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VIBRATION ANALYSIS OF FINGER TYPE OF CLUTCH USED IN POWER TRANSMISSION IN HPFM

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ABSTRACT: The Human Powered Flywheel Motor (HPFM) system uses human muscular power and stores the energy in the flywheel in the form of a rotational kinetic energy at an energy input rate convenient to man. After storing maximum possible energy in the flywheel, the same can be made available to the process unit upon engaging the clutch. The flywheel deceleration time after clutch engagement depends on the process resistance. Hence the processes which are of intermittent nature and non-sensitive to speed variation of the input shaft of the process unit can be energized implementing this concept of (HPFM) Human Powered Flywheel Motor as the energy. Spiral jaw clutch was used for engagement to transfer the flywheel energy to process unit. With the spiral jaw clutch operational difficulties of frequently break down of the jaw tip. The breakage of the jaw tip was due to severe impact, shock and wear.

It is required to have controlled acceleration device between flywheel shaft and input shaft of the process unit. The present situation needs a positive clutch having the capability of withstanding impact, shock and wear. These clutches are decided to be best for energizing process units adopting HPFM as an energy source. Clutches have fingers for transmission of driving forces and energy necessary for process units. These fingers work like cantilevers which are subjected to varying forces which induces forced vibrations in the fingers. The paper detailed the vibration analysis of fingers in a finger type torsional flexible clutches necessary for HPFM energies process machines.

KEYWORDS: Human power; Clutch; Vibration; Flywheel; Finger.

1 INTRODUCTION

On an average the horse power of an individual is 0.13. Continuous process of production of any article can be made manual if the horse power requirement of the process is about 0.13. If the process is of intermittent nature without affecting the end product, a machine system working on following concept can be used. Human Powered Flywheel Motor (HPFM) machine consists of three units, namely (1) Energy Unit (2) Appropriate clutch and torque amplification unit and (3) Process unit.Human Powered Flywheel Motor energized machine were used for various process units. Human Powered Flywheel Motor has been established for many village/rural based applications such as brick making, wood turning, algae formation process, wood strips cutting, smith’s hammer, winnowing, chaff cutting, mixing of ingredients of fertilizer mixer, electricity generation etc. Spiral jaw clutch is suggested because due to the helix cut, it can be engaged even if the input shaft is in motion and the shaft to be connected is on load and stationary. Upon engagement of the clutch in manually energized machine there is an instantaneous momentum exchange between the flywheel shaft and the rest of the system. On account of this momentum exchange there is generation of severe impact at the jaw tip. The magnitude and frequency of these forces is unpredictable. The magnitude of this generated force at the jaw tip reduces with time. This force is maximum at the instant of clutch engagement. But as the magnitude of generated force changes with time, it creates severe time varying stresses at jaw tip. This induces fatigue at the jaw tip resulting eventually in to a tip failure. The problem of development of controlled acceleration devices is not new in the literature of mechanical power transmission. Several practical situations do need these devices and in fact there are several scholarly contributions on the development of controlled acceleration devices. In order to overcome the above stated problems of spiral jaw clutch conceptually, therefore, it is required to have appropriate flexible member which will transfer energy from flywheel shaft to the load shaft gradually instead of almost instantaneously. This difficulty is overcome by torsionally flexible clutch which will transfer energy from flywheel shaft to the load shaft gradually instead of almost instantaneously. Force vibration analysis is done on the finger of the clutch which act as the cantilever beam. Fingers are subjected to sever transient forces which make fingers to act as cantilever beams. The forces are investigated and established experimental data based models in transient dynamics of finger type clutches using Laplace transform method.

2 MACHINE SYSTEM

The HPFM machine system uses human muscular power and stores the energy in the flywheel in the form of a rotational kinetic energy at an energy input rate convenient to man. After storing maximum possible energy in the flywheel, the same can be made available to the process unit upon engaging the clutch. The flywheel deceleration time after clutch engagement depends on the process resistance. Greater the resistance faster is slowing down. Thus, the process resistance even up to infinity can be overcome by this concept. Hence the processes which are of intermittent nature and non sensitive to speed variation of the input shaft of the process unit can be energized implementing this concept. The schematic of this machine is shown in Figure 1.

