"Dynamic Analysis Of Single Plate Friction Clutch"

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Abstract:

In this study, a simple transmission system consisting of engine, clutch, gearbox and load is designed for the load lifting application.

Stiffness of all the three shafts have been calculated and equivalent stiffness is calculated. Equivalent mass moment of inertia is also calculated.

From this data, using the concepts given by Prof. DOW, duration of engagement period is calculated for the selected power transmission system and energy dissipated during engagement is also plotted as a function of time.

The effect of excitation torque and damping coefficient on the amplitude of vibration is plotted for various values of excitation speeds.

Results show that with increase in damping coefficient, the amplitude of vibration decreases and with decrease in excitation torque, vibration amplitude decreases.

Keywords: Clutch; Friction; Frictional torque; Engine torque; Damping coefficient; excitation torque.

1. INTRODUCTION:

The energy necessary for the motion of a vehicle is transmitted by the engine to the wheels through the flywheel, the clutch system and the driveline. The clutch takes the energy from the flywheel and transmits it to the driveline. During the engagement process, the friction torque acts upon the friction surfaces of the clutch as an engaging force for the driveline. A part of the energy transmitted through the driveline is transformed into other forms of energy by positive damping effects. If for some reasons the damping becomes negative, a part of the energy transmitted by the clutch can induce self-excited torsional vibrations of the driveline which can induce judder.

The vibrations induced by an internal combustion engine are transmitted to the passenger compartment through the engine mounts and through the driveline. The clutch system, being mounted between the flywheel and the gearbox, influences the driveline vibrations and finally the vibrations and noise perceived by the driver.

Friction clutch is treated as a power transmission path that is subjected to time-varying normal load. Friction clutch is an essential component in the process of power transmission, therefore all designers want to obtain the best possible performance with comfortable condition (reduce the noise and vibration as much as possible) for the friction clutches.

The vibration and noise generated during clutch engagement is one of the biggest obstacles faced by designers. There are several disadvantages produced by vibration such as excessive and unpleasant stresses, rapid wear, large amplitude produced by resonance which will lead to failure of the system.

To increase the efficiency of performance and reduce the noise, it is necessary to investigate the dynamic characteristics of system employing friction clutches.

In order to study dynamic analysis of single plate friction clutch, system consisting of prime mover (engine), clutch, gear box and load shown in fig. has been considered. All the parts of the systems have been designed and using these dimensions, stiffness and mass moment of inertias have been calculated.

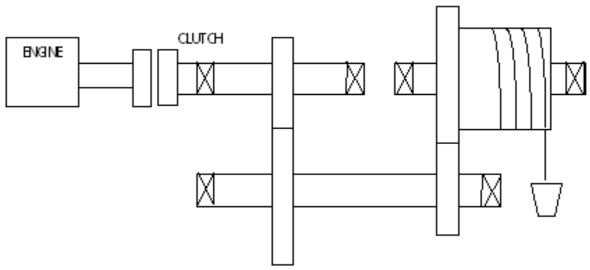


Figure 1: Load arrangement

This analysis is applicable to the considered system, but the same can be applied to any automobile transmission system containing manual clutch by simply altering the data.

2. SIMPLIFIED APPROACH TO DYNAMIC PROBLEMS INVOLVING FRICTION CLUTCHES [3]

In designing power- transmission system containing friction clutches situation frequently arise in which it is required to calculate energy dissipation in the clutch during the engagement period or simply to compute the inertial torques acting on the system during a transition from one ratio to the other.

This procedure is justified in cases where the natural frequencies of the system are relatively high and the resulting amplitude of oscillation small in comparison with the changes in velocities which take place as a consequence of the application of the clutch.

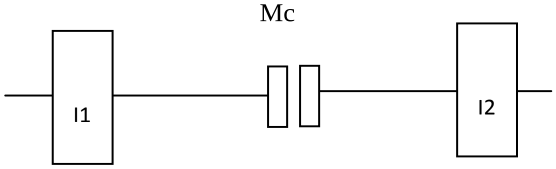


Figure 2: Simple two-inertia system

Consider a simple two- inertia system shown in Figure 2. Here, the coefficient of friction is assumed to be a constant, independent of speed. It is also assumed also that no external torques act on the system.

