the static and dynamic torque components, respectively, acting on the turbine runner. The values $a_1, a_2, \sigma_1$, and $\sigma_2$ are experimentally determined from the torque transfer characteristics of an actual torque converter. These parameters are approximated using the following equations based on the experimental data obtained using an actual vehicle:

$$a_1 = \frac{\partial T_p}{\partial N_p} \bigg|_{N_t=\text{const}} = \sum_{n=0}^{N_p} (2-m)b_n (\frac{N_p}{N_t})^n (\frac{N_p}{N_t})^\alpha,$$

$$a_2 = \frac{\partial T_p}{\partial N_t} \bigg|_{N_p=\text{const}} = \sum_{n=0}^{N_p} mb_n (\frac{N_p}{N_t})^n (\frac{N_p}{N_t})^\alpha,$$

$$\sigma_1 = \frac{\partial T_c}{\partial N_p} \bigg|_{N_t=\text{const}} = \sum_{n=0}^{N_p} (2-m)d_n (\frac{N_t}{N_p})^n (\frac{N_t}{N_p})^\alpha,$$

$$\sigma_2 = \frac{\partial T_c}{\partial N_t} \bigg|_{N_p=\text{const}} = \sum_{n=0}^{N_p} md_n (\frac{N_t}{N_p})^n (\frac{N_t}{N_p})^\alpha.$$

Next, the friction characteristics obtained from the actual vehicle experiment described in Section 2 are considered in
developing equation of motion. A friction force $\mu F R$ is acted on the piston, where $\mu$ is the friction coefficient, $F$ is the nominal force acting on the frictional surface, and $R$ is the effective radius for friction. Based on the experimental results, the friction coefficient is approximated by a linear function with a negative slope with respect to the relative slip velocity:

$$\mu = \mu_0 - \mu_1 (V - R\dot{\theta})$$  \hspace{1cm} (11)

where $V$ is the fixed circumferential velocity of the converter cover attached to the frictional material, $\mu_0$ is the slope of the friction characteristic curve, and $\mu_1$ is its y-intercept.

Equations of motion are derived for a 12-degree-of-freedom system with no dynamic absorber present, using a Lagrange’s equation that considers the characteristics of the torque converter and the effects of friction. The equation is written as:

$$\mathbf{M}\ddot{\mathbf{X}} + \mathbf{C}\dot{\mathbf{X}} + \mathbf{K}\mathbf{X} = \mathbf{T} \cos \omega \dot{\theta}$$  \hspace{1cm} (12)

where $\mathbf{X} = [\theta_1, \theta_2, \ldots, \theta_{11}, \dot{\theta}_{11}]^T$ is the displacement matrix, $\mathbf{T} = [T_0, 0, 0, 0, 0]^T$ is the external torque matrix, $T_0$ is the amplitude of the static external torque, $\omega$ is the angular frequency of the external torque, $\dot{\theta}$ is the time, $\mathbf{M}$ is the mass matrix, $\mathbf{C}$ is the damping matrix and $\mathbf{K}$ is the stiffness matrix.

The non-zero elements of the mass matrix, $M_{ij}(i, j=1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12)$, are given by:

$$M_{11} = J_2, M_{22} = J_1, M_{33} = J_2, M_{44} = J_1,$$

$$M_{55} = J_4 + n_1 J_4 + n_1 J_5, M_{66} = J_5 + n_1 J_4,$$

$$M_{77} = J_6 + n_1 J_7 + n_1 J_8, M_{88} = P_1 J_6 + n_1 J_7, M_{99} = p_1 J_7 + n_1 J_8, M_{10, 10} = q_1 J_7 + r_1 J_8, M_{11, 11} = r_1 J_8.$$