Human Powered Flywheel Motor (HPFM) machine consists of three units, namely (1) Energy Unit (2) Appropriate clutch and torque amplification unit and (3) Process unit.

i. Energy Unit

Energy unit consist of manually energized flywheel using a bicycle mechanism and a speed increasing gear pair. This energy unit is abbreviated as Human Powered Flywheel Motor (HPFM).

ii. Process Unit

Process unit consist of processing elements required for as per the requirement.

iii. The clutch and the transmission

The transmission unit consists of clutch to transmit the energy from flywheel to process unit.

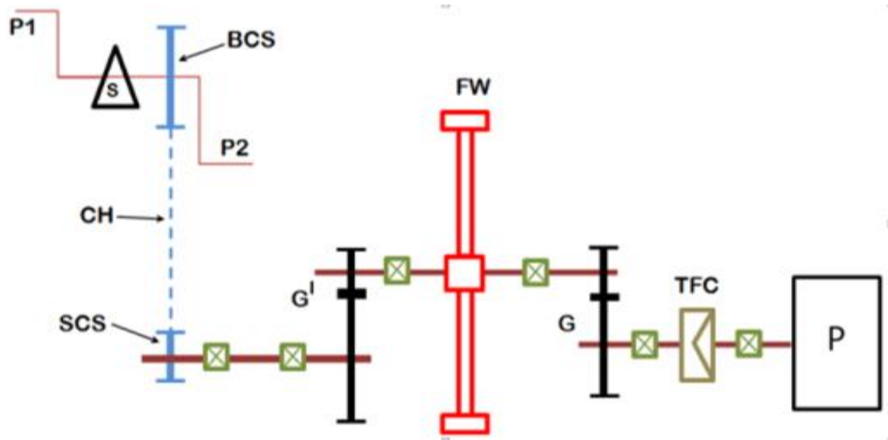

Figure 1. Schematic arrangement of the Human Powered Flywheel Motor

The machine developed on the basis of this concept consists of
S = Seat, P1, P2 = Pedals, BSC = Big Chain Sprocket, SCS = Small Chain Sprocket, CH = Bicycle Chain Drive, GI = Speed increasing gear pair, FW = Flywheel, TFC/SJC = Torsion ally Flexible/Spiral Jaw Clutch, P = Process Unit, G = Torque Amplification Gear pair

If the power requirement of the process is more than 0.13 HP and if the process can be of an intermittent nature without affecting the end product, a machine system working on the above concept is possible. This concept has been fully developed for some applications like, wood-turning [4], potter's wheel [5], wood strips cutting [6], algae formation [8], brick making [1,2, 9, 10], fertilizer mixing [11], chaff cutting [12] etc. This development of HPFM energized process machine is proved to be functionally feasible and economically viable.

3 CLUTCH AND TRANSMISSION

3.1 Spiral jaw clutch

In a manually energized process machine spiral jaw clutch is used. It has considerable operational difficulties on account of frequent break down of the jaw tip. The breakage of the jaw tip was due to severe impact, shock and wear. This is because upon engagement of the clutch, the process unit shaft used to get suddenly accelerated and then the flywheel shaft and the load shaft are subjected to deceleration on account of overcoming the process resistance. The operational difficulties of spiral jaw clutch necessitated the development of such a clutch which can be subjected to frequent ON-OFF of on load starting machine. In order to overcome the above stated problems of spiral jaw clutch conceptually, therefore, it is required to have appropriate flexible member which will transfer energy from flywheel shaft to the load shaft gradually instead of almost instantaneously [13].

3.2 Development of new type of clutch

In the existing manually energized machine, there is an instantaneous momentum exchange between the flywheel shaft and the rest of the system. On account of this momentum exchange there is generation of severe impact at the jaw tip. The magnitude and frequency of these forces is unpredictable. The magnitude of this generated force at the jaw tip reduces with time. This force is maximum at the instant of clutch engagement. But as the magnitude of generated force changes with time, it creates severe time varying stresses at jaw tip. This induces fatigue at the jaw tip resulting eventually in to a tip failure.

