The energetic balance of the friction clutches used in automotive

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Abstract: - The energetic transfer through the friction clutch is accompanied by dissipative processes. The causes that produce these processes are the friction from the plate package. The dissipative restrictions that are independent of the temperature gradient or the direction of the heat flux present in the clutch's mass are imposed by the maximum temperatures admitted by the friction materials, by the electrically insulating materials and the lubricant (if it takes place in a wet environment).

Key-Words: - clutch, friction, energy, load, dissipative, heating

1 Introduction

A clutch is a device which transfers energy for one rotating shaft to another in order to perform some useful work.

In the simplest terms, a clutch can be thought of as a starting device because that is what happens when a clutch is engaged. But, more importantly, while engaged it is transferring energy. The clutch takes energy from a power source such as an engine and transfers it to where it is required.

A clutch consists on two halves: a driving half and a driven half. The driving half is attached to the power source and rotates with it. The driven half is attached to the shaft requiring the energy and is started with each engagement. In addition, the clutch must have some means of engaging and disengaging the two halves.

Friction couples rely upon a frictional force occurring between two surfaces to develop the required torque. The torque is called dynamic torque when slippage occurs between the surfaces and static torque when no slippage occurs. Usually the two surfaces are of dissimilar materials. The combination of the two materials used is referred to as friction couple and their contacting surfaces as interfaces [7].

When the friction couples operates within a fluid, it is referred to as wet operation. Dry operation does not depend upon the presence of fluid. Also within the wet clutch oil additives, especially the extreme pressure/anti-wear additive has beneficial effects on the frictional characteristics and the wear of the friction material [3].

The torque transmitted is related to the actuating force, the coefficient of friction, and the geometry of the clutch or brake. This is a problem in statics which will have to be studied separately for each geometric configuration. However, temperature rise is related to energy loss and can be studied without regard to the type of brake or clutch, because the geometry of interest is that of the heat-dissipating surfaces [4].

The centrifugal clutch is used mostly for automatic operation. If no spring is used, the torque transmitted is proportional to the square of speed. This is particularly useful for electric-motor drives where, during starting, the driven machine comes up to speed without shock. Springs can also be used

ISBN: 978-1-61804-004-6

to prevent engagement until a certain motor speed has been reached, but some shock may occur.
Magnetic clutches are particularly useful for automatic and remote-control systems. Such clutches are also useful in drives subject to complex load cycles.
Hydraulic and pneumatic clutches are also useful in drives having complex loading cycles and in automatic machinery, or in robots. Here the fluid flow can be controlled remotely using solenoid valves. These clutches are also available as disk, cone, and multiple-plate clutches [4].
The thermal process caused by the relative slip of the friction surfaces (in direct contact or by the use of the lubricant) is the most intense and meaningful process. It’s importance in the energetic balance of engaging and disengaging can be shown through a simple artifice applied to the motion equations.

2 Energy considerations

When the rotating members of a machine are caused to stop by means of a brake, the kinetic energy of rotation must be absorbed by the brake. This energy appears in the brake in the form of the heat. In the same way, when the members of a machine which are initially at rest are brought up to speed, slipping must occur in the clutch until the driven members have the same speed as the driver. Kinetic energy is absorbed during slippage of either a clutch or a brake, and this energy appears as heat.
We have seen the torque capacity of a clutch or brake depends upon the coefficient of friction of the material and upon a safe normal pressure. However, the character of the load may be such that, if this torque value is permitted, the clutch or brake may be destroyed by its own generated heat. The capacity of a clutch is therefore limited by two factors, the characteristics of the material and the ability of the clutch to dissipate heat. In this section we shall consider the amount of heat generated by a clutching or braking operation. If the heat is generated faster than it is dissipated, we have a temperature-rise problem.
In the following drawing you can get a clear picture of what happens during a simple clutching or braking operation, which is a mathematical model of a two-inertia system connected by a clutch. As shown, inertias I_1 and I_2 have initial angular velocities of ω_1 and ω_2, respectively. During the clutch operation both angular velocities change and eventually become equal. We assume that the two shafts are rigid and that the clutch torque is constant.

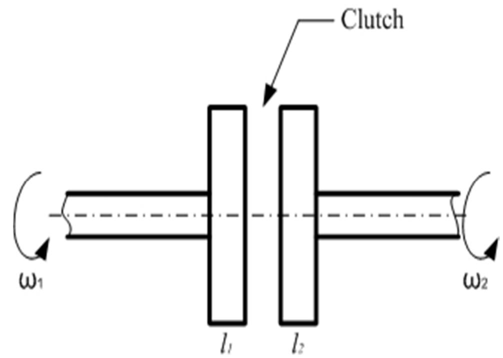

Fig. 1. Dynamic representation of a clutch.

