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THE DUAL MASS FLYWHEEL

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Introduction

As shown in the previous presentation, it is almost always possible to eliminate gear rattle occurring in neutral idle and coast mode using a properly tuned clutch disc. We can successfully keep the natural frequency below the idle speed by using a very flat damper characteristic curve. The low vibration excitation in coast mode can generally be damped.

Matters are different for strong drive mode excitation, where conventional dampers are often inadequate for satisfactory vibration damping or for reducing the natural frequency below the operating speed.

Theoretically, a torsion damper rate of less than 1 Nm/degree would push the natural frequency out of the operating speed range of the vehicle. However, we can't implement this in practice, because the clutch package space will not accommodate the necessary damper size. Besides, a torsion damper of this size on the transmission input shaft would create a mass moment of inertia that would make it difficult to synchronize the transmission.

Consequently we have to find another way to reduce the natural frequency of the transmission below the vehicle operating speed.

Resonance Displacement by an Additional Inertia on the Transmission Input Shaft

The drawing at the top of Figure 1 shows a simplified vibration model. It consists of:

- J_1 The mass moment of inertia of the engine with the flywheel and the clutch
- J_2 The mass moment of inertia of the rotating parts of the transmission
- J_3 The mass moment of inertia of the vehicle
- C_1 The torsion rate of the torsion damper
- C_2 The torsion rate of the rest of the drive train

The size of the respective circles corresponds to the mass moments of inertia: the extremely low inertia of the transmission is clearly evident at once.

Without presenting the exact formulaic relationship for the resonance, the elementary formula $\omega = \sqrt{C/J}$ for a single mass oscillator should suffice to illustrate the fact that increasing the mass moment of inertia J will cause the natural frequency to decrease if the torsion rate C cannot be decreased any further.

Therefore if we could sufficiently increase the very low effective mass moment of inertia of the transmission J_2 by adding inertia, we would be closer to solving the problem. This is only true if we wouldn't have to synchronize this inertia when shifting gears (see the bottom drawing in Figure 1).

In a practical example, Figure 2 shows the effect of increasing the mass moment of inertia J_2 of the transmission. Engine speed fluctuations (solid line) and those of the transmission input shaft (broken line) are plotted vs. the engine speed.

One can see that a mass moment of inertia J_2 of about 0.3 kg m^2 decreases the natural frequency to an engine speed of 500 rpm, although the resonance amplitude rises significantly. This is not generally a problem because the resonance is below the operating speed.

Figure 3 again shows for different damper torsion rates how the natural frequency decreases with increasing mass moments of inertia. According to this pattern, we would be able to achieve resonance-free operation above approximately 600 rpm using a torsion rate of C_1 = 15 Nm/degree and an inertia J_2 = 0.1 kg m^2. In order to do this, the mass moment of inertia J_2 for the transmission will have to be increased 20 to 30 times the value of conventional solutions.

Schematic Design of the Dual Mass Flywheel

Figure 4 compares the conventional clutch system design with an additional inertia (top drawing) to a new kind of design which takes into consideration the information cited above and leads us to the dual mass flywheel.

The flywheel is split into two sections. The primary flywheel is attached to the crank shaft. A ball-bearing is used to pilot the secondary flywheel on the primary flywheel. The secondary flywheel is coupled to the primary flywheel via a vibration damper. The clutch is mounted on the secondary flywheel in the usual fashion. Since there is a vibration damper located between the flywheels, a solid clutch disc is adequate for load transmission. As a result of this configuration, the additional inertia doesn't have to be synchronized.

The dual mass flywheel (DMFW) is the ideal solution to the theoretical problem of attaching an inertia to the transmission input shaft without compromising synchronization. Synchronization is actually improved considerably: because the torsion vibration damper has been moved out of the clutch disk into the dual mass flywheel, the clutch disk has a lower mass moment of inertia.

Basic Guidelines for Designing Dual Mass Flywheels

First of all there is the obvious requirement that the sum of the mass moments of inertia of the two flywheels be equal to the mass moment of inertia of the conventional flywheel.

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Figure 5 shows the optimum inertia distribution under this theoretical condition. Here the natural frequency is shown dependent upon the ratio of the mass moment of inertia J_1, which consists of the primary flywheel and the engine, and the mass moment of inertia J_2, which consists of the secondary flywheel, the clutch, the clutch disc and the rotating parts of the transmission.

This is shown for all gears because the parameters of the vibration system change as a result of different transmission ratios.

The lowest natural frequencies are achieved for an inertia ratio between 0.5 and 1, which means that the engine together with the primary flywheel should, if possible, have less mass moment of inertia than the total inertia of the secondary flywheel, the clutch, the clutch disc and the transmission.

The lower primary flywheel inertia increases engine irregularity because the irregularity is limited only by the inertia connected directly with the crank shaft. This is of almost no importance for the dual mass flywheel, but it could lead to damage of the crank shaft or accessory drives. Consequently we sometimes have to deviate upward from the optimum J_1/J_2 ratio.

This graph also clearly illustrates how strongly the natural frequency increases in 5th gear. This also explains why rattle problems have increased so significantly with the general introduction of the 5-speed transmission.

In Figure 6 we can read out the required total mass moments of inertia and the torsion damper torque rates for a natural frequency of 500 rpm and an inertia ratio of $J_1/J_2 = 1$.

At identical torque rates, the 4-cyl. engine requires the highest mass moment of inertia, and the 8-cyl. engine the lowest. It is relatively easy to control the 8- and even the 6-cyl. engines. On the other hand, the 4-cyl. engine requires very low torsion rates and/or high inertias in order to achieve vibration isolation that is as effective as that of the 6-cyl. engine. Consequently we can expect greater design problems with the 4-cyl. engine. In contrast to this, the mass moment of inertia of a dual mass flywheel for an 8-cyl. engine can be less than that of a conventional system.

Vibration Isolation with the Dual Mass Flywheel

Figure 7 shows vibration measurements for a 6-cyl. engine in drive mode with wide open throttle at 800 rpm. We have compared a well-tuned conventional damper (top graph) to a dual mass flywheel (bottom graph). While the torsional vibrations of both the crank shaft (solid line) and the transmission input shaft (broken line) are almost identical with the conventional damper, the dual mass flywheel exhibits excellent vibration isolation because the natural frequency is significantly lower than the operating speed.

As explained previously, engine irregularity is greater with the dual mass flywheel because the mass moment of inertia of the primary flywheel is less than that of the conventional flywheel along with the clutch.

With the dual mass flywheel, engine irregularity is controlled only by the primary flywheel. The secondary flywheel doesn't have a stabilizing influence on engine speed fluctuations because of the good vibrational isolation. The maximum acceptable irregularity of the engine and any accessory drives determines the size of the primary inertia.

With conventional torsion dampers, a larger than necessary flywheel is used in order to keep transmission noises within acceptable limits by reducing engine irregularity. This is not required with the dual mass flywheel. Provided that we observe the limitations cited above, we can choose a primary flywheel mass that is smaller than the conventional flywheel mass.

Figure 8 compares calculation and measurement of vibrations with a dual mass flywheel in neutral idle for another vehicle. The top graph shows the vibration calculation, the bottom one the vibration measurement. In both graphs, the engine is represented by the solid line and the transmission input shaft by the broken line. Measurement and calculation coincide well and again show the excellent vibration isolation of a dual mass flywheel, even in neutral idle.

Passing Through the Resonance Point

If we use the dual mass flywheel and choose the proper inertia and damper torsion rate, the natural frequency will be below the normal operating speed (Figure 9); however, the speed of the engine must pass through the resonance point while the engine is started up and shut off. This can also happen when the vehicle is driven at drastically low speeds.

Resonance magnification and associated torque peaks vary in direct proportion to the mass moments of inertia of the vibrating masses. Consequently, with the dual mass flywheel resonance will be significantly stronger while the engine is started up or shut off than it is with a conventional torsion damper. We have to take measures to ensure that this resonance does not have any damaging effects.

Therefore, as you can see in Figure 10, we install an additional damper device in series with the actual torsion damper. The actual torsion damper is called the inner damper because of its design configuration. The purpose of the additional damper is to consume much of the vibration energy during high torque peaks. This damper device can take the form of either an additional torsion damper with high friction damping capability, called the outer damper, or it can be a slip clutch controlled by centrifugal force.

The engine, with the primary flywheel, passes the vibration excitation on to the outer damper or to the centrifugal clutch, respectively. Under normal load these components remain rigid because of their high inherent friction, and the vibrations reach the inner damper unchanged. The outer damper or the centrifugal clutch takes effect only at very high torque peaks, that is, when passing through the resonance point.

Optimum friction for the outer damper is between 100 Nm and 300 Nm. With the centrifugal clutch the friction torque must, of course, always be reliably higher than the engine torque.

Figure 11 shows the slip torque curve for a centrifugal clutch vs. engine speed. The engine torque curve is also plotted on the graph. The slip torque of such a clutch is dependent upon centrifugal force, so that at high speeds, where maximum engine torque occurs, the centrifugal force ensures reliable torque transmission. On the other hand, brief slippage at very low speeds and relatively low torque peaks provides ideal resonance suppression.

A standard slip clutch has a constant slip torque over the entire engine speed range. This constant torque would have to be greater than the maximum engine torque. Hence it would be considerably less effective against resonance. This disadvantage is all the more serious since dry friction varies considerably in practical operation. This means that the slip torque has to be far higher than the engine torque to provide sufficient safety.

The top graph in Figure 12 shows vibration measurements of a dual mass flywheel without an outer damper at a speed of 400 rpm, which is just above the engine stall speed. The primary inertia and especially the secondary inertia exhibit very high vibration amplitudes. These are unacceptable for various reasons.

A well-tuned outer damper is capable of reducing excessive amplitudes to acceptable values (bottom of Figure 12). The vibration amplitudes of the engine and particularly those of the transmission have been reduced considerably.

Figure 13 shows the measurement for an engine start-up cycle. Without an outer damper (top graph) resonance occurs, with the associated high amplitudes and loud noises. With the outer damper (bottom graph) the resonance is suppressed to a great extent, and the noise level corresponds to a start-up cycle with a conventional flywheel.

The top graph in Figure 14 shows an engine shut-off cycle with a short rattle interval. In contrast to the start-up cycle, the engine shut-off cycle is a more difficult problem to solve, because there are practically no extraneous noises. At vibration amplitudes with torque peaks lower than the response threshold of either an outer damper or a centrifugal clutch, gear rattle can occur in the transmission.

To further reduce the relatively low vibration amplitudes, we can install an additional speed-dependent friction device parallel to the inner torsion damper. This additional friction only takes effect below idle speed during the engine shut-off cycle. The bottom graph shows the vibration curve measured with the additional friction device. The curve shows clearly reduced vibration amplitudes, eliminating much of the gear rattle during engine shut off.

Dual Mass Flywheel Design

The following discussion illustrates the design features we use to achieve our theoretical goals.

First I would like to illustrate the basic design of the dual mass flywheel (Figure 15).

The most important flywheel components are:

- a first inertia (primary flywheel) bolted directly to the crank shaft. The starter gear is attached to this flywheel.

- an additional inertia (secondary flywheel), piloted by a ball-bearing on the primary flywheel. The conventional clutch is bolted to the secondary flywheel.

- a torsion damper system connecting the primary and secondary flywheel and providing for rotational elasticity.

As shown in Figure 16, engine torque is transmitted from the crank shaft to the primary flywheel and from there via the spacer bolts to the side plates in the outer damper.

The torque is passed on via the friction device [not shown] and the coil springs in the outer torsion damper to the flange placed between the side plates. From the flange the torque passes via the coil springs and the basic friction device [not shown] in the inner damper to the inner side plates and on via the spacer rivets to the secondary flywheel. From the secondary flywheel the torque is transmitted to the clutch, clutch disc and transmission input shaft in the familiar fashion.

Figure 17 shows one of LuK's production dual mass flywheel designs.

We retained the current pattern to bolt the primary flywheel to the crank shaft. We manufacture the primary flywheel from nodular iron because the narrow package space requires a slim cross-section. A specially developed bearing assembly makes it possible to mount the secondary flywheel directly on the primary flywheel. In spite of the unfavorable load situation (non-rotating motion) we don't have any problems with the bearing. If designed properly, the bearing assembly can even take the high ambient temperatures generated by the clutch friction work.

Acceptable bearing function relies on the following factors:

- sufficient cooling and/or thermal insulation
- a temperature-stable bearing grease
- exactly tuned mating tolerances.

In this case two serial torsion dampers provide the elastic connection between the primary and secondary flywheel. Both torsion dampers are similar to the damper in a conventional clutch disc. The coil springs are guided by windows in the side plates and the flange. Since the damper is always functioning, that is, it is always carrying out small relative motions, these guides must be very carefully designed in order to prevent premature wear of the coil springs and steel plates.

The case-hardened side plates for the outer damper are connected to the primary flywheel with spacer rivets. The rotational angle of the flange is limited with respect to the side plates by the spacer rivets which stop on the flange windows.

Friction damping in the outer damper is provided by friction segments bonded to the flange. We use a diaphragm spring to clamp the flange between the outer side plates.

The side plates for the inner damper, which are also case-hardened, are connected to the secondary flywheel with spacer rivets.

The hardened side plates and the tempered flange must be manufactured flat to guide the coil springs satisfactorily and to avoid parasitic friction in the narrow package space.