Hence, it is required to have controlled acceleration device between flywheel shaft and input shaft of the process unit. It should have a provision of bringing –in this subsystem in a power flow of main system or keeping it out from the power flow of main system. The problem of development of controlled acceleration devices is not new in the literature of mechanical power transmission [14-15]. Several practical situations do need these devices and in fact there are several scholarly contributions on the development of controlled acceleration devices. There are three basic characteristics of the present situation which warrants adoption of above said controlled acceleration devices.

i. These devices are very costly which would defeat the original concept of evolution of manually energized process machine.
ii. These devices presume that the input motion is constant and at high speed. This condition is not getting fulfilled in the present main application because upon clutch engagement the flywheel shaft speed starts dropping.
iii. Some controlled acceleration devices are not feasible to operate at low speeds.

Therefore, although literature has an answer for controlled acceleration devices, the present situation needs an altogether different kind of controlled acceleration device having low power capacity, slower speeds of operation, low cost and should be in a position to sustain severe impact, shock and wear. This leads to the development of new controlled acceleration devices, having following salient features.

a. Capability of connecting the high-speed flywheel shaft to the stationary load shaft.
b. Capability of withstanding severe impact and associated shock.
c. Capability of resisting transverse severe wear between elements connecting the two sub systems.
d. Gradually accelerating the load shaft.

A solution to this problem is to be evolved. Conceptually what is needed is a clutch, which will gradually accelerate the driven shaft. This means the clutch has to permit required rigid body relative displacement. The friction clutches permit a relative rigid body displacement. Accordingly, an attempt was made to design, develop and test a single plate friction clutch for this machine.

Although, clutch required is a positive having the capability of withstanding impact, shock, wear and to transmit flywheel energy to the load shaft with controlled acceleration. Such clutches have not been reported so far in the literature. What is reported in the literature is the clutches with axial flexibility. Therefore, a clutch, with such a provision, is conceptualized for the first time in the literature as “Torsional Flexible Clutch”.

3.3 The proposal for new type of clutches

3.3.1 Finger type clutch

This clutch is described in Figure 2(a). It consists of two parts A and B.

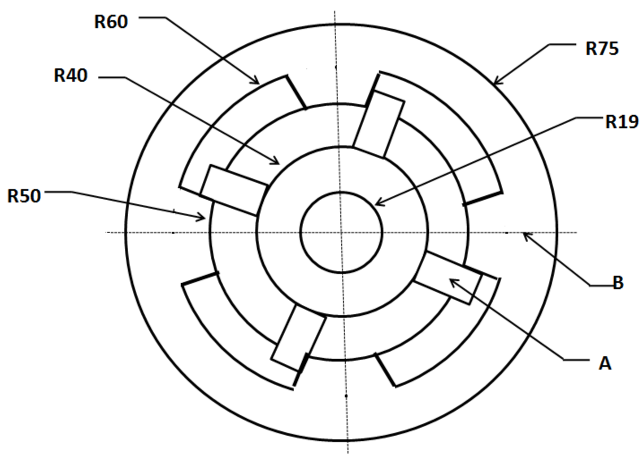

Figure 2(a). Finger type clutch

Part B has four internal jaws at equal angular spacing fixed to a load shaft at required position. Part A is connected to the flywheel shaft having four fingers at equal spacing on the outer periphery. This part has an axial movement by virtue of splines on the shaft, bush and shifter arrangement.

3.3.2 Face tooth type clutch

This clutch is described in Figure 2(b). Two identical parts A & B having four jaws on its end faces. B is rigidly fixed to the load shaft while A is mounted on the flywheel shaft. Part A can perform an axial movement by virtue of splines with zero relative angular rotation with respect to flywheel shaft. The axial movement of A is initiated with the help of shifting lever.

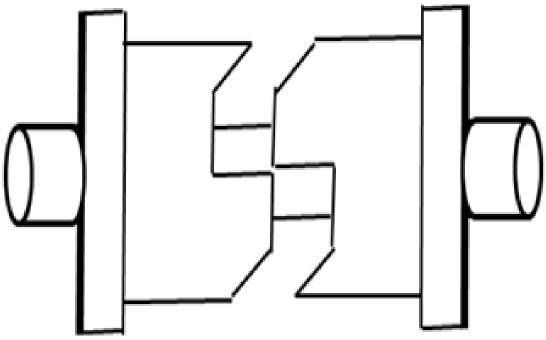

Figure 2(b). Face tooth type clutch

3.3.3 Toothed gear type clutch

This clutch is described in Fig. 2(c). Two toothed gears A and B are keyed to flywheel shaft and load shaft respectively. The ring gear C with same number of teeth as on A and B meshes with A. C is made to slide axially on A in order to engage with B. Amongst these the FINGER TYPE CLUTCH is the best one from the point of view of imparting controlled and gradual acceleration of the input shaft of the process unit.