The non-zero elements of the stiffness matrix, $K_{ij}(i, j=1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12)$, are given by:

$$K_{22} = k_1, K_{33} = k_2 + k_3, K_{44} = k_2 + k_3 + k_4,$$

$$K_{55} = k_2 + n_1^2 (k_2 + k_3), K_{66} = k_2 + k_4 + n_1^2 (k_2 + k_3), K_{77} = k_4 + n_2^2 k_4 + n_1^2 k_4 + n_1^2 k_5, K_{88} = p_1 k_4,$$

$$K_{99} = q_1^2 k_4 + 4 r_1^2 k_4, K_{11, 11} = r_1^2 (k_4 + k_5).$$

$$K_{12, 12} = r_1^2 (k_4 + k_5), K_{23} = K_{32} = -k_3,$$

$$K_{34} = K_{43} = -k_3, K_{45} = -k_3, K_{46} = K_{56} = -k_4,$$

$$K_{55} = n_1 n_2 k_1, K_{56} = K_{57} = n_2 p_1 k_2, K_{58} = K_{68} = -n_2 p_1 k_2, K_{59} = K_{69} = -n_2 p_1 k_2,$$

$$K_{69} = K_{79} = -n_2 k_5, K_{79} = K_{89} = -n_2 k_5, K_{99} = -2 r_1^2 k_5.$$

The non-zero elements of the damping matrix, $C_{ij}(i, j=1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12)$, are given by:

$$C_{11} = a_1 - \mu R F, C_{22} = -\mu F R^2,$$

$$C_{33} = a_2 + c_1, C_{44} = a_2 + c_1 + c_2,$$

$$C_{45} = a_2 + n_1^2 (c_2 + c_3), C_{56} = c_1 + c_2 + n_1^2 (c_3 + c_4), C_{77} = c_1 + n_2^2 c_1 + n_1 n_2 c_2 + n_3 q c_3 + c_4 = p_1 c_4,$$

$$C_{10, 10} = q_1^2 c_4 + 4 r_1^2 c_4, C_{11, 11} = r_1^2 (c_3 + c_4), C_{12, 12} = r_1^2 (c_3 + c_4), C_{13} = C_{14} = -c_4,$$

$$C_{35} = C_{46} = -c_4, C_{45} = C_{46} = -c_4,$$

$$C_{56} = n_2 n_3 (c_3 + c_4), C_{56} = n_2 n_3 (c_3 + c_4), C_{66} = -n_2 p c_2,$$

$$C_{78} = -n_2 p c_2, C_{78} = -n_2 p c_2, C_{79} = -n_2 p c_2,$$

$$C_{79} = n_2 n_3 c_3, C_{79} = n_2 n_3 c_3, C_{79} = n_2 n_3 c_3,$$

$$C_{89} = -n_2 q c_3, C_{10, 11} = -2 r_1^2 c_4,$$

$$C_{10, 11} = C_{12, 10} = -2 r_1^2 c_4, C_{11, 12} = r_1^2 c_4, \dot{\theta} = a k.$$

where $\dot{\theta}$ corresponds to $a k$, and $\alpha$ is a constant. The elements $C_{11}$ and $C_{22}$ give rise to negative damping due to the negative slope of the friction characteristics. This is negative damping that contributes to the occurrence of shudder.

In the present paper, the use of a dynamic absorber is proposed in order to suppress shudder. As shown in Fig. 5, a dynamic absorber with a moment of inertia $J_d$, a damping coefficient $C_d$, and a spring coefficient $k_d$ is attached to the piston. The equations of motion for the piston and the dynamic absorber are given by Eq. (16) and Eq. (17), respectively:

$$J_d \ddot{\dot{\theta}}_d + c_2 (\dot{\dot{\theta}} - \dot{\theta}) + k_d (\dot{\theta} - \dot{\theta}) - \mu R^2 F \dot{\theta} = 0$$

$$J_d \ddot{\dot{\theta}} = c_2 (\dot{\dot{\theta}} - \dot{\theta}) + k_d (\dot{\theta} - \dot{\theta}) = 0$$