Writing the equation of motion for inertia 1 gives:

$$I_1\ddot{\theta}_1 = -T \qquad (1)$$

Where $\ddot{\theta}_1$ is the angular acceleration of I_1 and T is the clutch torque. A similar equation for I_2 is:

$$I_2\ddot{\theta}_2 = T \qquad (2)$$

We can determine the instantaneous angular velocities $\dot{\theta}_1$ and $\dot{\theta}_2$ of I_1 and I_2 after any period of time *t* has elapsed by integrating equations (1) and (2). The results are:

$$\dot{\theta}_1 = -\frac{T}{I_1}t + \omega_1 \qquad (3)$$

$$\dot{\theta}_2 = \frac{T}{I_2}t + \omega_2 \qquad (4)$$

The difference in the velocities, sometimes called relative velocity, is:

$$\dot{\theta} = \dot{\theta}_1 - \dot{\theta}_2 = -\frac{T}{I_1}t + \omega_1 - \left(\frac{T}{I_2}t + \omega_2\right) =$$

$$\omega_1 - \omega_2 - T\left(\frac{I_1+I_2}{I_1 I_2}\right)t \qquad (5)$$

The clutching operation is completed at the instant in which the two angular velocities $\dot{\theta}_1$ and $\dot{\theta}_2$ become equal. Let the time required for the entire operation be t_1. Then $\dot{\theta}= 0$ when $\dot{\theta}_1 = \dot{\theta}_2$ and so equation (5) gives the time:

$$t_1 = \frac{I_1 I_2(\omega_1-\omega_2)}{T(I_1+I_2)} \qquad (6)$$

This equation shows that the time required for the engagement operation is directly proportional to the velocity difference and inversely proportional to the torque.
We have assumed the clutch torque to be constant. Therefore, using equation (5), we find the rate of

energy-dissipation during the clutching operation to be:

$$u = T\dot{\theta} = T\left[\omega_1 - \omega_2 - T\left(\frac{I_1+I_2}{I_1 I_2}\right)t\right] \qquad (7)$$

This equation shows that the energy-dissipation rate is greater at start, when $t = 0$.
The total energy dissipated during the clutching operation is obtained by integrating equation (7) from $t = 0$ to $t = t_1$. The result is found to be:

$$E = \int_0^{t_1} u\,dt = T\int_0^{t_1}\left[\omega_1 - \omega_2 - T\left(\frac{I_1+I_2}{I_1 I_2}\right)t\right]dt = \frac{I_1 I_2(\omega_1-\omega_2)^2}{2(I_1+I_2)} \qquad (8)$$

Note that the energy dissipated is proportional to the velocity difference squared and is independent of the clutch torque.
Note that E in equation (8) is the energy lost or dissipated.

3 Problem formulation

The motion equations can be written in the differential form:

$$(M_1 - M_{a1})\cdot\omega_1(t)dt = (M_2 + M_{a2})\cdot\omega_2(t) + M_C(t)\cdot(\omega_1(t)-\omega_2(t))dt \qquad (9)$$

Or in the integral form :

$$W_1 = W_u + W_a + W_d \qquad (10)$$

Where:

$M_{a1,2} = J_{1,2}\cdot\dfrac{d\omega_{1,2}}{dt}$,dynamic friction clutch (of acceleration) ;

W_1 - the total energy ; Wu – the useful energy; Wa – the energy needed to accelerate the reduced masses at the secondary shaft ; Wd –dissipated energy through friction.
The evaluation of the dissipation percentage when engaging on both states loaded and unloaded is accomplished by the dimensionless loss coefficient:

$$\alpha_W = \frac{W_d}{W_u + W_a + W_d} \quad ; \alpha_{W0} = \frac{W_d}{W_a + W_d} \qquad (11)$$

This relation proves that the less favorable situation under the energetic balance aspect belong to the no load status (with some simplified hypothesis [1, 2, 5 and 6] for $\omega_{1,2}$ (t) and M_C (t) $\alpha_{W0} \cong 0.5$, because

$$W_d \cong W_a = \frac{1}{2}\cdot J_2 \cdot \omega_2^2$$

When engaging with load, the most common case found, there are two phases:

$$a)\forall t \in t_m; t_{a1} \cup M_c(t) \le M_2 \exists \omega_r(t) = \omega_1(t) \cup \omega_{20} = 0$$

When the entire energy taken over by the clutch is transformed into the thermic energy and dislocation energy (usage energy):

$$W_1\Big|_{t_m}^{t_{a1}} = W_d\Big|_{t_m}^{t_{a1}} = \int_{t_m}^{t_{a1}} M_C(t)\cdot\omega_r(t)dt \qquad (12)$$

$$b)\forall t \in t_m; t_a \cup M_C(t) > M_2 \exists \omega_r(t) < \omega_1(t) \cup \omega_2(t) > 0$$

that correspond to the acceleration of the secondary shaft , when from the total energy taken over by the shaft with the separation of the effects, you can distinguish the components for :

