The design and testing of an automatic clutch

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MS received 25 November 1985

Abstract. The design and testing of an automatic clutch working on the centrifugal principle is hereby described. Two brake shoes of a Honda 90cc motorcycle were initially converted to clutch shoes. The assembled clutch was tested using a Villiers Petrol engine, and later, a Petter Diesel engine. The load in each case was an absorption-type dynamometer. The clutch with Honda brake shoes could transmit torques over 8 Nm at speeds around 2400 RPM for the small diesel engine with fast idling speed. On using aluminium clutch shoes the performance was considerably better.

Keywords. Automatic clutch; centrifugal principle; motorcycle brake shoes; clutch shoes; absorption-type dynamometer.

1. Introduction

Many small scale industries in the rural areas of developing countries use light internal combustion engines or small independent electric plants without gear boxes or clutches. These prime movers are used for driving machines for corn milling and grinding, palm kernel cracking, air compressing, cassava processing and light concrete mixing for small block industries.

It is difficult and at times impossible to start the engines under load and even without. It is equally difficult to start the engines when they become old and thus inefficient. In the case of electric motors with current supply from independent generating plants, starting under load can cause overloading of the plant and sudden reduction in the power to electric machines which operate from the same supply. The excessive currents are necessary for accelerating not only the rotating “inertias” of the electric motor but also those of the machine and may be also to overcome a load.

Although shift-pulleys could be used to disengage the drive from the load at starting, such a design would require an operator to handle the engagements and disengagements. A shift pulley also tends to pick up the load rather impulsively and so impact loading of the bearings remains a problem.

A design with a simple automatic clutch will remove the problems posed by that with shift pulleys. Moreover, the automatic clutch is more compact and so can be more easily used in restricted places. An automatic clutch can also be more easily

employed for retrofitting existing drives which are directly coupled. In this study a simple clutch working on the centrifugal principle has been designed.

2. The centrifugal clutch

A centrifugal clutch (figure 1) works on the principle of the centrifugal force acting on a spring loaded mass, *A*, which moves radially to contact the driven drum, *B*, at a predetermined speed of the driving (input) shaft. In this design the centrifugal clutch is formed partially from the brake shoes of a Honda motorcycle. These brakes are relatively cheap and are available in fabricated form in markets in developing countries. (It should be pointed out here that other shoe materials can be used in the centrifugal design concept.)

2.1 The braking capacity of a Honda shoe

A brief analysis of the braking capacity of member, *A*, when used in a Honda motorcycle illustrates the feasibility of converting these shoes to a centrifugal clutch.

Let the total mass of a rider, his Honda (90 cc model) motorcycle and the equivalent mass of all rotating inertias which form part of the motorcycle equal 150 kg. When travelling at 20 metres per second, if the machine is braked hard enough to stop within a distance of eight metres, the average deceleration is given by the basic equation:

$$a = -U^2/2s, \quad (1)$$

where, *a* = deceleration, *U* = initial speed = 20 m/s, *S* = braking distance = 8 m, i.e., *a* = -25 m/s².

Therefore the friction force contributed by each of two shoes

$$= (150 \times 25)/2 = 1875 \text{ N.}$$

For a driving wheel of 0.5 m diameter equivalent torque exerted on the driving hub (with balancing friction on the ground)

$$= 1875 \times (0.5/2) = 468.75 \text{ Nm,}$$

per shoe. This is the braking capacity of the Honda brakes.

These brakes have aluminium alloy webs and ribs onto which Ferrodo-Asbestos linings are rivetted (figure 2). The external diameter of the lining is 108 mm, the width is 23 mm and the lining thickness is 3 mm. The initial mass of each shoe was

Figure 1. Single pivot centrifugal clutch.

0.032 kg which was increased to 0.112 kg for the purposes of the clutch design (by rivetting small steel plates onto the shoe frame).

The aluminium alloy frame has provisions for the attachment of brake springs and this feature comes in handy for spring-loading the centrifugal clutch during assembly.

2.2 The driving plate and hub

A 10 mm thick, 120 mm diameter mild steel plate is welded centrally to a 42 mm diameter, 32 mm long mild steel hub to form the driving plate and hub assembly (figure 2). The hub has an internal key-way with which it is connected to an engine output shaft. Two pins, 9 mm diameter and 40 mm long, are diametrically fixed on the driving plate on either side of the centre. The two driving pins form the anchoring positions for the two driven clutch shoes.