4. Results and Discussion of Numerical Calculations

4.1. Natural Frequencies and Natural Modes

Table 1 shows the undamped natural frequency $f_0$ and the amplitude of the first to fifth natural modes of the system when damping, external force and the coefficient of friction are set to zero. The natural frequency of the second mode is 35.0 Hz, which
is close to the shudder frequency. In the first mode, the components from the piston to the differential gear vibrate as a rigid body supported by the drive shaft. In the second mode, the piston vibrates with the largest amplitude, and all components from the turbine to the wheels vibrate in phase. These characteristics are consistent with the experimental results. In the third and fourth modes, the wheels vibrate with the largest amplitude, whereas in the fifth mode, the turbine vibrates with the largest amplitude.

4.2. Complex Eigenvalue Analysis

A complex eigenvalue analysis was performed taking into account the negative slope of the friction characteristics and system damping. In this analysis, the external force is set to zero. The friction characteristic is approximated using the linear function in Eq. (10), and \( \mu \) is set to 0.011 s/m. The value of \( \alpha \) in Eq. (14) is approximated as \( c_1 / k_1 \). The value of \( c_1 \) in Eq. (14) is determined by assuming that the real part of the eigenvalue for the second mode is 4.0 s\(^{-1}\). This ensures consistency with the experimental result that the shudder amplitude increased exponentially, with a value of 4.0 s\(^{-1}\) for the real part of the eigenvalue.

Table 2 shows the results of the eigenvalue analysis; here, \( f \) is the frequency and Re is the real part of the eigenvalue for each mode. The results indicate that the second mode is unstable because Re is positive.

In the experimental investigation of the characteristics of shudder described in Section 2, it was found that shudder rarely occurs in the first mode. To verify this numerically, the rate of change of Re with \( \mu \), defined as \( H \), was calculated using the following equation:

\[
H = \frac{\text{Re}_{\mu} - \text{Re}_{\mu_0}}{\Delta \mu}
\]

where \( \text{Re}_{\mu_0} \) is defined as the real part of the eigenvalue when \( \mu_0 = 0.011 \), \( \Delta \mu \) is the change in \( \mu \), and \( \text{Re}_{\mu} \) is the real part of the eigenvalue when \( \Delta \mu \) changes to \( \mu + \Delta \mu \).

Table 3 shows the calculated values of \( H \) for the five modes. Larger values indicate a tendency towards instability of the system. It can be seen that the second mode is most likely to become unstable, followed by the first mode. This is consistent with the actual vehicle experimental result that shudder often occurs in the second mode and sometimes in the first mode.

4.3. Effects of Internal and External Damping

As countermeasures against shudder, the effects of internal damping and external damping were investigated. First, the effect of internal damping on system instability was considered. Figure 6(a) shows the real part of the eigenvalue for the first and second modes when the damping \( c_1 \) between the piston and the turbine is \( s \) times larger than the fundamental damping value. The real part of the eigenvalue for the second mode becomes negative, which...
indicates the system is stable, because of increments in system damping. It can be seen that as $s$ increases, the second mode becomes more stable, but the first mode does not. For the second mode, the relative velocity between piston and turbine is large in shudder condition. Due to this reason, the suppression effect of shudder also is large. Meanwhile, the relative velocity between piston and turbine for the first mode is small. Due to this reason, the suppression effect of shudder also is small.

The effect of external damping was next considered. An additional damping $\tilde{c}$ is added to the turbine in the torque converter as a base support element, which increases the viscosity of the ATF. The fundamental value of $\tilde{c}$ is set to the value of $c_f$, and $s$ again represents the number of times that $\tilde{c}$ is larger than the fundamental value. Figure 6(b) shows the real part of the eigenvalue for the first and second modes as a function of $s$. Here, it is seen that as $s$ increases, there is no effect on the second mode, and the first mode becomes only slightly more stable. There is no effect of external damping on the second mode because the vibration amplitude of turbine is small in shudder condition.