In order to generate the low basic friction, special diaphragm springs press caps made from temperature-resistant plastic against the front faces of the inner ring of the bearing.

An additional friction plate comes into play dependent upon the torque transmitted. This so-called friction control plate is required to damp high vibration amplitudes.

The friction control plate is controlled by the rivet head of the spacer rivets on the secondary flywheel side after a pre-set free angle has been bridged. The friction control plate is manufactured from plastic to prevent pin contact noises.

The secondary flywheel is also manufactured from nodular iron in order to satisfy the slim design requirements dictated by the package space. It carries an extremely flat conventional clutch. There is no torsion damper in the clutch disc with which it is paired.

As explained above, it is also possible to use a centrifugal clutch instead of the outer torsion damper to consume energy when resonance occurs. Figure 18 shows the principle of a dual mass flywheel with a centrifugal clutch. The engine torque is transmitted from the primary flywheel via the centrifugal clutch to the flange. The flange in turn carries the torque on to the inner damper, which performs the actual vibration isolation just the same way as shown above for the series damper design. The slip torque of this clutch is controlled by a diaphragm spring whose fingers increase the clamp load proportionate to the centrifugal force and quadratic to the speed. Thus, in spite of the wide variation of the coefficient of friction, we can keep the slip torque in the desired range closer to the engine torque.

Torsional Characteristic Curves of the Dual Mass Flywheel

The combined wind-up curve for the dual mass flywheel I have introduced here is relatively complicated because of the two serial dampers. Therefore we will talk about the individual characteristic curves first.

Figure 19 shows a typical wind-up curve for an inner damper with an idle stage and a main stage, but without friction control plate. Since a damper's vibration isolation properties are in direct proportion to the extent to which we can reduce the natural frequency below the operating speed, we frequently push the natural frequency far below the neutral idle speed using a very flat neutral idle stage.

For this purpose we need a torque rate of about 1 Nm/degree to 6 Nm/degree. A conventional clutch disc usually has a torque rate of about 0.2 Nm/degree.

With dual mass flywheels we can use torque rates that are about ten times as high as those we need for neutral idle dampers in clutch discs; the torsional angles are correspondingly shorter as well.

Consequently we can improve tip-in, tip-out clunk by introducing the dual mass flywheel in place of the clutch disc with a neutral idle damper. As already described, the basic friction of the inner torsion damper is relatively low. It remains constant over the entire rotational angle of the inner damper.

Figure 20 shows the wind-up curve of the outer torsion damper. As a result of the high friction, the damper is only activated at high torque amplitudes, such as occur when the engine is started-up.

In order to consume as much energy as possible, the coil springs don't engage until after a long travel between the flange and the side plates. The torque rate is tuned so that we can avoid torque peaks while passing through the resonance.

Figure 21 shows the overlayed total wind-up curve for the dual mass flywheel. Different line types have been used to distinguish better between the different partial wind-up curves.

Only the inner damper operates at low torque fluctuations, as indicated by the dash-dot curve. The outer damper comes into play during sudden, strong torque changes and/or when resonance occurs during engine start-up or shut-off (broken line). The damper capacity of both dampers is greater than the engine torque.

The solid line represents the combined wind-up curve. The system only passes through the entire curve during extreme torque peaks, such as when the engine is started.

Summary

Advantages and Disadvantages of the Dual Mass Flywheel

Starting from the conventional clutch disc damper design, I have explained how the dual mass flywheel can be used to achieve significant improvements in transmission noise performance. I have discussed theoretical influencing variables and presented ideas for optimization which should be taken into consideration in designing dual mass flywheels.

Using existing dual mass flywheels, I have introduced design considerations and explained how the dual mass flywheel works. Special attention was paid to resonance problem which occur when the engine is started or shut off, and possibilities for solving this problem were noted.

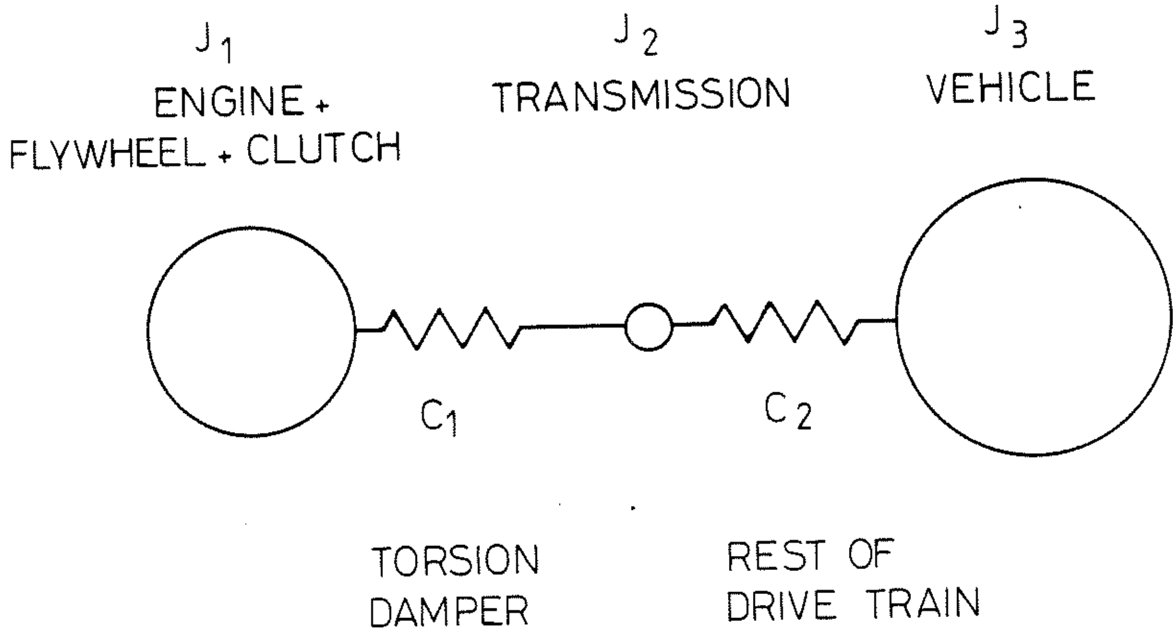
Based on several years' experience we are convinced that the introduction of the dual mass flywheel can be used to eliminate existing transmission rattle problems, in spite of the need to optimize the design.

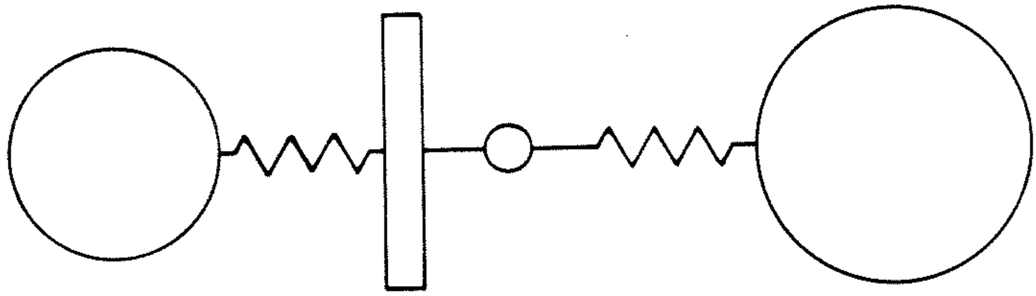
In contrast to fluid couplings, these improvements can be made without significant loss in efficiency. On the contrary, it appears that fuel consumption will be improved because it will be more comfortable to drive at lower engine speeds.

In addition our experience shows that other noise problems can be eliminated, such as body boom excited by torsional vibrations at low engine speeds.

At the moment we must weigh these advantages against relatively high costs and additional space requirements.

Initial understanding of the design also leads us to believe that the introduction of the dual mass flywheel would help to reduce the "sophistication" of the transmission, improve shift-ability and increase transmission service life.