There is, also, a centring pin protruding from and integral with the driving plate in a position opposite the hub.

2.3 The driven drum and hub

The drum made of mild steel has 120 mm external diameter and 108 mm internal diameter (to register with the clutch shoes during full surface contact) (figure 2). The internal width of the drum is 35 mm and the driven (output) hub, 42 mm external diameter and 40 mm long, is integral with the drum. Also protruding from the drum towards the centring pin from the driving plate is a bush bearing which locates the centring pin. The driven hub (output) also possesses a key-way which will receive the driven machine's input shaft. The periphery of the drum is machined to register with a groove on the driven plate to provide sealing against particle contamination of the interior of the clutch assembly.

Figure 2. The clutch components.

At engine idling speed, ω_0 , the driven member is stationary and so

$$T_F = 0.$$

Therefore,

$$F_s = \frac{1}{2}(M\omega_0^2 r_g). \quad (7)$$

Equation (7) gives the value of the required spring load.

At the engine running speed, ω , the clutch is engaged and the torque,

$$T_F = \{2(a+b)rd/[(a+b)-\mu r_d]\} (M\omega^2 r_g - 2F_s),$$

$$\text{i.e. } T_F = \{(2\mu M)(a+b)r_g rd/[(a+b)-\mu r_d]\} (\omega^2 - \omega_0^2). \quad (8)$$

Equation (8) gives the torque capacity of the clutch assembly for a range of values of engine running speed, ω .

At idling speed, $\omega = \omega_0$ and $T_F = 0$, and so, the clutch should not engage before the idling speed is exceeded. Equation (8) also indicates that the torque, T_F , is proportional to the value of shoe mass, M , the choice of which is guided by:

- (i) the maximum machine torque expected in service,
- (ii) the maximum pressure at which the material of the shoe linings would not be crushed

The values of the constants in the design are:

$$a = 30 \text{ mm}, b = 15 \text{ mm}, r_g = 39 \text{ mm}, r_d = 54 \text{ mm}, \mu = 0.3.$$

Hence the general clutch equation from (8) becomes

$$T_F = 22.22 \times 10^{-4} (\omega^2 - \omega_0^2). \quad (9)$$

4. The combined engine and clutch characteristic curves

For different corresponding values of engine running and idling speeds, and ω_0 , respectively, a plot of the centrifugal clutch characteristics, T_F , (9), is made. To find the speed at which the clutch would bring a driven machine into full engagement using a specified engine, the characteristic torque curves of the engine and the automatic clutch are superimposed (figure 4). The crossing point between the two curves gives the operational torque available to the driven machine (e.g. a corn grinder) and the speed at which full engagement is made (George 1970).

In the example, the characteristic torque curve of a small Petter Diesel engine (torque curves are usually supplied with new engines at purchase) is superimposed with the characteristic torque curve of the centrifugal clutch designed in this study. From the curve, slip stops and full engagement takes place at 2020 rpm, and the clutch slips for all speeds less than 2020 rpm but is fully engaged for all speeds greater than this value, the design being for fast prime movers.

5. Conclusions from tests

The equipment representing a driven machine in the tests is an absorption-type dynamometer which is equipped with a tachometer. Two engines, one, a 2.5 kW

Figure 4. The clutch and engine torque outputs.

Petter Diesel and the other, a 2 kW Villiers Petrol, were individually used for the tests. Because of its rather very low speed range (320 rpm idling and 510 rpm maximum speeds) the Villiers Petrol engine requires a different class of low speed clutches with low-stiffness springs. The Petter Diesel gave very good results, though its fast idling speed (1000 rpm) requires stiffer spring loading.

At a speed of 2400 rpm the torque transmitted to the dynamometer was around 9 Nm which compares reasonably with the expected value (figure 4).

For a commercial production of these centrifugal clutches, two varieties therefore would be specified, one for low speed engines (usually petrol) and the other for higher speed engines (usually the diesel and electric motors), spring stiffnesses being the main difference.

It is also necessary to provide some means of cooling the clutch to dissipate the heat generated during the occasional engagement/disengagement slip periods using fast engines.

When these measures are taken this design offers a successful solution to the problem of starting prime movers under load especially those used in the cottage industries.

The author wishes to express his gratitude to Prof. G O Ezekwe of PRODA, Enugu, Nigeria, for help rendered in the course of this study.

References

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