4.4. Shudder Suppression Using Dynamic Absorber

The effectiveness of a dynamic absorber at suppressing shudder was next investigated. Although the optimum design of the dynamic absorber has been established in the field of forced vibrations, the optimum design of dynamic absorber for self-excited vibrations has not yet established. There have been some studies on the optimum design of dynamic absorbers for completely suppressing disc brake squeal in cars and bicycles\textsuperscript{11,12}. Two types of mechanisms were identified with regard to frictional vibration. The first is associated with asymmetry of the stiffness matrix, and the second involves a negative slope of the friction characteristics. The optimum design of the dynamic absorber depends on which mechanism is involved. Therefore, in the present study, the second mechanism is the one that should be focused on in order to suppress shudder.

Since the shudder has a vibration mode which is primarily the piston vibrates, the case in which the dynamic absorber is attached to the piston was considered. The moment of inertia $J_s$ of the dynamic absorber is assumed to be small, so as not to influence the main system. In this calculation, $J_s$ is set to be 3% of the moment of inertia of the piston $J_d$. The damping ratio $\zeta_s$ and the natural frequency $f_s$ of the dynamic absorber can be written as follows:

$$\zeta_s = \frac{c_f}{2\sqrt{J_d k_f}}, \quad f_s = \frac{1}{2\pi} \sqrt{\frac{k_f}{J_s}}$$  \hspace{1cm} (20)

The 13-degree-of-freedom equation of motion described in Section 3 was used in the analysis. Figure 7 shows Re plotted as a function of both $\zeta_s$ and the ratio of the natural frequency of the dynamic absorber $f_s$ to the second mode shudder frequency $f_d$. The white region represents the values for which the system is stable, because Re is less than zero. Larger values of Re indicate a tendency towards instability of the system. In order to completely suppress shudder, the dynamic absorber characteristics should be selected from the white region. From Fig. 7, it can be seen that the optimum characteristics are obtained when the natural frequency of the dynamic absorber is close to the shudder frequency, and a relatively large amount of viscous damping is applied to suppress shudder. This is the method used to design a dynamic absorber to suppress the frictional vibration caused by negative damping with respect to the relative slip velocity. If a dynamic absorber with a larger moment of inertia is used, the stable area will become large.

Based on these results, a dynamic absorber with the required characteristics was attached to the automatic transmission of an
actual vehicle, as shown in Fig. 8, and a travelling test was carried out. The dynamic absorber was a torsional type and was composed of steel and rubber. The rubber was used as a base support element, which exhibits stiffness and provides damping, whereas the steel provides the moment of inertia. The ratio of the moment of inertia of the steel to that of the piston was set to 10%. The natural frequency of the dynamic absorber was adjusted to be similar to the shudder frequency, and the damping ratio was approximately 0.1.

Figures 9(a) and 9(b) show the variation in the rotation speed of the turbine without and with the dynamic absorber, respectively. It can be seen that the dynamic absorber completely suppresses the shudder. When the frictional self-excited vibrations are suppressed by the dynamic absorber, neither the overall system nor the dynamic absorber vibrates.

4.5. Evaluation of Performance of Dynamic Absorber during Engine Forced Vibration

In the section 4.4, the optimum parameter of dynamic absorber to suppress shudder is calculated. The optimum parameter of dynamic absorber is selected from white region in Fig. 7 to make sure it can suppress the occurrence of shudder. The dynamic absorber with damping ratio \( \zeta_d = 0.1 \) and natural frequency ratio \( f_d / f_c = 1 \) is selected. This dynamic absorber is tested under engine forced vibration condition as stated in section 4.5.