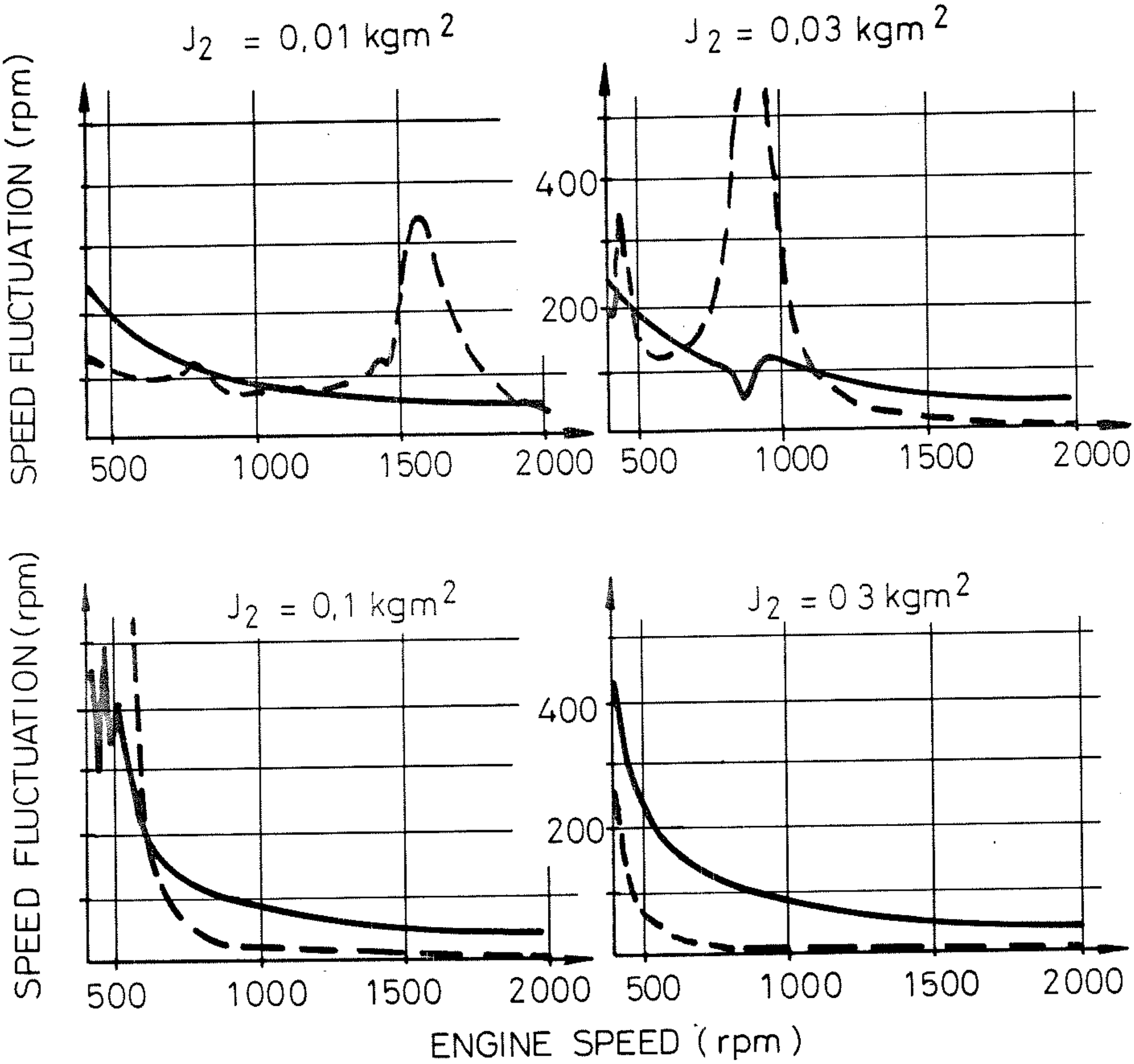




ADDITIONAL INERTIA
ON TRANSMISSION
INPUT SHAFT

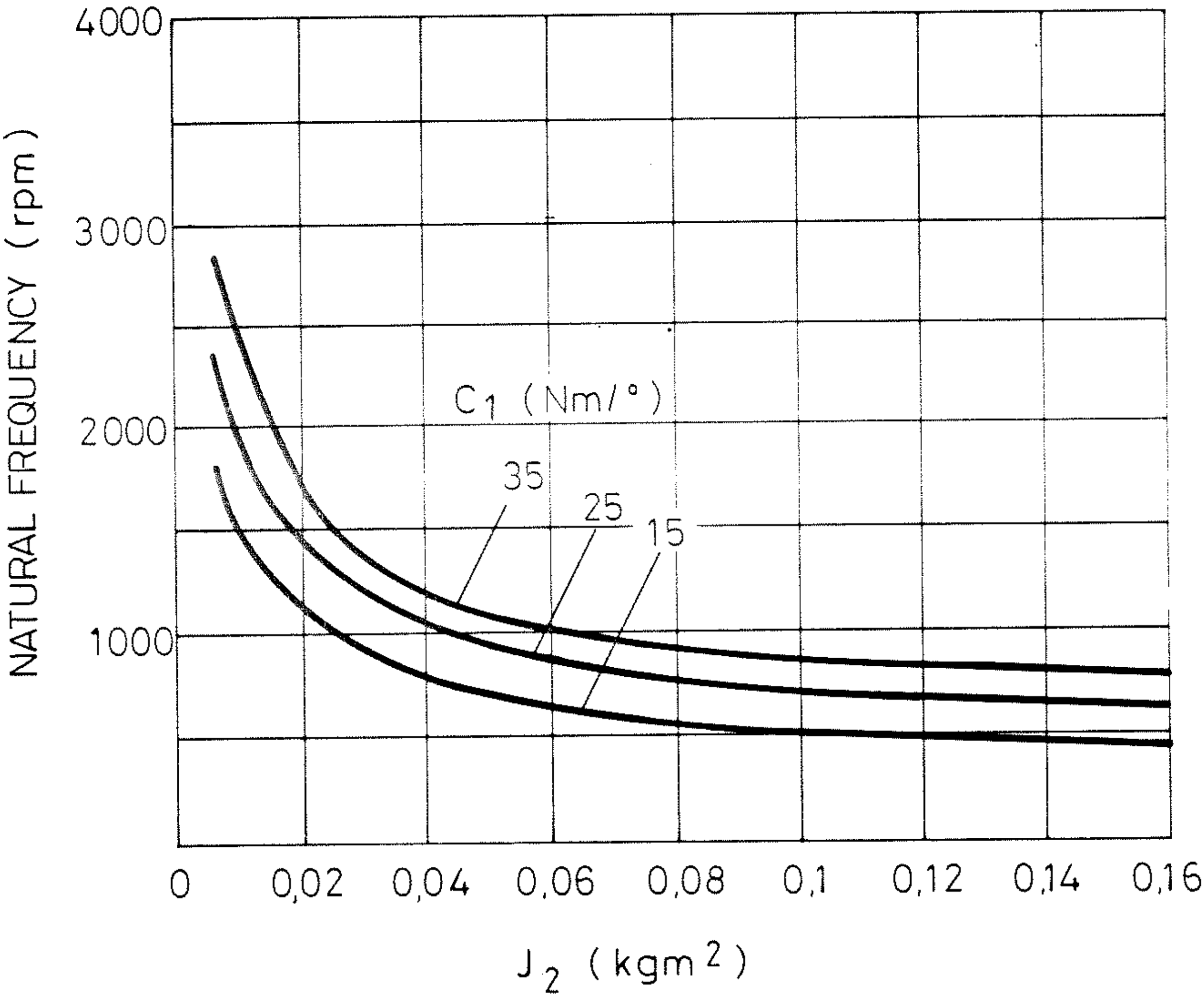
ELEMENTARY FORMULA FOR THE NATURAL FREQUENCY $\omega = \sqrt{\frac{C}{J}}$

CHANGES OF TRANSMISSION INERTIA



J_2 = TRANSMISSION INERTIA + ADDITIONAL INERTIA

02 02 86	EFFECT OF THE TRANSMISSION INERTIA ON THE NATURAL FREQUENCY	LUK

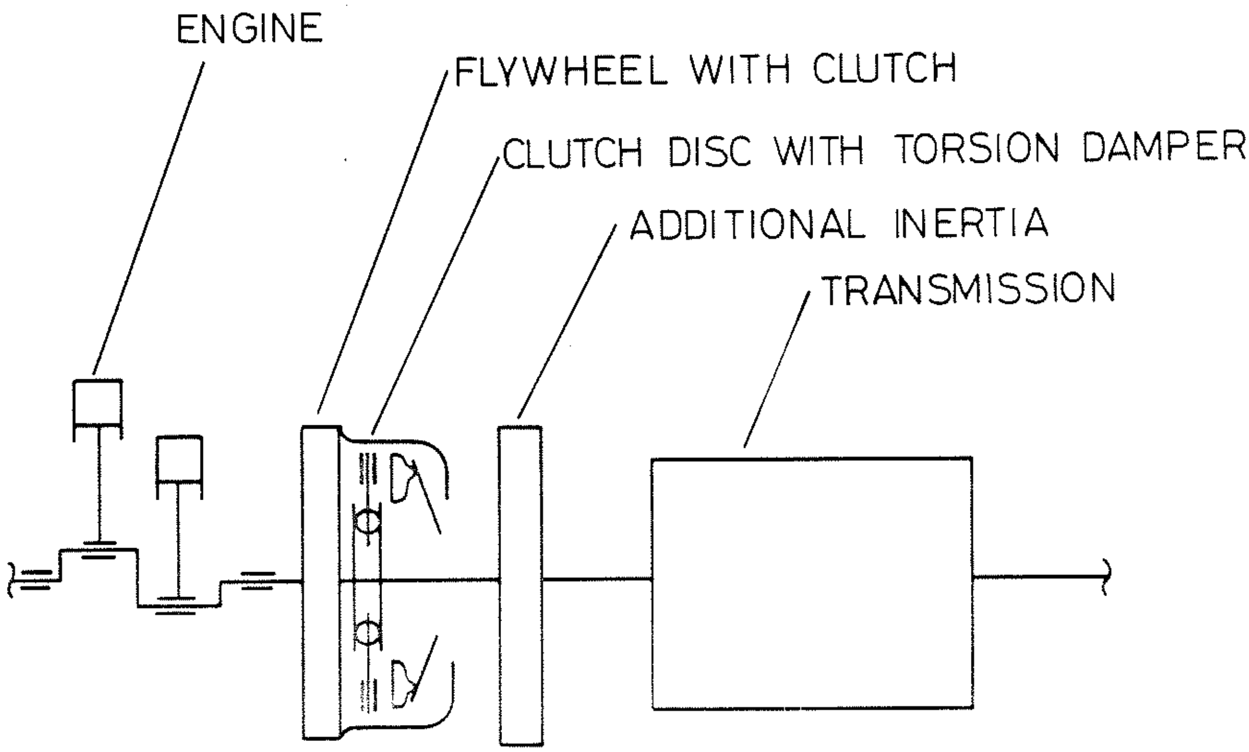


C_1 = DAMPER TORSION RATE

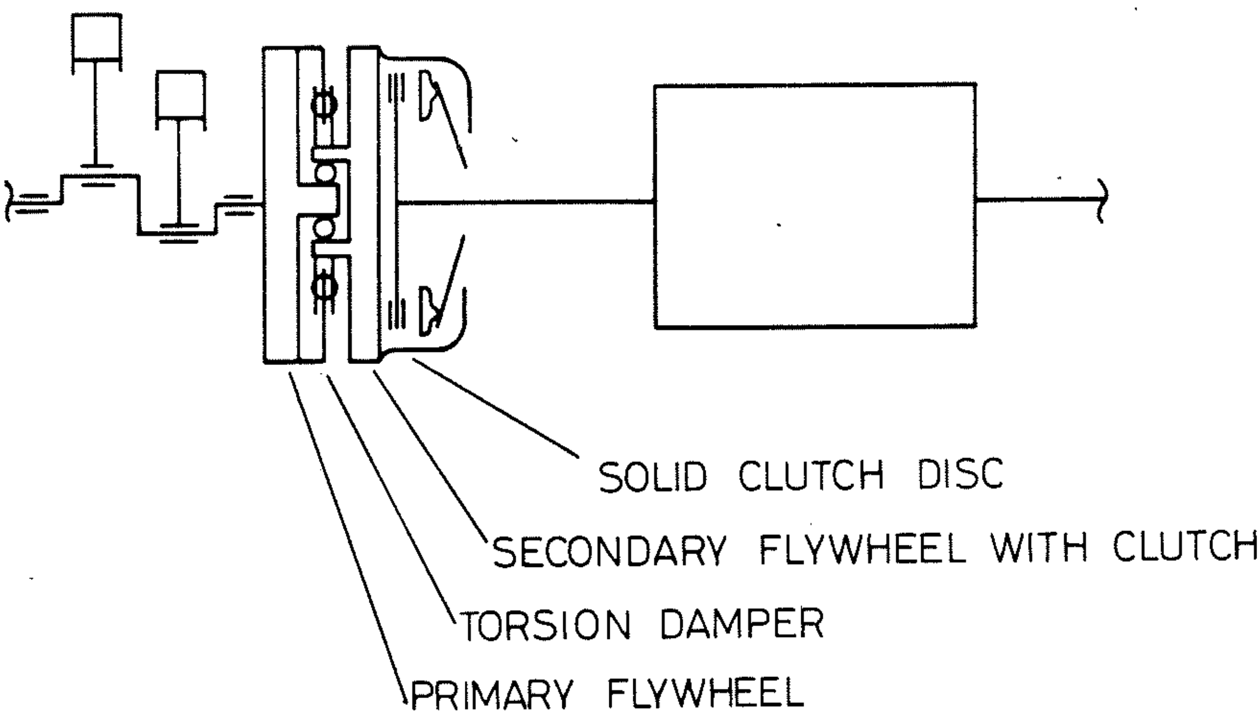
J_2 = TRANSMISSION INERTIA + ADDITIONAL INERTIA