In Fig. 2, the two flywheels initially rotate at two different angular velocities Ω_1 and Ω_2. Let the clutch be instantaneously applied at time t = 0 and let the torque exerted by it on the two flywheels me M_c = constant. This torque will act on the system only as long as these exists a difference in speeds between the two shafts

Equations for speed, rate of heat dissipation etc., are therefore valid for time $0< t< t_0$, where t_0 is the time required to couple the two shafts.

The equations of motion for the two sides of the clutch are

$$I_1(d\omega_1/dt) = -M_c \text{ ....(1.1)}$$

$$I_2(d\omega_2/dt) = M_c \text{ ... ... (1.2)}$$

Where $\omega_1 = \dot{\theta}_1$ and $\omega_2 = \dot{\theta}_2$ are the angular velocities of I_1 and I_2, respectively. Integrating equations (1.1) and (1.2) and applying the conditions at $t = 0, \omega_1(t) = \Omega_1, and\ \omega_2(t) = \Omega_2$, yeilds

$$\omega_1(t) = -(M_c/I_1)t + \Omega_1 \text{ .....(1.3)}$$

$$\omega_2(t) = -(M_c/I_2)t + \Omega_2 \text{.....(1.4)}$$

The relative velocity of I_1 with respect to I_2 is given by

$$\omega_1(t) - \omega_2(t) = \omega_r(t) = -M_c[(I_1 + I_2)/I_1I_2]t + \Omega_1 - \Omega_2 \text{ ....(1.5)}$$

The rate at which energy is dissipated in the clutch during the engagement period is given by $q(t) = M_c\{\Omega_1 - \Omega_2 - M_c[(I_1 + I_2)/I_1I_2]t\}$......(1.6)

The duration of the engagement period, found by considering that when $t = t_0, \omega_1(t) = \omega_2(t)$, is given by

$$t_0 = I_1I_2(\Omega_2 - \Omega_2)/M_c(I_1 + I_2) \quad \text{sec.......(1.7)}$$

The total energy dissipated is obtained by integration of Eq. (1.6):

$$Q = \int_0^{t0} q(t)\,dt = \frac{I_1 I_2 (\Omega_1 - \Omega_2)^2}{2(I_1 + I_2)} \quad N.m \ldots \ldots (1.8)$$

The efficiency with which energy is transferred from I_1 to I_2 during the clutching operation is:

$$\eta = 1 - Q/E_0 \ldots (1.9)$$

Where, E_0 = kinetic energy of the system at a time $t = 0$.

Thus, $$\eta = \frac{(I_1\Omega_2 + I_2\Omega_2)^2}{(I_1+I_2)(I_1\Omega_1^2 + I_2\Omega_2^2)} \ldots\ldots (1.10)$$

For the considered application,

M_c = *Frictional Torque*: $M_c = 196.34$ N.m

At t= 0: $\omega_1(t) = \Omega_1$ and $\omega_2(t) = \Omega_2$

From equation 4.7,

Duration of engagement period:

$t = t_0$, $\omega_1(t) = \omega_2(t)$

$$t_0 = \frac{I_1 I_2 (\Omega_1 - \Omega_2)}{M_c (I_1 + I_2)} \text{ sec}$$

For the considered system,

$I_1 = 0.2$ kg.m^2; $I_2 = 0.149$ kg.m^2; $M_c = 196.34$ N.m.