A numerical analysis was next carried out to evaluate the performance of the dynamic absorber that design for suppressing shudder during forced engine vibration. In this case, the external force vector for the 13-degree-of-freedom system was set to \( T = [T_0, 0, \ldots, 0, 0, 0]^T \). The effect of the negative slope of the friction characteristics and system damping were taken into account. The engine speed was varied from 500 to 3000 rpm. Since a four-stroke engine was considered in this study, the forced vibration frequency was double the crankshaft frequency. The value of \( c_1 \) was increased to ensure that shudder did not occur in every stage of gear.

The fifth gear was again analyzed to determine the effect of the dynamic absorber in the non-lock-up, slip, and lock-up conditions of the torque converter. Figure 10 shows the dependence of the piston angular displacement amplitude on the engine speed.

Figure 10(a) shows the results for the non-lock-up condition. In this condition, although there is a speed difference between the engine and the piston, friction is not considered because the piston and torque converter are not in contact. It can be seen that there is no effect of the dynamic absorber on the system, because dynamic torque from the engine is not directly transferred to the piston.

Figure 10(b) shows the results for the slip condition. Again, there is a speed difference between the engine and the piston. In this case, since the piston and torque converter cover are in contact, friction must be taken into consideration. It can be seen that the dynamic absorber suppressed the amplitude of the forced vibration.

Finally, Fig. 10(c) shows the case for the lock-up condition. Here, there is no speed difference between the engine and the piston so friction is not considered. It can be seen that the second-mode frequency is reduced and there is no peak at an engine speed of 1050 rpm. In this case also, the dynamic absorber has no effect on the system.

The above results indicate that a dynamic absorber optimized to suppress shudder does not lead to any detrimental effects under engine forced vibration conditions.

The effect of the speed difference between the engine and the piston on the piston angular displacement amplitude under slip conditions was next investigated. Figures 11(a) and 11(b) show the results without and with the dynamic absorber, respectively. In both cases, the speed difference is seen to have little effect. This is because speed differences in this range do not significantly change the torque transfer characteristics.

Figure 12 shows the effect of the moment of inertia of the dynamic absorber on the system stability. It is clear that the piston angular displacement amplitude decreases with increasing moment of inertia. As shown in Fig. 11 and Fig. 12, the dynamic
Figure 11 Effect of speed difference for engine forced vibration

Figure 12 Effect of moment of inertia of dynamic absorber for engine forced vibration

The dynamic absorber that designed for suppressing shudder by stabilized the system is not vibrates on any condition. Therefore, there is no need of maintenance for the dynamic absorber. However, if the dynamic absorber is designed for forced vibration, the dynamic absorber vibrates in large amplitude only on the small rotational frequency range around 1000 rpm. The dynamic absorber does not vibrate continuously. Therefore, the dynamic absorber has high durability although in the engine forced vibration condition. If the nonlinear behaviors are generated due to the negative slope of the friction characteristics of the damper, the new optimum design for the dynamic absorber will be needed. However, in this experiments and analysis with specified parameters, nonlinear vibration does not occur.

5. Conclusion

This study focused on clarifying the mechanism that gives rise to shudder and evaluating the ability of a dynamic absorber to suppress it. The performance of the dynamic absorber under forced engine vibration conditions was also investigated. The following conclusions were drawn from the numerical and experimental results:

1. Shudder occurred because of the experimentally established negative slope of the frictional characteristics. The vibration frequency associated with shudder was experimentally determined to be 32.7 Hz, which was similar to the natural second-mode frequency of the system obtained from the numerical analysis.

2. The numerical analysis results indicated that only the second mode is unstable, and the piston vibrates primarily in this mode. However, the first mode can be generated if the negative slope of the frictional characteristics is changed.

3. Adding internal damping is effective for suppressing the shudder, whereas adding external damping to the turbine is effective for suppressing first-mode shudder.

4. Both the numerical and experimental results confirmed that shudder can be completely suppressed by the use of an optimized dynamic absorber. In addition, using the same dynamic absorber under engine forced vibration conditions led to no detrimental effects.
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