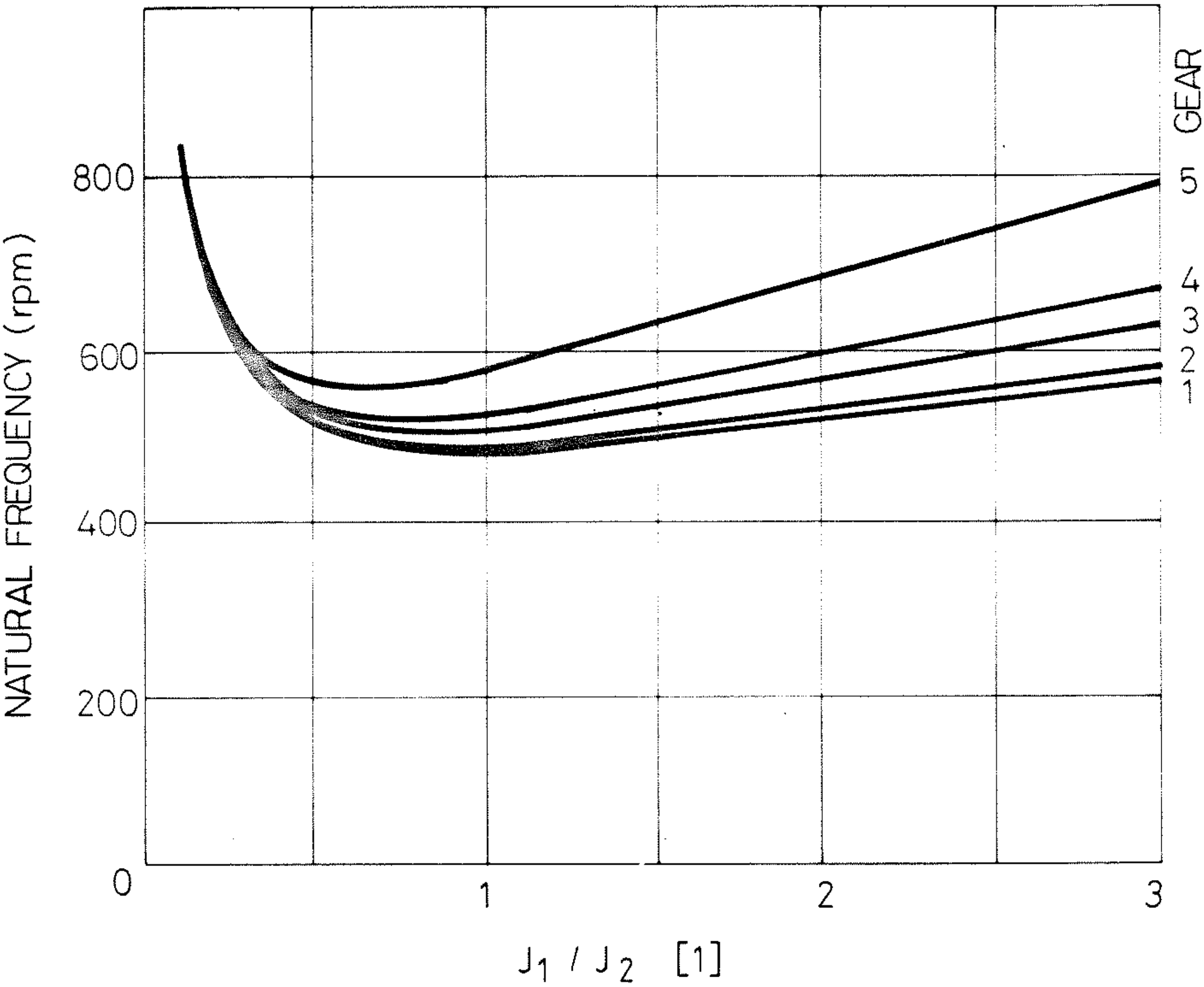
03 02 86	NATURAL FREQUENCY DEPENDENT UPON THE TORSION RATE AND THE INERTIA	LUK

ADDITIONAL INERTIA ON THE TRANSMISSION INPUT SHAFT



DUAL MASS FLYWHEEL





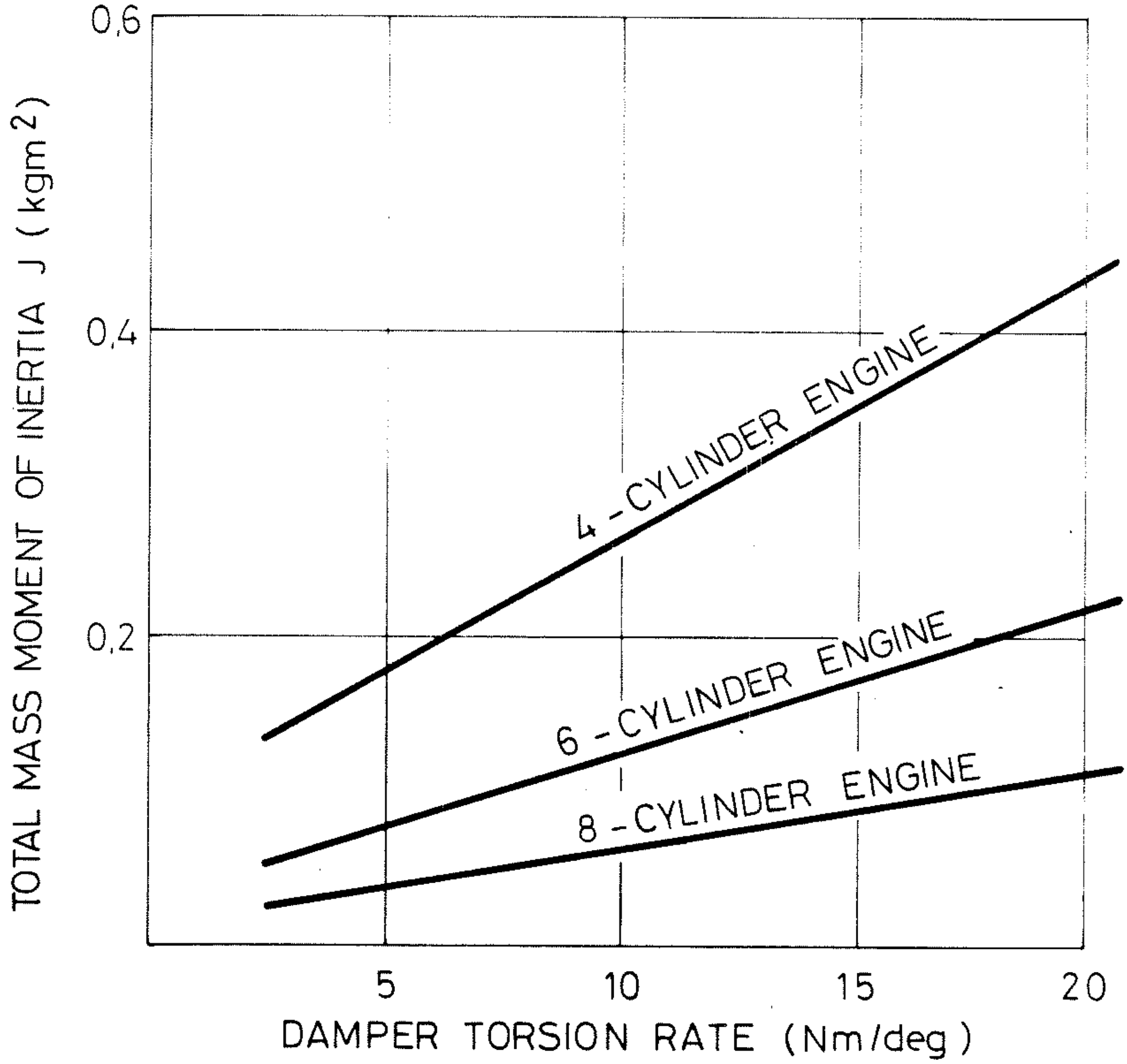
J_1 = MASS MOMENT OF INERTIA OF THE COMPONENTS AHEAD OF THE TORSION DAMPER

J_2 = MASS MOMENT OF INERTIA OF THE TRANSMISSION + ADDITIONAL INERTIA

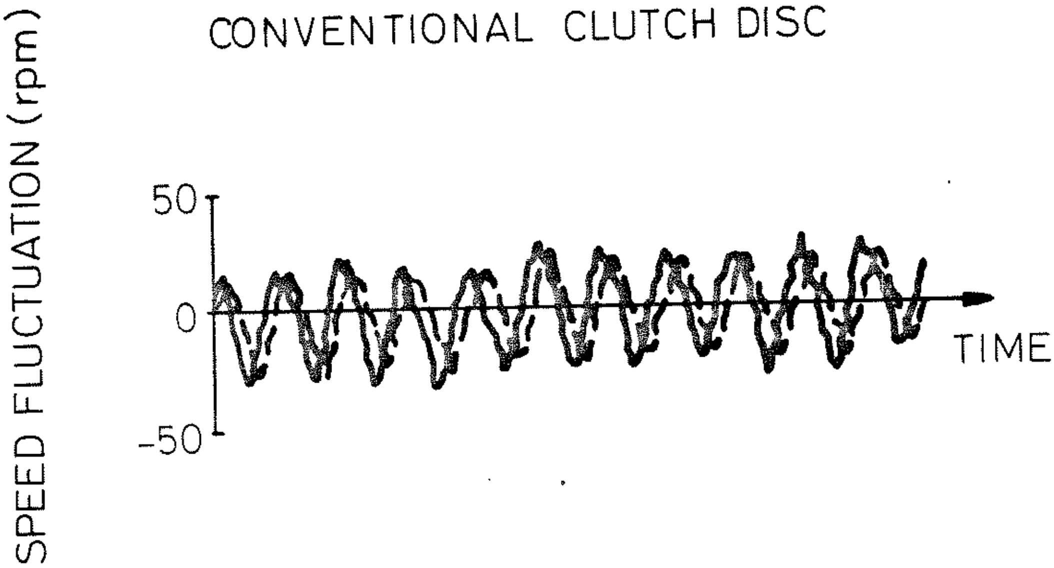
$J_1 + J_2$ = CONSTANT

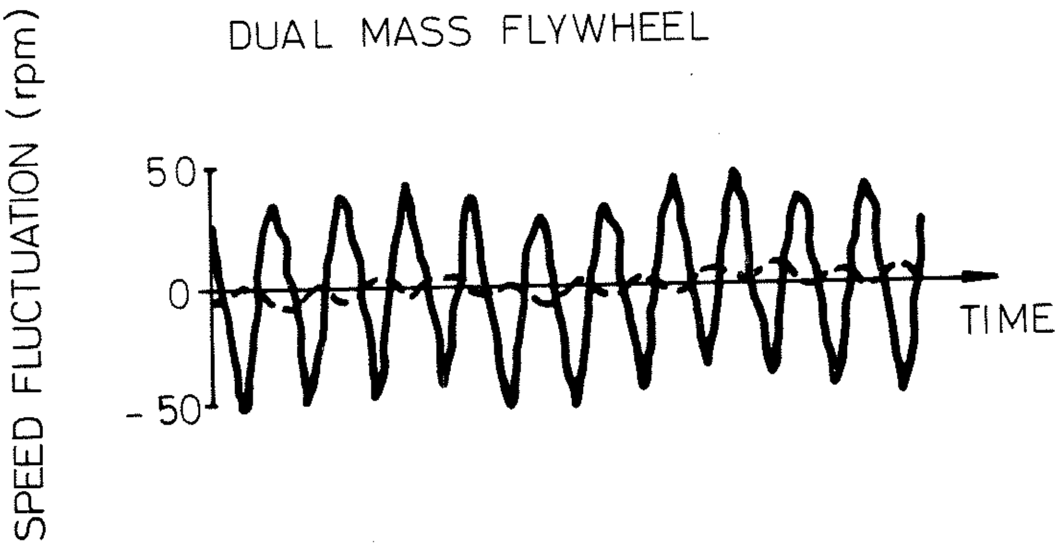
05 02 86	EFFECT OF INERTIA DISTRIBUTION ON THE NATURAL FREQUENCY	LUK

REQUIRED TOTAL MASS MOMENT OF INERTIA $J = J_1 + J_2$
FOR NATURAL FREQUENCY OF 500 RPM WITH $J_1 / J_2 = 1$
(4 TH GEAR WITH DRIVE LINE TORSION RATE C_2 = 10 Nm/deg.)