$\Omega_1 = 2400$ rpm

$= 251.32$ rad/sec

$\Omega_2 = 1000$ rpm (Assuming that clutch is applied at output shaft speed 1000rpm)

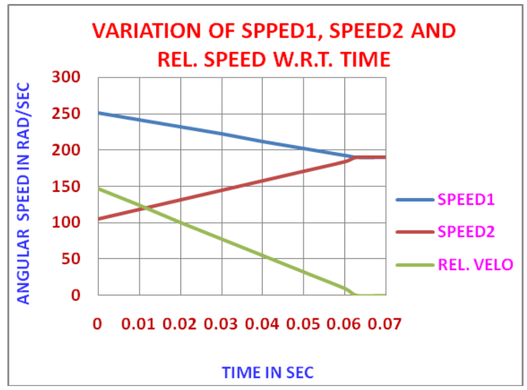
$= 104.71$ rad/sec

Then, duration of engagement period,

$$t_0 = \frac{I_1 I_2 (\Omega_1 - \Omega_2)}{M_c (I_1 + I_2)} \text{ sec}$$

$$\therefore t_0 = \frac{0.2 \text{ X } 0.149\ (251.32 - 104.71)}{196.34 \text{ X } (0.2 + 0.149)}$$

$$t_0 = 0.063 \text{ sec}$$



From equation 1.6, the rate at which the energy is dissipated in the clutch during engagement period is given by:

$$q(t) = M_c \left\{\Omega_1 - \Omega_2 - M_c \left[\frac{(I_1 + I_2)}{I_1 I_2}\right] t\right\}$$

For different values of 't', energy dissipated in the clutch during engagement period is calculated and tabulated as below:

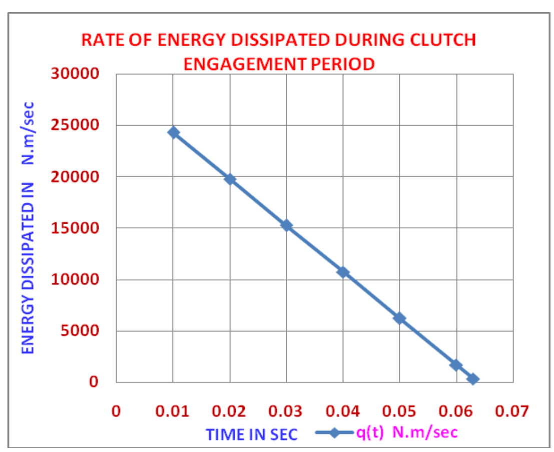


Table: 1

time(sec)	q(t) N.m/sec
0.01	24270.72
0.02	19756.05
0.03	15241.37
0.04	10726.69
0.05	6212.01
0.06	1697.34
0.063	342.93

3 AN APPROACH TO ESTIMATE VIBRATION RESPONSE OF UNIT [4]

3.1 EQUATIONS OF MOTION OF THE MECHANICAL SYSTEM

We assume a three- mass model of a mechanical system which consists of an engine (E), friction clutch(C), reduced mass (RM) and a working machine (WM), as shown in fig.1. Structural friction occurs between the cooperating surfaces of discs of a friction clutch (C).

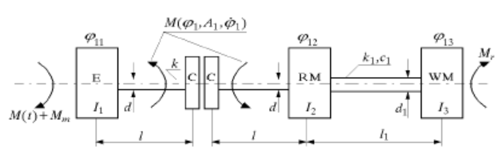


Figure 3: Physical model of considered power transmission system

Therefore, equation of motion of the considered system may be written down as follows

$$I_1\ddot{\varphi}_{11} + M_z = M(t) + M_m$$

$$I_2\ddot{\varphi}_{12} - M_z + k_1(\varphi_{12} - \varphi_{13}) + c_1(\dot{\varphi}_{12} - \dot{\varphi}_{13}) = 0$$

$$I_3\ddot{\varphi}_{13} - k_1(\varphi_{12} - \varphi_{13}) - c_1(\dot{\varphi}_{12} - \dot{\varphi}_{13}) + M_r = 0$$

Where

I_1, I_2, I_3 – mass moment of inertia of the driving and driven part, respectively.

φ_{11}, φ_{12}, φ_{13} – angular displacements

M_z – clutch friction torque dependent on the relative angular displacement, its vibration amplitude and its sign of velocity.