-Defeating the exterior resistances (M_2)

$$W_u\Big|_{t_{a1}}^{t_a} = \int_{t_{a1}}^{t_a} M_2\cdot\omega_2(t)dt \qquad (13)$$

-The acceleration of the reduce masses at the secondary shaft

$$W_a\Big|_{t_{a1}}^{t_a} = \int_{t_{a1}}^{t_a} M_{a2}\cdot\omega_2(t)dt = J_2\cdot\frac{\omega_2^2}{2} \qquad (14)$$

-Dissipative processes caused by friction

$$W_d\Big|_{t_{a1}}^{t_a} = \int_{t_{a1}}^{t_a} M_2\cdot\omega_r(t)dt + \int_{t_{a1}}^{t_a} M_{a2}\cdot\omega_r(t)dt \qquad (15)$$

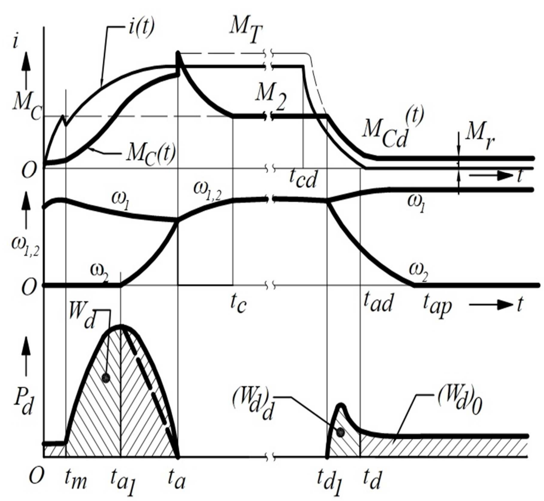

Fig. 2. Energy balance

For the whole loaded engaged process, the dissipative energy has a value (fig. 2.):

$$W_d = W_d\Big|_{t_m}^{t_{a1}} + W_d\Big|_{t_{a1}}^{t_a} = \int_{t_m}^{t_{a1}} M_C(t)\cdot\omega_r(t)dt + \int_{t_{a1}}^{t_a} M_2\cdot\omega_r(t)dt +$$

$$+\int_{t_{a1}}^{t_a} M_{a2}\cdot\omega_r(t)dt = \alpha^2\cdot M_{0\max}\cdot$$

$$\cdot\int_0^{t_a}\left\{1-\exp\left[-\frac{t-t_{m(\tau,S)}}{\beta\cdot K\cdot\eta\cdot\tau}\right]\right\}^2\cdot\omega_r(t)dt \qquad (16)$$

Dealing in a similar way, it can be determine the expression for the dissipated energy at disengaged status.

$$(W_d)_d = W_d\Big|_{t_d}^{t_a} = \int_{t_{d1}}^{t_d} M_{Cd}(t)\cdot\omega_r(t)dt =$$

$$= M_{C\max}\cdot\int_{t_{d1}}^{t_d}\left[\exp\left(-\cdot\frac{t}{\tau\cdot do}\right)\right]^2\cdot\omega_r(t)dt \qquad (17)$$

Because of the particularities in the process of engage and disengaged, you will always have :

$$W_d > (W_d)_d$$

If functioning for a long period of time in the disengaged status, there can appear an important plate package heating, due to the remaining torque.
The dissipated energy for this date can be expressed in quantity with the expression:

$$(W_d)_0 = W_d\Big|_0^{t_{ad}} = \int_0^{t_{ad}} M_r(\omega_r)\cdot\omega_r(t)dt \cong$$

$$\cong M_r\cdot\omega_r\cdot t_{ad} \qquad (18)$$

The variation of the measurements that determine the energetic balance in possible and functional regimes are shown in figure . The hatched surfaces show the dissipated energy when $\omega_r > 0$.
From the analysis of the last three equations (16),(17) and (18) , it can be seen that the heating of the clutch is intense the longer slip regime . In the case of a wrong dimensioning, the friction surfaces can get worn prematurely and even be destroyed through overheating. If the frequency of the switching between regimes is more often, the conditions become harder.