06 02 86	TOTAL REQUIRED MASS MOMENT OF INERTIA	LUK

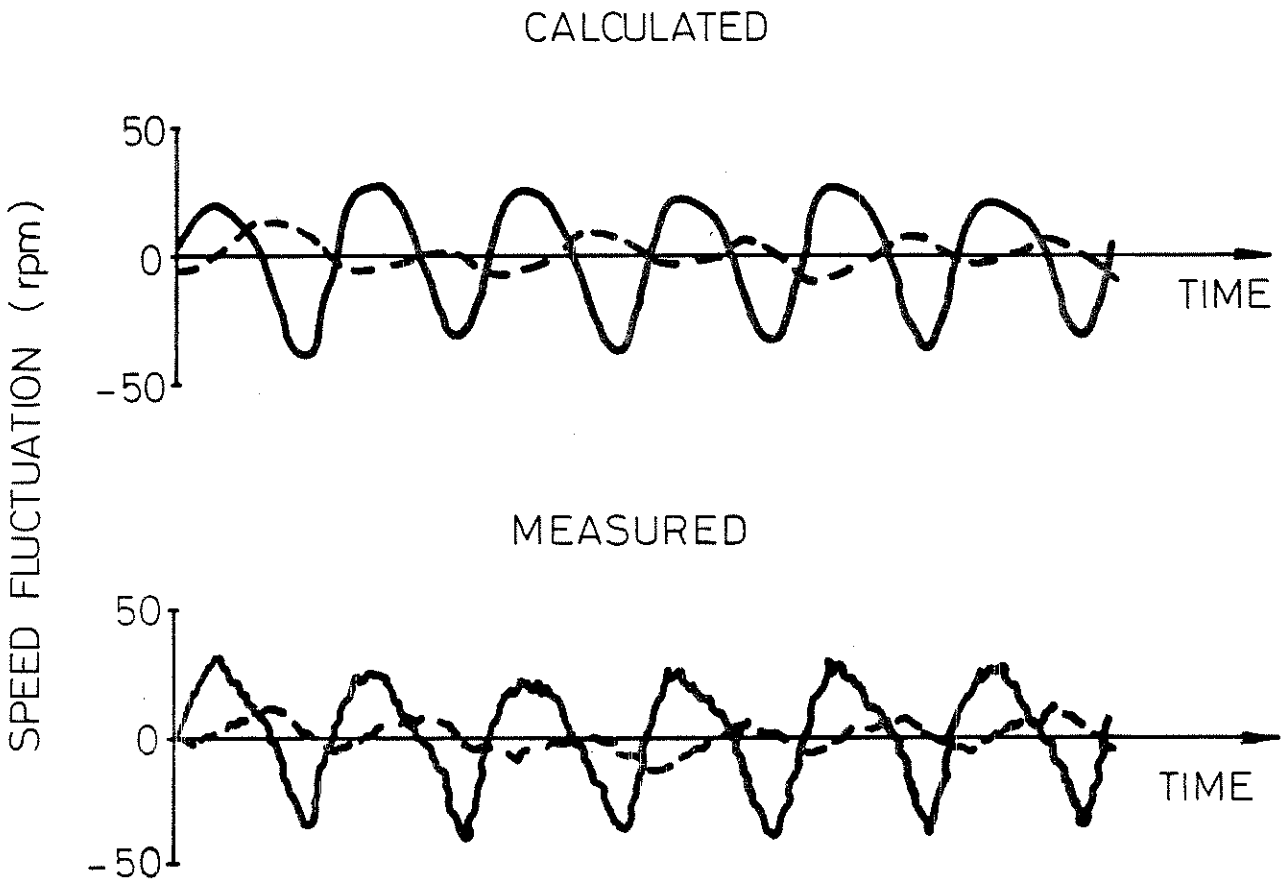




ENGINE SPEED 800 RPM

——— ENGINE
- - - - TRANSMISSION

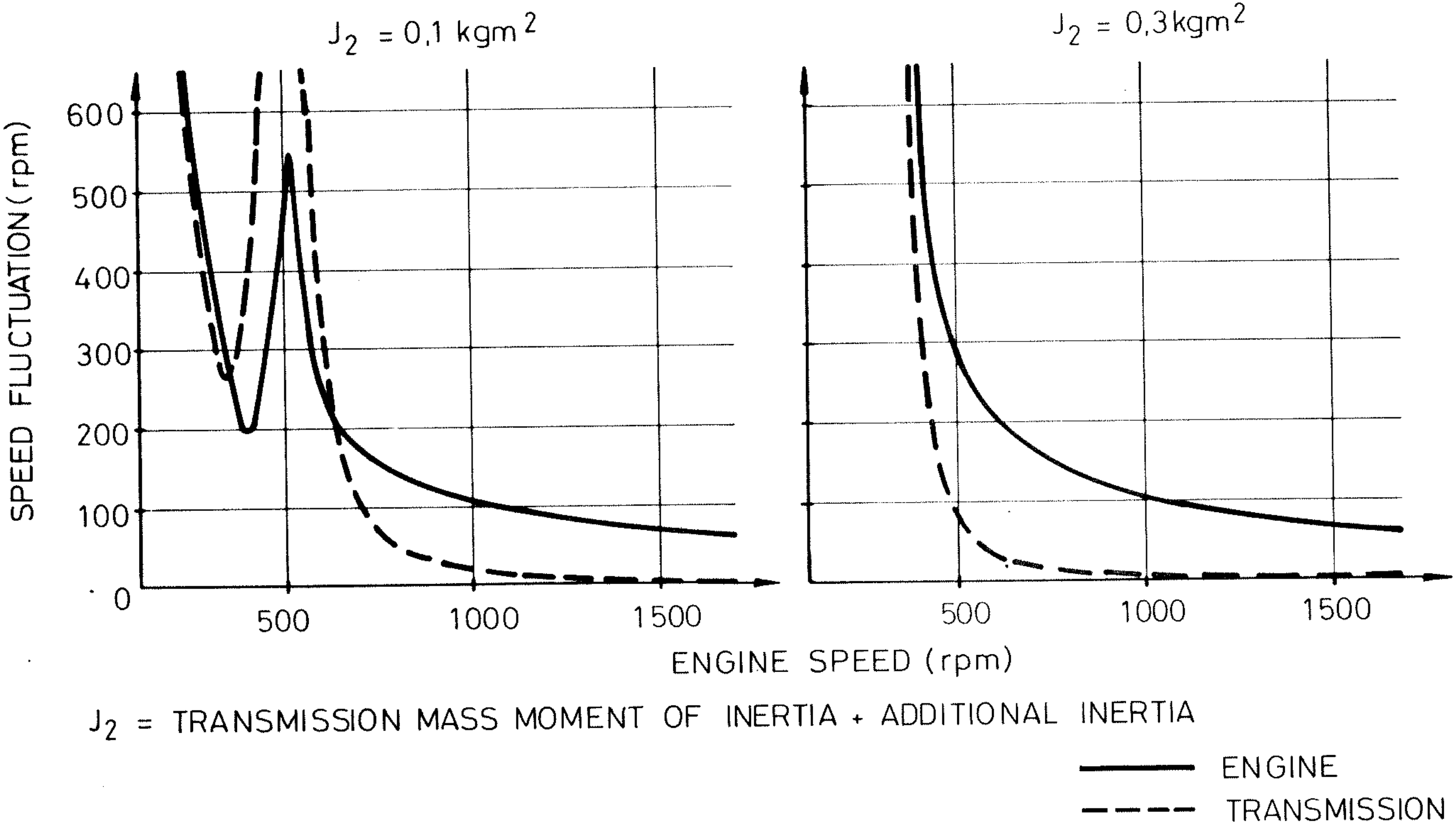
07 02 86	MEASUREMENT OF TORSIONAL VIBRATIONS IN DRIVE MODE	LUK



J_1 = 0,140 kgm^2
J_2 = 0,120 kgm^2
C_1 = 3,3 Nm/°
Hy. = 10 Nm

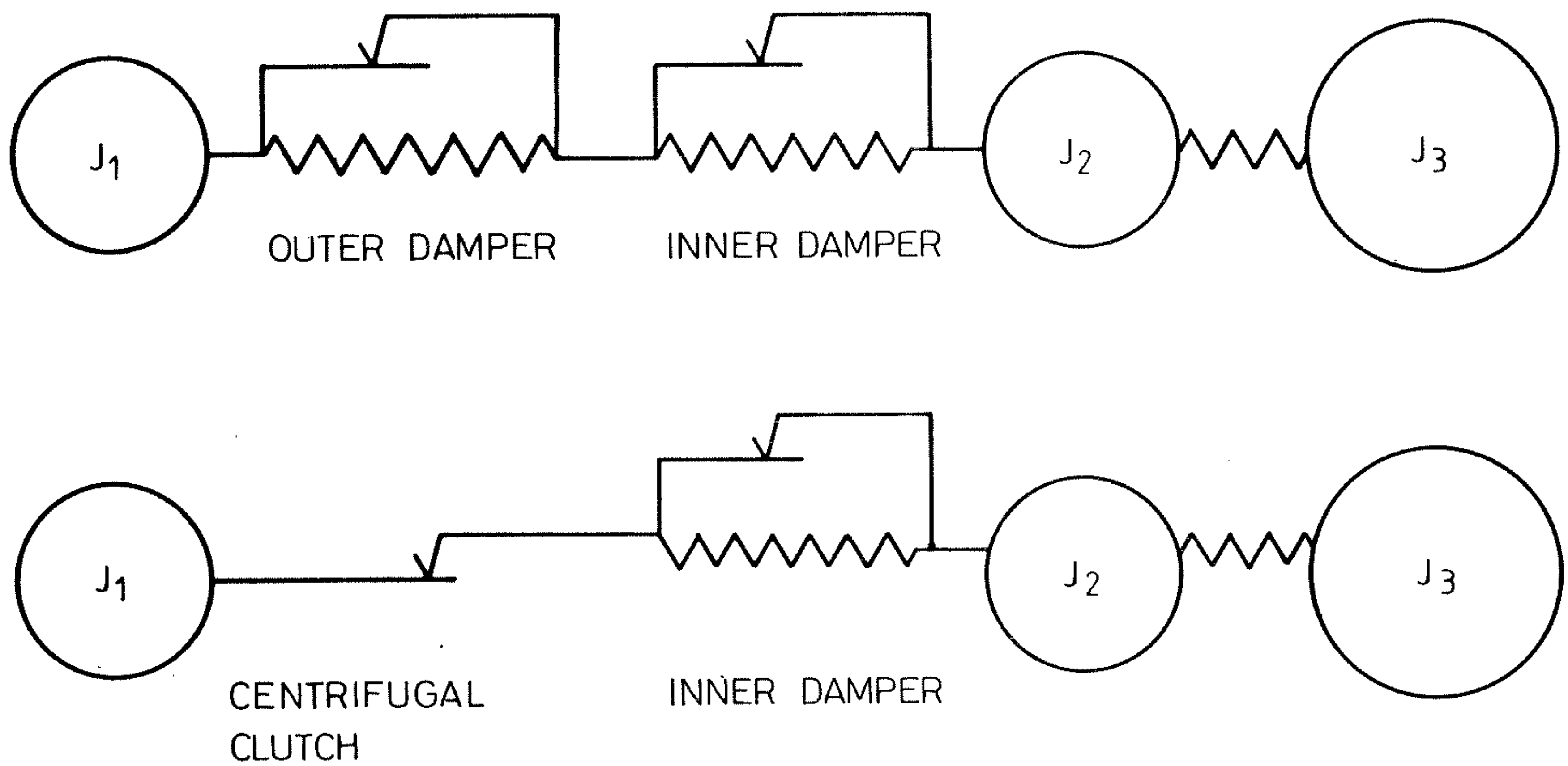
—— ENGINE
– – – – TRANSMISSION