C_1 – coefficient of viscous damping

M_r – resistance torque

M_0 – amplitude of the excitation torque

ω – Angular velocity of the excitation torque

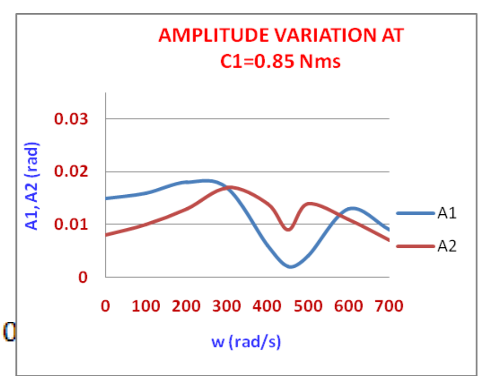
t – Time

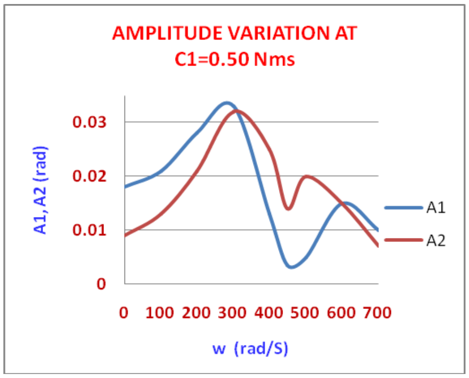
The solution to the equation of motion is as given below:

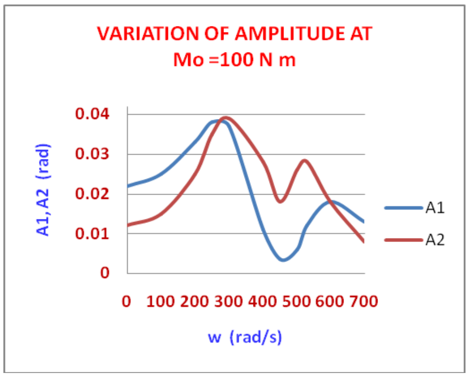
$$T_8A_1^8 + T_7A_1^7 + T_6A_1^6 - T_5A_1^5 + T_4A_1^4 + T_3A_1^3 + T_2A_1^2 + T_1A_1 + T_0 = 0 \quad \ldots\ldots(1)$$

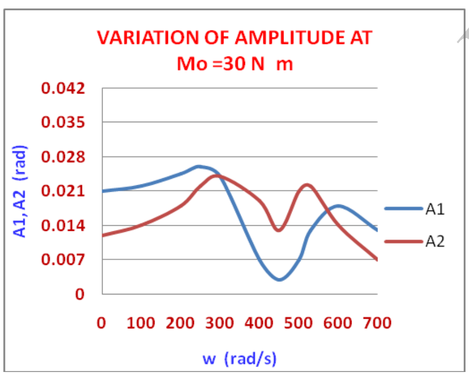
$$A_2^2 = \frac{x}{\pi^2 I_{21}^2}\left\{\frac{32A_1}{9\eta 3} + \left[4F\left(\frac{2}{3}c_2A_1^2\omega^2 - a_2\right) - \frac{4\sqrt{A_1}}{3\sqrt{2\eta_s}}\right]\right\} \quad \ldots(2)$$

Equation 1 was solved and any one from the eight roots of equation which would satisfy the physical condition is chosen. For such a value of A_1, the value of A_2 was calculated with formula 2 in function of the forced vibration frequency.









CONCLUSION:

1. From the equation1.7 , duration of engagement period is calculated and found to be equal to 0.063 sec.
2. For various values of 't', energy dissipated w.r.t. engagement period is plotted and graph shows that during engagement, energy dissipated goes on decreasing.
3. Amplitude of vibration depends on damping coefficient and it decreases with increase in damping coefficient.
4. Amplitude of vibration also depends on excitation torque and it decreases with decrease in excitation torque.

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