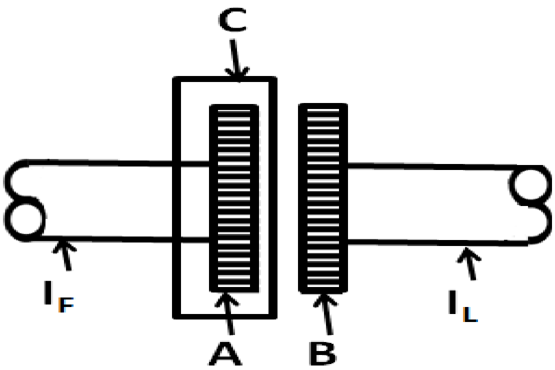

Figure 2(c). Toothed gear type clutch

4 ESTIMATE OF FORCE TRANSMISSION BY FINGERS

Fingers are subjected to sever transient forces which make fingers to act as cantilever beams. The force estimation is not possible based on logic here the earlier investigations and established experimental data based models in transient dynamics of finger type clutches. In this approach all independent quantities involved in the phenomena of force and kinetic energy transmission are varied experimentally over widest possible range, response data is collected.

Based on this response data the analytical relationships are established correlating causes and effects of the phenomena of severe transient dynamics of these clutches during clutch engagement period. Established models for angular acceleration induced in the flywheel shaft are as under.

$$\ddot{\theta} \;\; x \;\; t \;\; = \left[(-0.816)\;\left(\frac{I_f}{T_l t^2}\right)\right]^{-1.13877} \left[\frac{I_l}{T_2 l}\right]^{-0.0448} + \left[(-0.0101)\;\left(\frac{I_f}{T_l t^2}\right)\right]^{-1.0448} \left[\frac{I_l}{T_2 l}\right]^{-0.03877} \tag{1}$$

The nomenclature for Eq. (1) is as below:
I_f = Moment of Inertia of Flywheel Shaft, I_f = 9.0 kgf-m-Sec^2
I_l = Moment of Inertia of Load Shaft, I_l = 0.9 kgf-m-Sec^2
t = Arbitrary time constant during clutch engagement period
t is changed in the interval 0.02 to 0.07 seconds.
The variation of the angular acceleration of flywheel shaft as t has changed is as shown in Table 1 given below:

Table 1. Variation of angular acceleration of flywheel shaft versus t

Sr. no.	t	$\ddot{\theta}_f * t$	$\ddot{\theta}$
1	0.02	-0.00415	-0.2075
2	0.03	-0.00590	-0.1966
3	0.04	-0.00756	-0.1890
4	0.05	-0.00968	-0.1936
5	0.06	-0.01238	-0.2063
6	0.07	-0.01224	-0.1749

The finger tip load F can then be estimated based on transient dynamics of flywheel shaft as under.

$$- \;\; I_f \;\; \ddot{\theta}_f \;\; - \;\; b_f \;\; \ddot{\theta}_f \;\; - \;\; NFL = 0 \tag{2}$$

In Eq. (2)
bf =Bearing friction torque constant
N= Number of fingers
L = length of the finger
As the bearings are well maintained and amply lubricated
N = 4.0, L = 2.0 cm

5 LOAD CALCULATION AT FINGER TIP

The finger tip load F can then be estimated based on transient dynamics of flywheel shaft as under Equation 2.

$$F = \left[\frac{\left(I_f \quad \ddot{\theta}_f \right)}{(N \quad L)} \right]$$

At time t = 0.02 seconds

$$\ddot{\theta} = 0.2075$$

Further,

$$I_f = 9.0 \text{ Kgf m sec}^2$$

$$\therefore F = \frac{9.0 * 0.2075}{4 * 2} * 100$$

$$F = 23.34 \text{ kgf}$$

Likewise, for the other time instants such as 0.03, 0.04, 0.05, 0.06 & 0.07 seconds values of F would be 22.1175kgf, 21.26 kgf, 21.78 kgf, 23.20 kgf, 19.67 kgf respectively. Figure 3 show the variation of Force F with the specific time instant t in seconds.