08 02 86	COMPARISON OF CALCULATED AND MEASURED VIBRATION CURVES IN NEUTRAL IDLE



J_2 = TRANSMISSION MASS MOMENT OF INERTIA + ADDITIONAL INERTIA

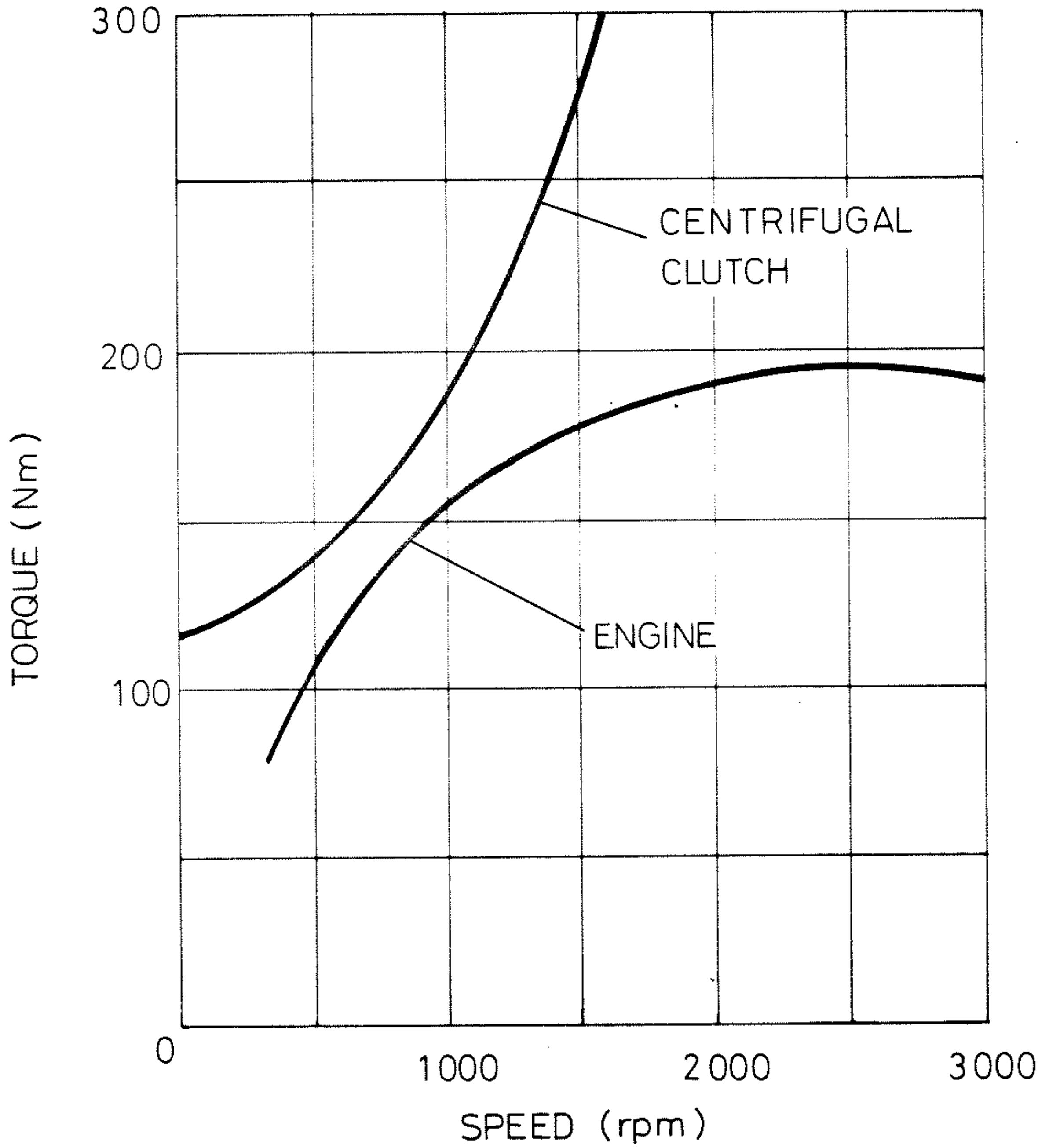
09 02 86	RESONANCE CURVE	LUK



10 02 86

DAMPER ARRANGEMENT IN THE DMFW

LUK

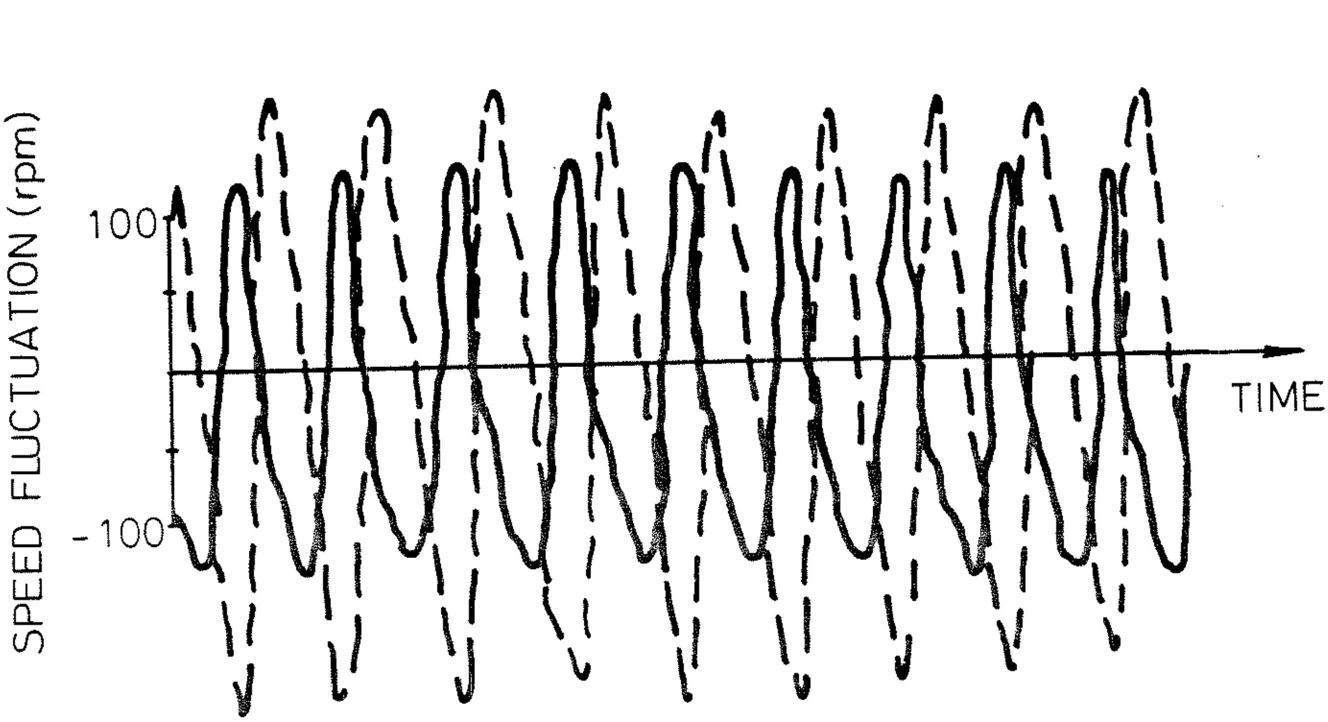


11 02 86

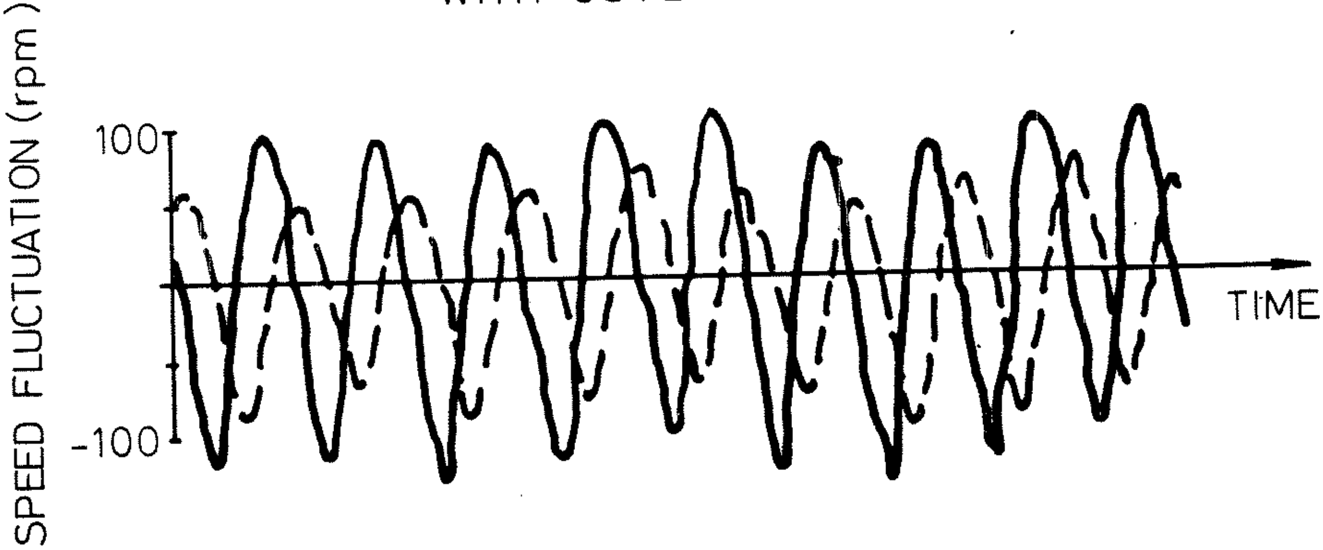
TORQUE CURVE OF ENGINE
AND CENTRIFUGAL CLUTCH