4 Problem Solution

Although the dissipative phenomenon caused by the remaining torque is less intense than the one from engage or disengaged states, in a prolonged period of time, the amount of heat developed can be higher than the one produced in acceleration and braking. Thus, this functioning regime must be accorded a special attention, because ignoring it could lead to major deficiencies.
For the quantity determination of the dissipative processes you must know: the engaging kinematics ($\omega_{1,2}(t)$,so $\omega_r(t)$), the tribological and electromagnetic behavior of the clutch $M_C(t)$,$M_{Cd}(t)$ and $M_r(\omega_r)$.
Because the possibilities for thermal charging depend o the cooling conditions, when verifying the heat calculus you must consider the work regime of the clutch.
Thus in hard coupling conditions the total dissipated energy at a single use must be compared to the admitted energy for dissipating through this inequality:

$$W_{d1} \le (W_{d1})_{adm} \qquad (19)$$

In the easy and moderate coupling conditions, the maximum dissipated energy in one hour will be compared with the admitted value :

$$W_{dh} = Z\cdot W_{d1} \le (W_{dh})_{adm} \qquad (20)$$

$$(W_d)_{oh} \le (W_{dh})_{adm} \qquad (21)$$

Where:
Z – rate of activations per hour .
The rate of activations per hour can be seen in Fig. 3 where it is shown its dependence on the mix load work of friction (in loaded and unloaded states) and the clutch dimension.

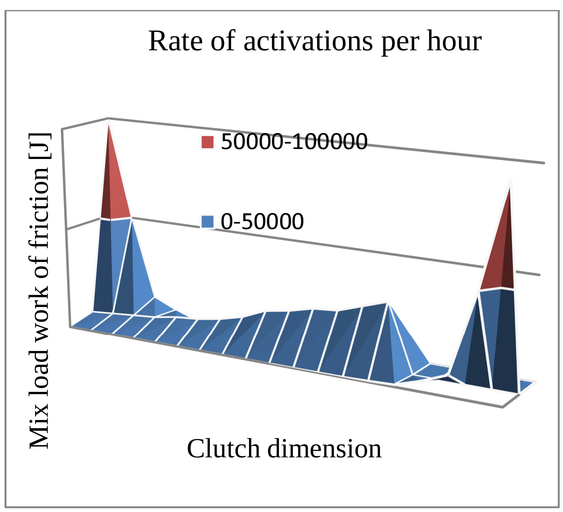

Fig. 3. Rate of activations per hour.

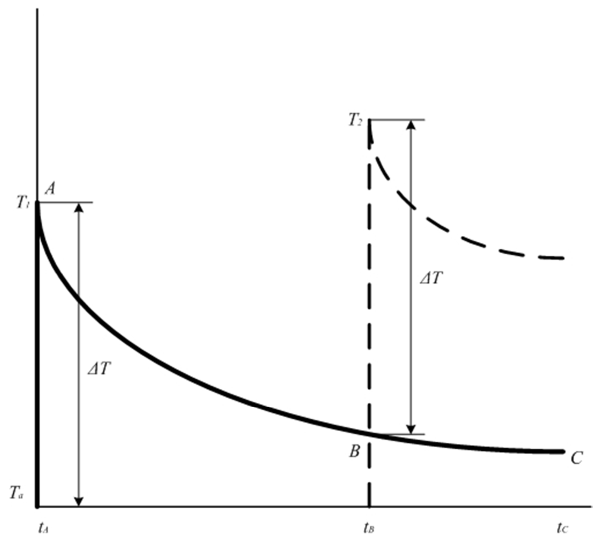

Fig. 4. The effect of clutching operations on temperature.

Figure 4 shows the effect of clutching operations on temperature.T_a is the ambient temperature. Note that the temperature rise ΔT may be different for each operation. At time t_a a clutching or braking operation causes the temperature to rise to T_1 at *A*. Though the rise occurs in a finite time interval, it is assumed to occur instantaneously. The temperature then drops along the decay line *ABC* unless interrupted by another clutching operation. If a second operation occurs at time t_b, the temperature will rise along the dashed line to T_2 and then begin an exponential drop as before.

5 Conclusion

Depending upon application, it may be desirable to have a large or small differential between the static and the dynamic torques. For instance, when engagement is made at rest(no slippage between interfaces), as would occur for a clutch-coupling or a holding brake, the static torque should be more dominant. If the clutch or brake is required to slip continuously, as in a tensioning application, very little differential is desirable to avoid a stick-slip condition.

The experimental program has a vital importance because of the big influence of the cooling way of the clutch and the environmental fluid used for this purpose. The results reveal some phenomenological particularities, offering thus useful data for the practical applications. Even smaller quantities produced in mechanical transmissions that have a large number of clutches and brakes in their build, or that have the cooling conditions difficult, must not be neglected.

6 Acknowledgment

This work was partially supported by the strategic grant POSDRU/88/1.5/S/50783, Project ID50783 (2009), co-financed by the European Social Fund – Investing in People, within the Sectoral Operational Programme Human Resources Development 2007 – 2013.

This work was partially supported by the strategic grant POSDRU 107/1.5/S/77265, inside POSDRU Romania 2007-2013 co-financed by the European Social Fund – Investing in People.

This work was partially supported by the strategic grant POSDRU/21/1.5/G/13798, inside POSDRU Romania 2007-2013, co-financed by the European Social Fund – Investing in People”.

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