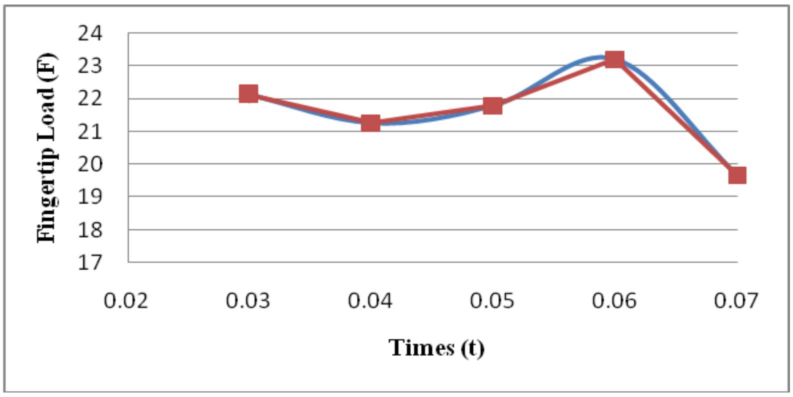

Figure 3. Finger Tip Load F versus t the time instant during clutch engagement period.

The variation of F versus time t in higher order polynomial form will be as under.

$$F= 19.15 +9.458t - 6.898t^2 + 1.752t^3 -0.143t^4 \quad (3)$$

Neglecting the 3rd term and onwards on right hand side of Eq. (.3)

$$F = 19.15 + 9.458t - 6.898t^2 \quad (4)$$

6 ESTIMATION OF FINGER BENDING VIBRATIONS

Figure 4 show schematically a single finger being treated as cantilever beam subjected to fingertip time varying force F inducing bending vibrations in the finger. The mass of the finger will be M, the bending stiffness at tip is K and C is the damping coefficient. Then the governing equation of vibratory motion of the tip of the finger would be.

$$M\ddot{x} + C\dot{x} + Kx = 19.15 + 9.458t + 6.898t^2 \qquad (5)$$

In the above Eq. (5) M, C and K are respectively mass of the finger, damping co-efficient of the finger and stiffness of finger at the tip

Now, $M = 8*1*1*\frac{2}{981}10^{-3}\,kgfcm^{-1}Sec^2 = 1.6\times10^{-5}\,kgfcm^{-1}Sec^2$

Now, $K = 3EI/L^3$

Substituting, the appropriate numerical values in the above equation

$$K = (3*2.6*10^6)/\,(2.0)^3*12 = 0.8125*10^5 \text{ Kgf/cm}$$

Further $C = 0.7\ c_c$

This is so because the damping is only due to material internal friction or hysteresis. But as $c_c = 2\sqrt{k}$ where C_C is the critical damping co-efficient

$$C = 0.7*2*\sqrt{k}$$

Substituting for K= 0.8125 x 105 Kgf / cm

$M = 1.61 * 10^{-5}$ $Kgfcm^{-1}Sec^2$

$C=0.07*2*(0.8125 \text{ x } 10^5 \text{ x } 1.610 \text{ x } 10^{-5})^{1/2}$

C = 1.6307 kgf / cm /sec

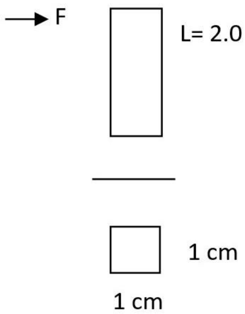

Figure 4. F inducing bending vibrations

Equation (5) will now take the form as under

$$(1.61 \text{ x } 10^{-5}) + (1.6307) + (0.8125 \text{ x } 10^5)\, x = 19.15 + 9.458\, t - 6.89 \qquad (6)$$

Performing Laplace transform on Eq. (6) with all initial condition to be equal to Zero

$(1.61\times10^{-5})\, s^2 x(s) + (1.6307)\, s\, x(s) + (0.8125\times10^5)\, x(s) = (19.75/s) + (9.45/s^2) - (6.5/s^3)$ (7)

Equation (7) presents fingertip vibratory motion. It is now necessary to estimate x(t) for different time instants t such as t=0.02, 0.03 ,0.04 ,0.05 ,0.06 and 0.07 seconds The second and third term on RHS of Eq. (7) shows that the tip deflection will be highly oscillatory. As the clutch engagement time duration is going to be very very small of the order of not more than 100 milliseconds, it is proposed to consider only first term on right hand side.