LuK

WITHOUT OUTER·DAMPER



WITH OUTER DAMPER

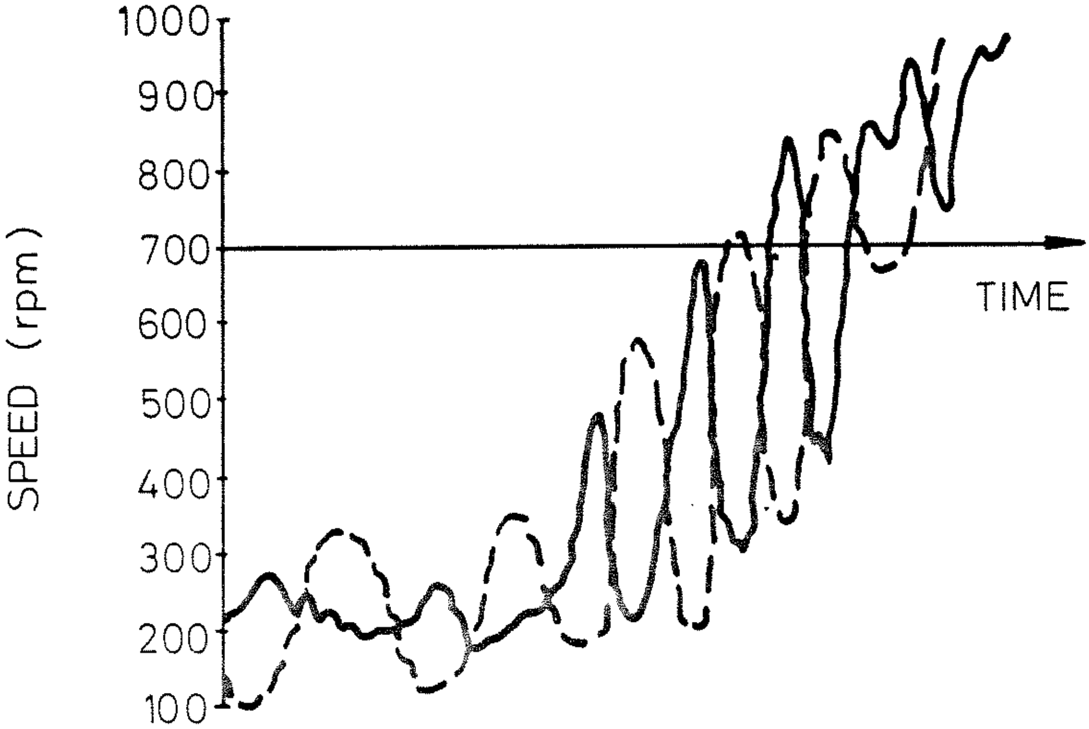


ENGINE SPEED 400 RPM

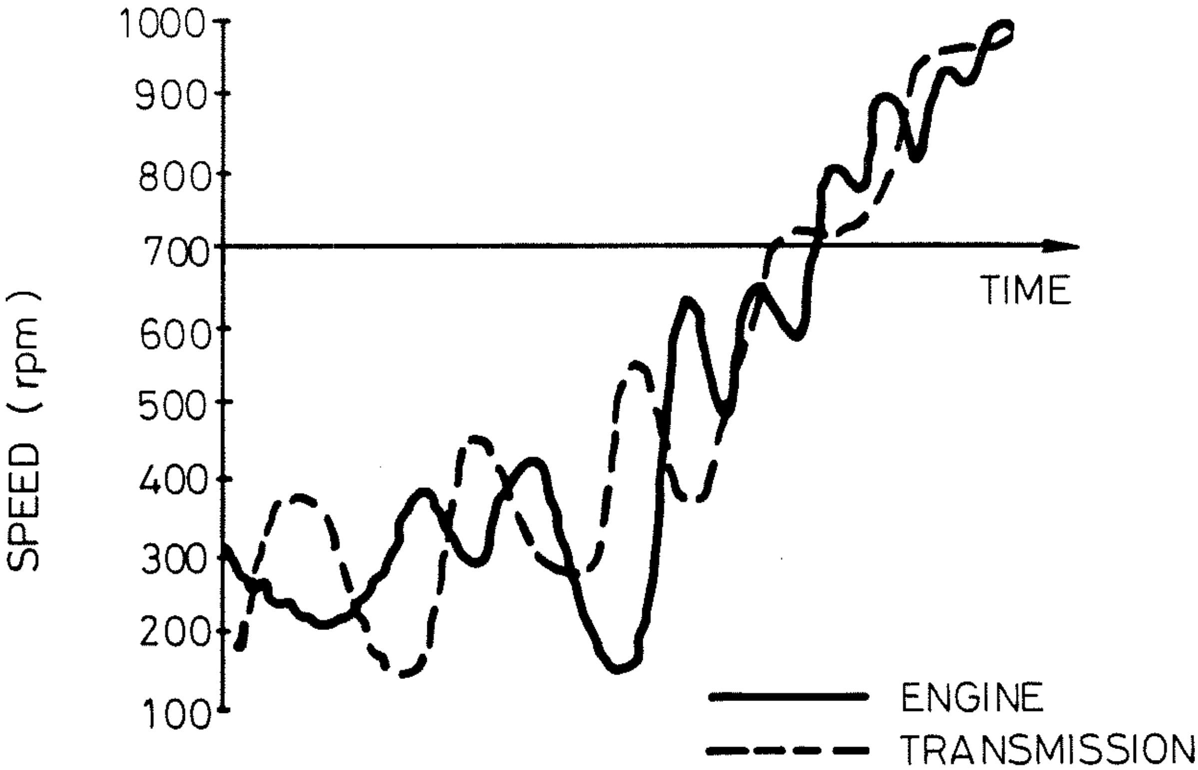
——— ENGINE
- - - - TRANSMISSION

12 02 86	RESONANCE VIBRATION MEASUREMENT WITH A DUAL MASS FLYWHEEL	LuK

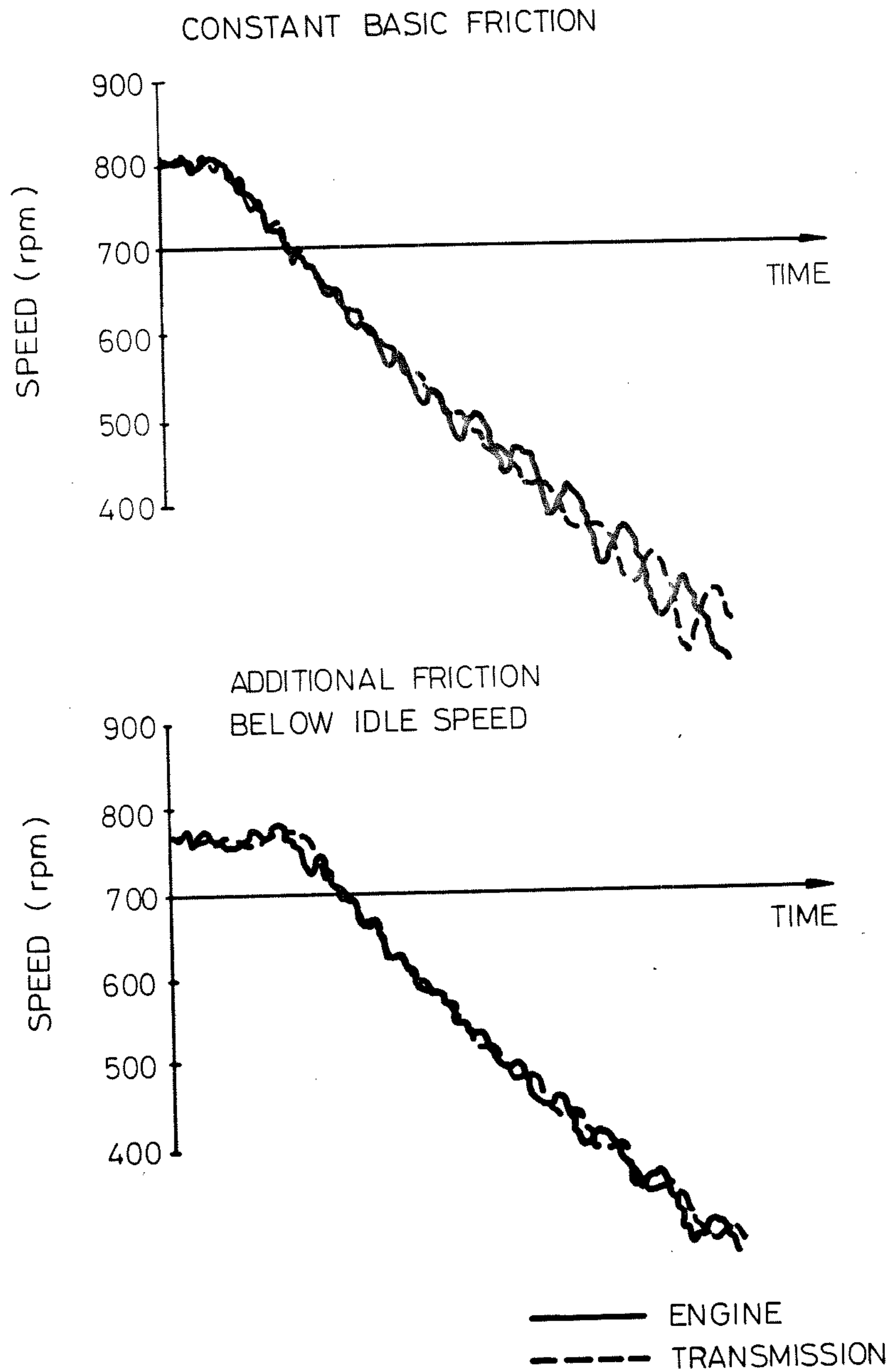
WITHOUT OUTER DAMPER



WITH OUTER DAMPER



13 02 86	MEASUREMENT OF ENGINE START-UP WITH THE DUAL MASS FLYWHEEL	LuK



14 02 86	MEASUREMENT OF ENGINE SHUT-OFF WITH THE DUAL MASS FLYWHEEL

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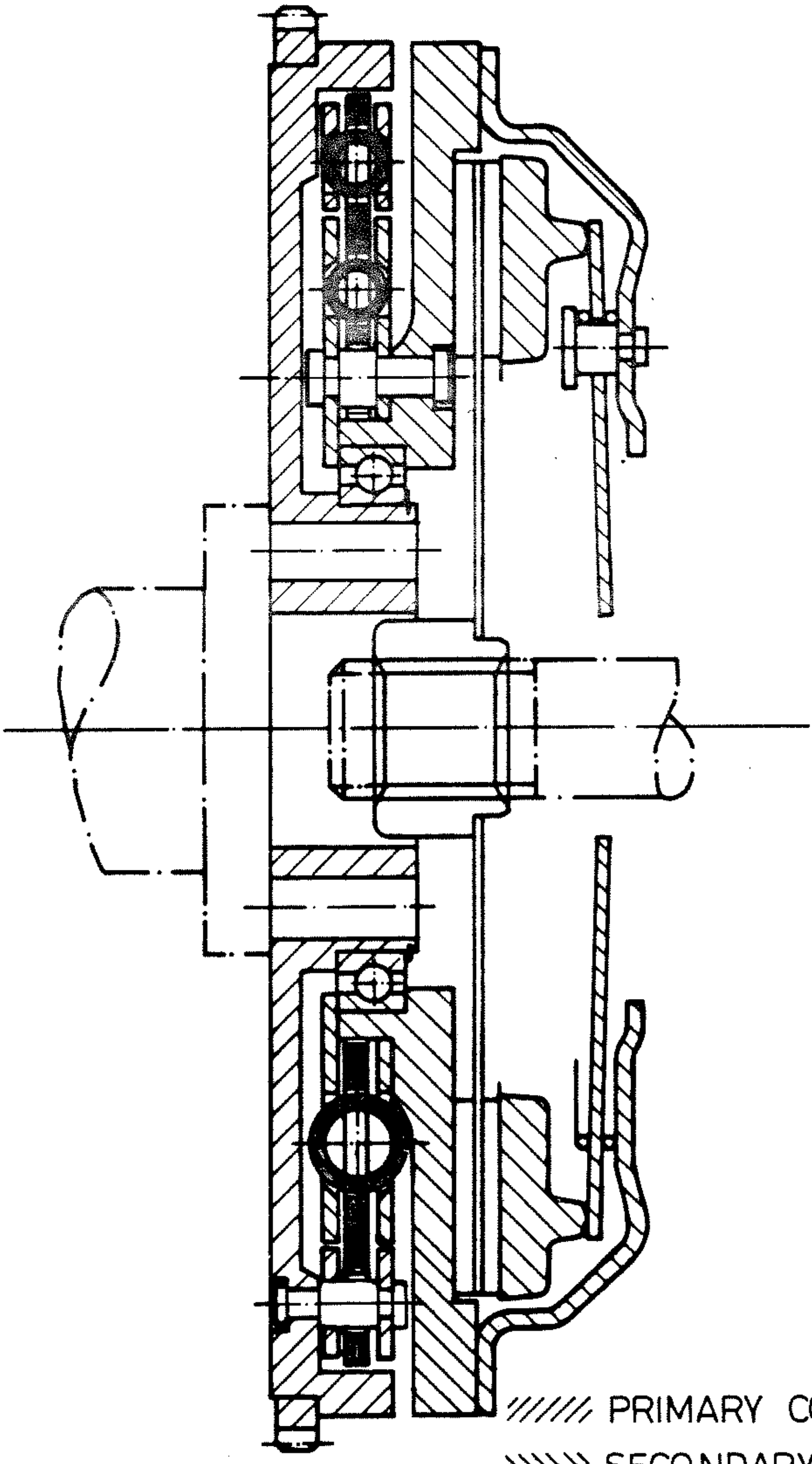
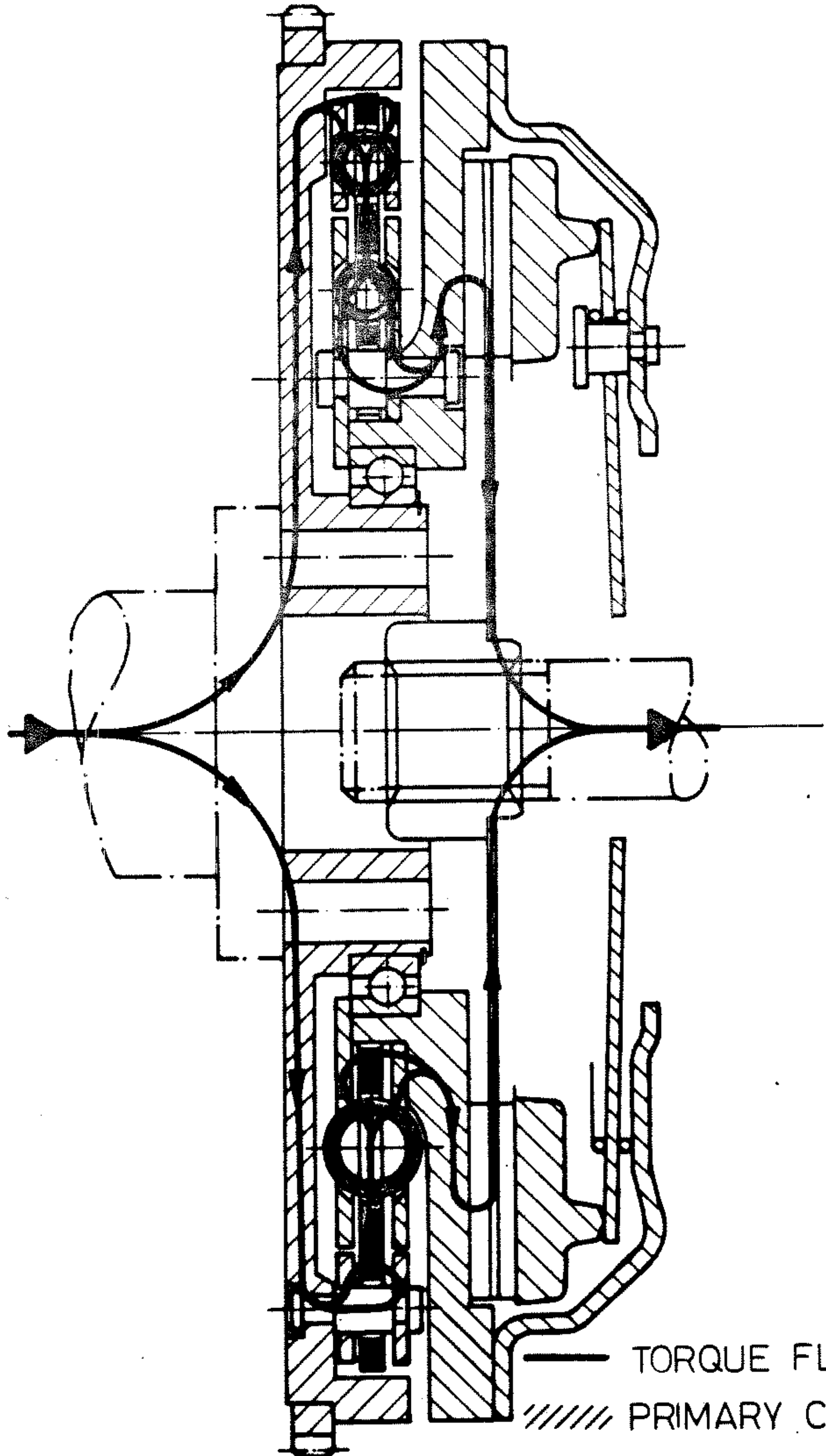


15 02 86	BASIC DUAL MASS FLYWHEEL	LUK



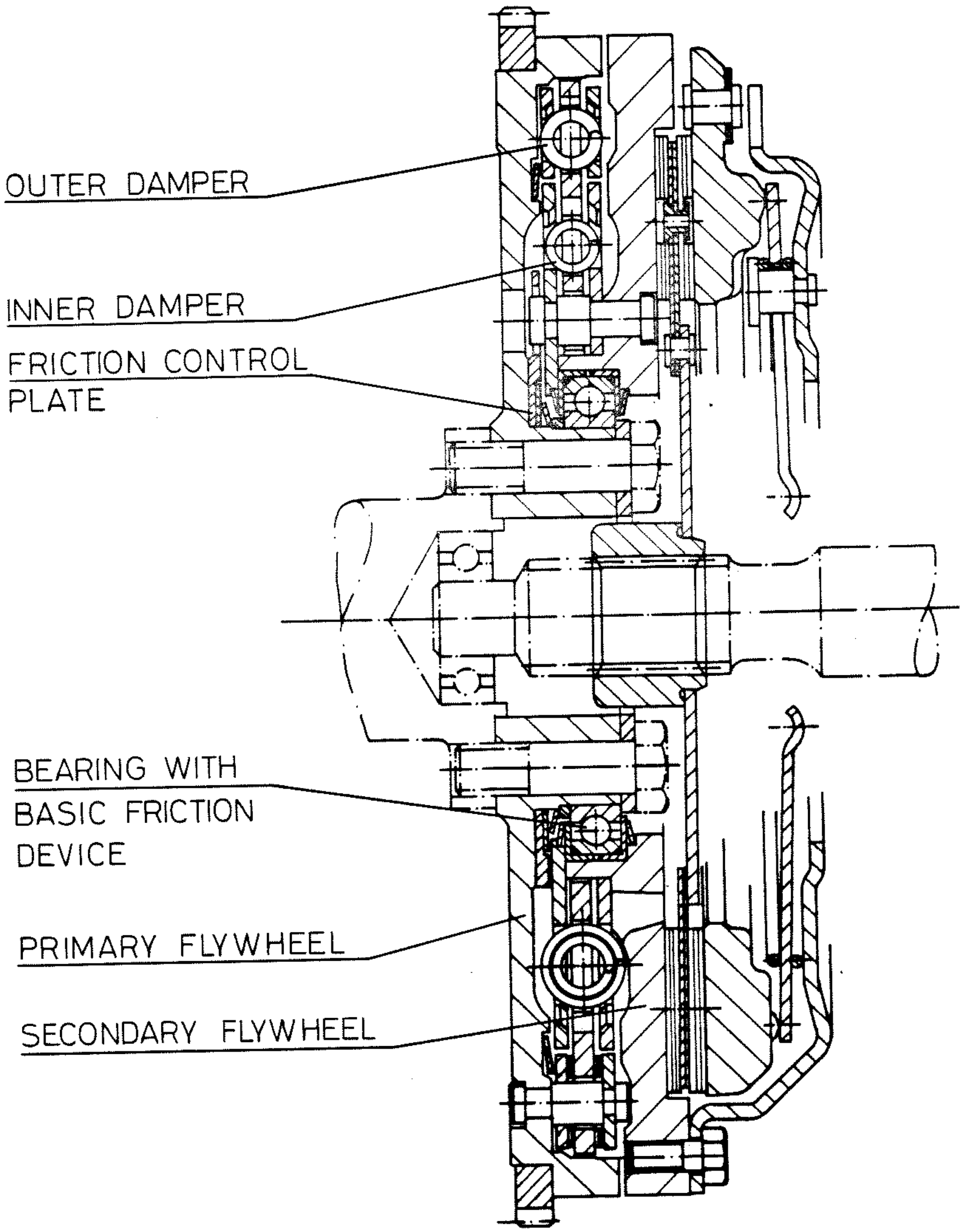
— TORQUE FLOW

////// PRIMARY COMPONENTS

\\\\\\ SECONDARY COMPONENTS

▬ TORSION DAMPER

16 02 86	TORQUE FLOW THROUGH THE DUAL MASS FLYWHEEL



17 02 86	DUAL MASS FLYWHEEL WITH SERIAL DAMPER

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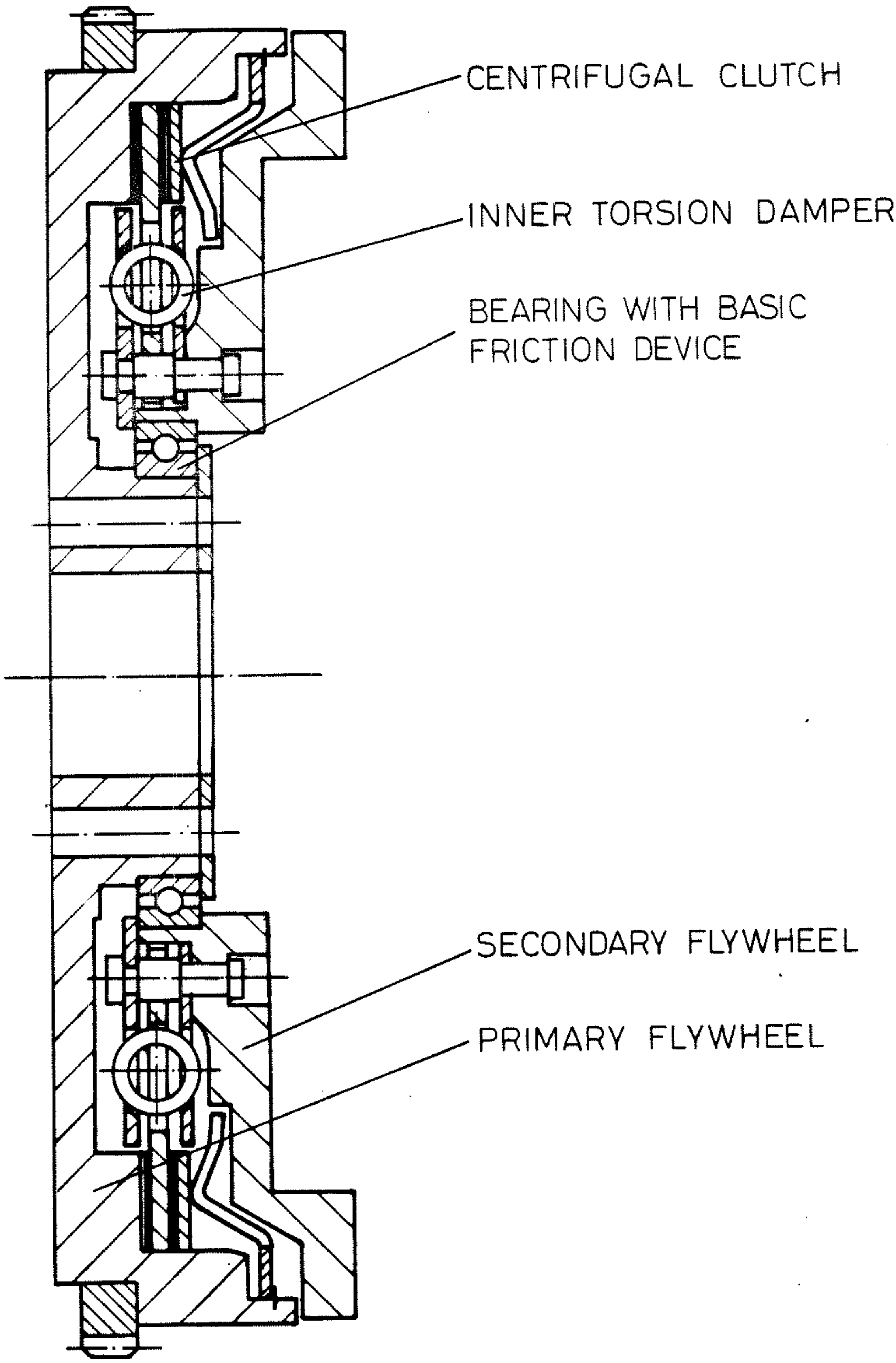
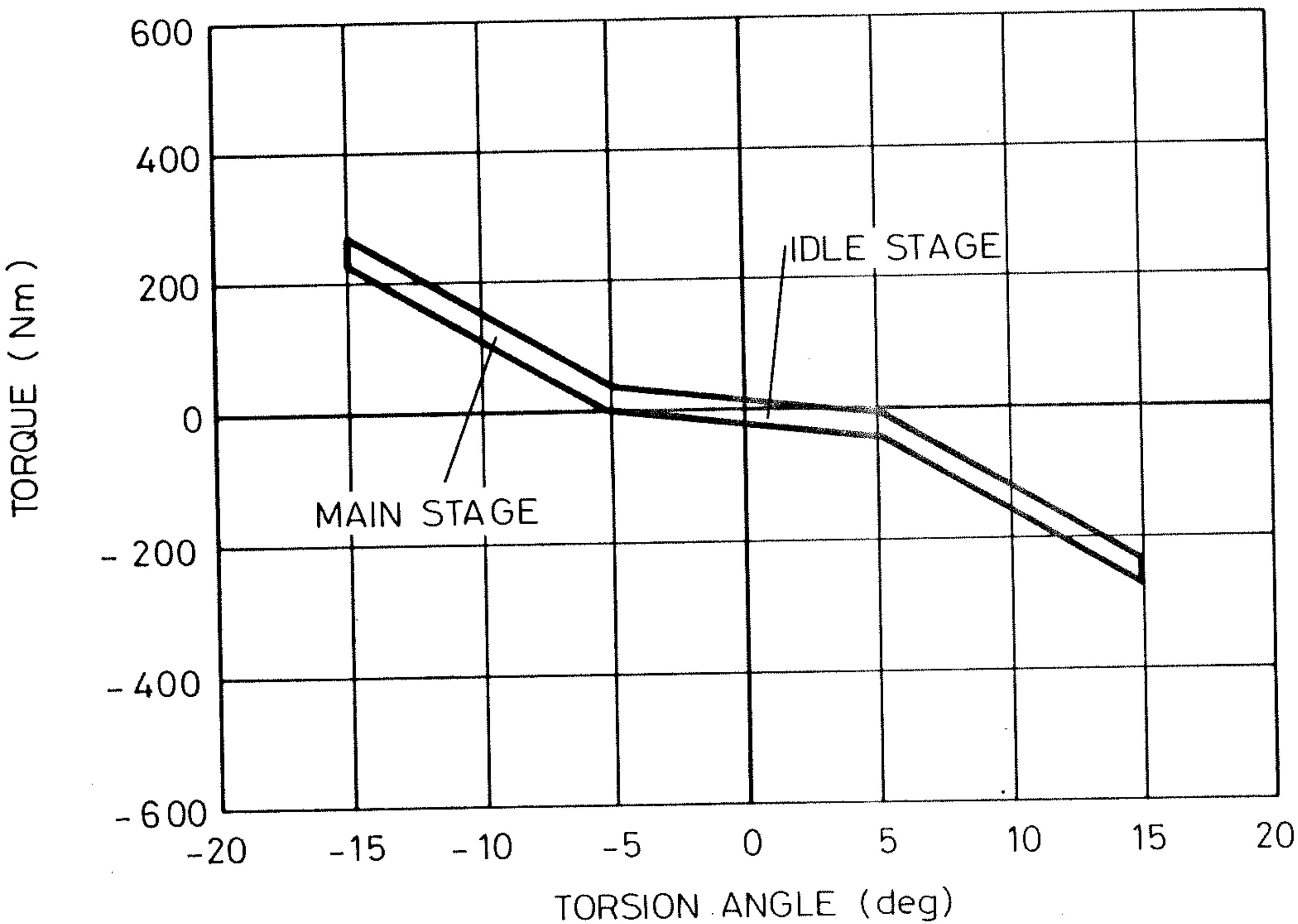
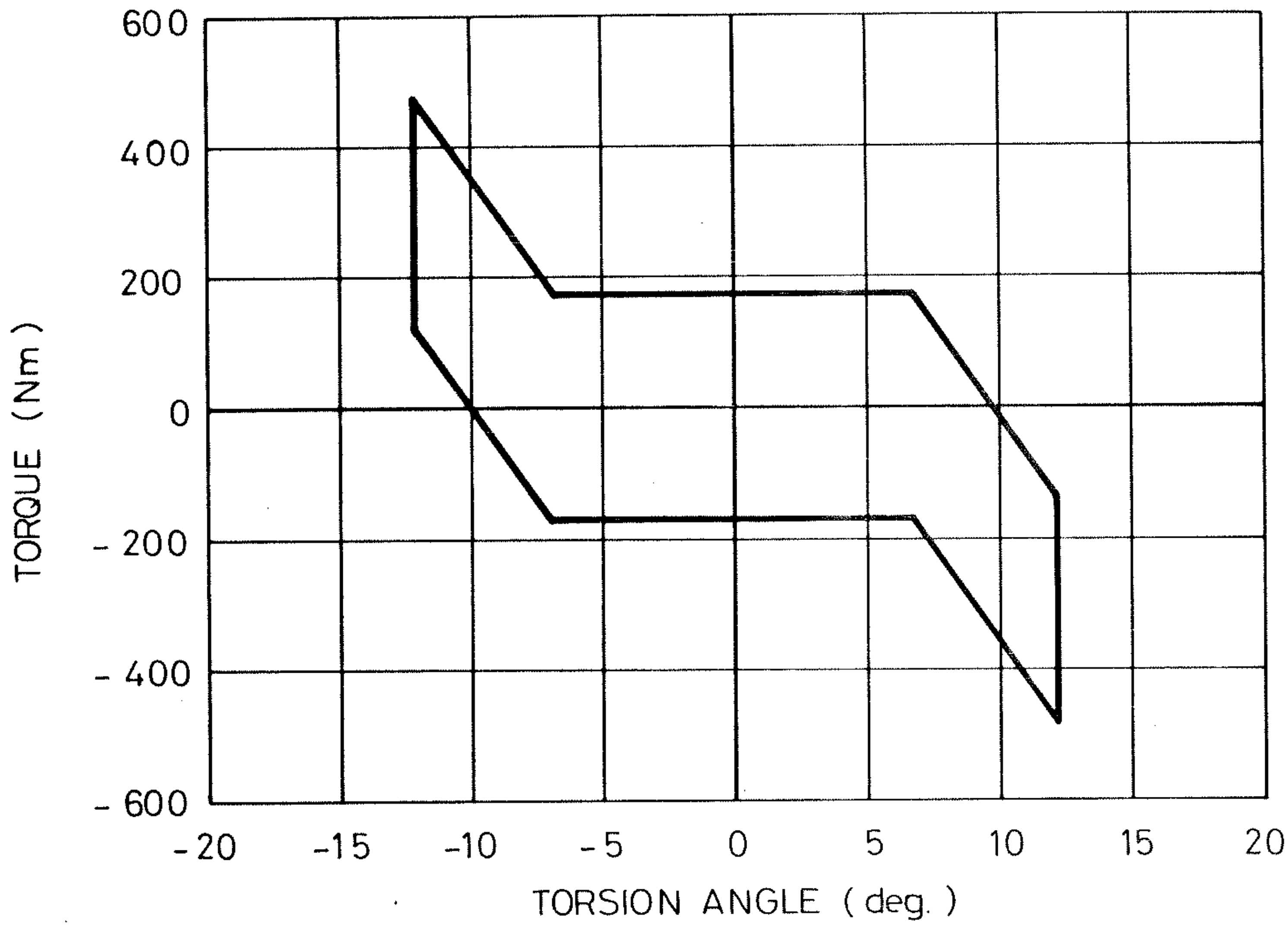


18 02 86	DUAL MASS FLYWHEEL WITH CENTRIFUGAL CLUTCH	LUK



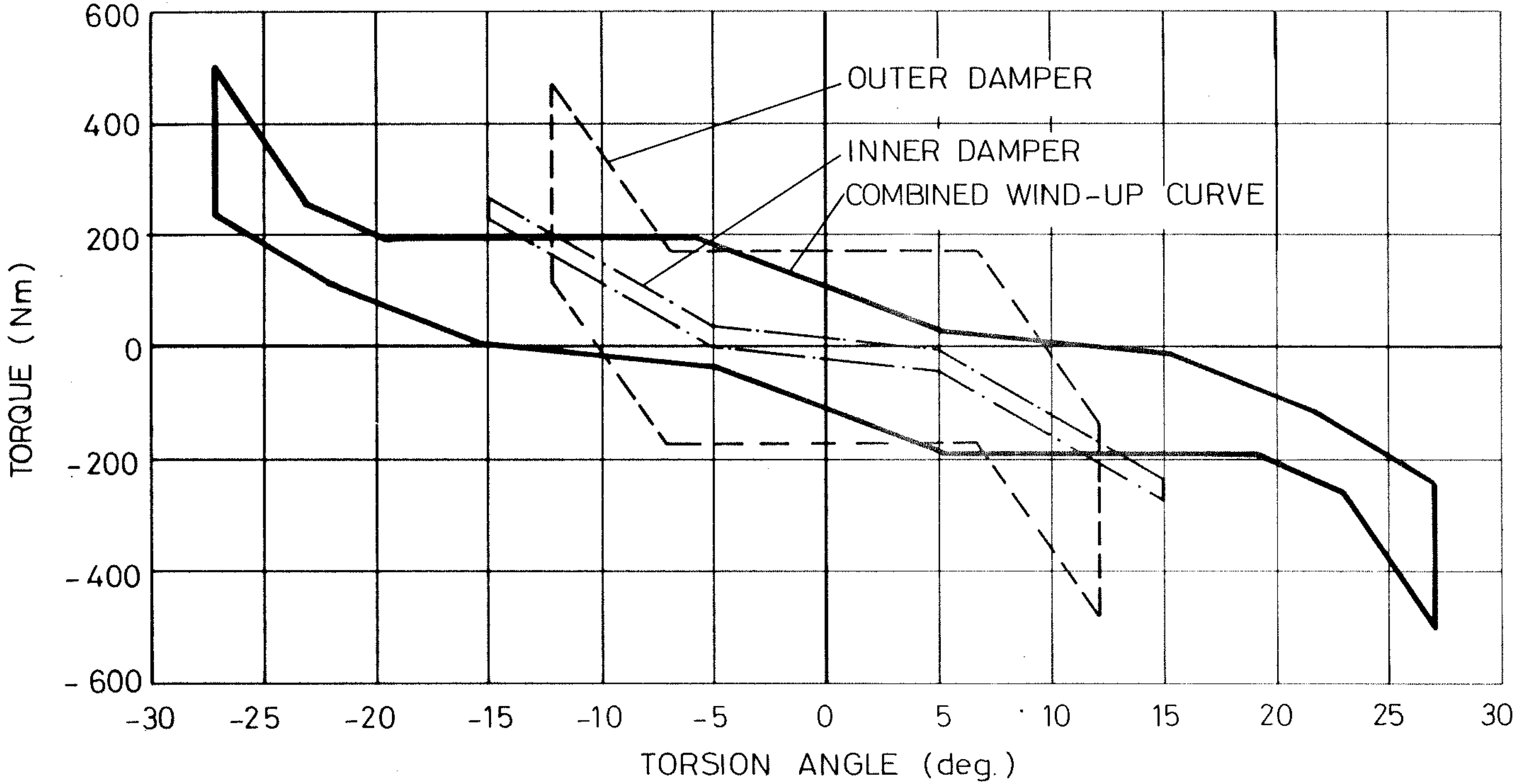
19 02 86	WIND UP CURVE OF THE INNER DAMPER	LUK



20 02 86

WIND - UP CURVE OF THE OUTER DAMPER

LUK



21 02 86	COMBINED WIND-UP CURVE OF THE DMFW WITH SERIAL DAMPER	LUK