The value of x (t) for variation of t in the range t = 0.01 to 0.1 second would be as under in Table 2 given below.

Table 2. Fingertip vibratory response

Sr.No.	Times	x(t)
1	0.02	0.00552
2	0.03	0.01242
3	0.04	0.02208
4	0.05	0.0345
5	0.06	0.04968
6	0.07	0.06762
7	0.08	0.08832
8	0.09	0.11178

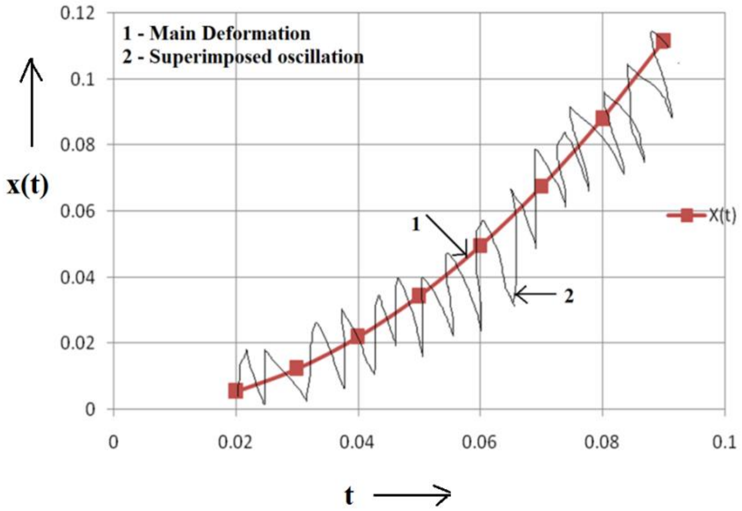

Figure 5. Variation of Finger Tip (Elastic Deformation) verses time t

7 DISCUSSION OF RESULTS

The vibration responses evaluated are as under

Influence of axial rub

The axial rub between the surface of fingertip and corresponding surface of jaw along the shaft axis causes lot of loss of flywheel energy. This lot of flywheel energy losses would reduce flywheel stored kinetic energy. This will result into further deceleration of flywheel. This will reduce fingertip vibration.

Finger flexibility as a cantilever

The flexural ability of fingers in this case is much less. This so because, cross sectional area of the fingers is much higher as compared to cantilever length. In addition, the fixity of fingers at base is introducing considerable error in the estimation of fingertip vibration.

Influence of finite area of tip

The finger is treated as cantilever. Accordingly, the fingertip deflection is estimated as if the load at end of the finger is acting at real geometric end point. But in fact, it is acting on some finite area of tip. Hence, it is distributed over certain length of the finger from actual end of the tip. In addition, the length is so less as compared to the cross section of the finger that estimation of fingertip load as a conventional cantilever would cause much less tip deflection than the one estimated in this investigation.

Influence of shaft deflection

The one half of the clutch is on one end of flywheel shaft. The other half is on the end of the input shaft of torque amplification gear pair. Therefore, the contact of surfaces of fingertip and the corresponding surface of the jaw are not exactly containing the complete line of axis of two shafts. This makes the ideal force analysis as assume in this case fairly erroneous.

Treatment of random vibration

In the present investigation the figure 5 is in fact a transient vibration signal. It can also be considered as a random signal. In that case it will be worthwhile establishing harmonic analysis of this transient signal

4 CONCLUSIONS

The paper presents possible vibration analysis of fingers in a finger type torsional flexible clutches necessary for HPFM energies process machines. Paper have relevant for estimation of induced stresses in the column of the bridge particularly for small size bridge where the column length would be fairly small compared with length of bridge.

NOMENCLATURE

b_f = Bearing friction torque constant
C = Damping co efficient of the finger
F = Finger tip load (kgf)
I_f = Moment of Inertia of Flywheel Shaft
I_l =Moment of Inertia of Load Shaft
L = Length of fingers
M = Mass of the finger (Kg)
N = Number of fingers
t = Specific time instant during clutch
θ_f = angular acceleration of flywheel shaft

CONFLICT OF INTEREST

The authors confirm that there is no conflict of interest to declare for this publication.

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