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TORSIONAL VIBRATIONS AND TRANSMISSION NOISE

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TORSIONAL VIBRATIONS AND TRANSMISSION NOISE

Introduction:

In the last few years, the development of clutches and clutch discs for passenger cars has focused more and more on the torsion damper. The torsion damper has nothing to do with the basic clutch function of connecting the engine with the drive train. However, it plays an important role of reducing the vibrations transmitted to the drive train to an acceptable level. These vibrations are produced by the engine as a result of each individual combustion cycle.

Driver comfort is the primary concern, that is, decreasing gear rattle or body boom. Only minimal attention is given to the added stress on the transmission due to the torsional vibrations. We can assume that transmission service life will be increased if most of the torsional vibrations are filtered out before reaching the transmission. However, the following discussion will only deal with the comfort aspect.

The Internal Combustion Engine as a Source of Torsional Vibration

Each time the combustion mixture in a cylinder ignites, the gas pressure produces an angular acceleration of the crank shaft, followed by a deceleration due to compression in the next cylinder. This causes fluctuations of the engine speed.

The middle graph in Figure 1 shows the measurement of typical engine speed fluctuations of a 4-cyl. engine with 2 ignitions per revolution. Consequently we talk about a 2nd order excitation.

Integration of the engine speed fluctuations results in the advance and retard of the torsional angle (Figure 1, top graph). This gives us, for example, the minimum torsional angle for the neutral idle range of the damper.

Differentiation of the engine speed yields the angular acceleration curve $\mathring{\omega}$ (Figure 1, bottom graph), from which we can determine the alternating torque values based on the formula $T = J \cdot \mathring{\omega}$ (T = torque, J = moment of inertia). These values are important for calculating the torsional vibrations in the drive train. The angular acceleration is also advantageous for our calculations in that its amplitude, in both idle and drive mode, is dependent on the throttle opening, but hardly at all on the engine speed.

The bottom graph in Figure 2 shows the almost constant amplitude curve of the angular acceleration for drive mode with wide open throttle.

As the engine speed N decreases, speed fluctuations increase (middle graph) by approximately 1/N.

The angular displacement is even more strongly dependent on the engine speed, and increases by a factor of $1/N^2$.

This relationship, which also applies to neutral idle, leads to very significant increases in the amplitude of the vibration angle when the idle speed is reduced.

As shown above, the amplitude of the angular acceleration is practically independent of the engine speed, so it is well suited for comparing different engines.

Figure 3 illustrates the peak-peak amplitude of the angular acceleration for various engine types--diesel, fuel-injected and carbureted--for idle mode as well as for drive with wide open throttle.

In neutral idle 4-cyl. diesel engines exhibit especially high angular accelerations. These differences are less distinct in drive mode. Of course, these values are also strongly influenced by the size of the flywheel.

Effect of Torsional Vibrations on the Transmission

The torsional vibrations produced by the engine are transmitted to the transmission. There is always lash between the transmission gears, so the torsional vibrations can cause rattle noises due to the gear teeth impacting with one another.

How high the torsional vibrations can be--without the gear rattle becoming annoyingly noticeable--depends upon many factors, such as: extraneous noises, noise paths through the body, damping in the transmission, and transmission gear lash.

Figure 4 illustrates the relationship between subjective noise rating and the input of speed fluctuations into the transmission at idle speed. This is shown for a vehicle with a 4-cyl. diesel engine and for a vehicle with a 4-cyl. gasoline engine. For the subjective noise evaluation we used the universal rating scale, beginning with 0 for "very loud," and ending at 10 for "inaudible."

While we were able to get an acceptable noise rating of better than 5 for the diesel engine at speed fluctuations under 70 rpm, speed fluctuation had to be decreased below 20 rpm for the gasoline engine. This could be attributed to the significantly higher noise level of the diesel engine drowning out the transmission noise, and the better acoustic insulation of the diesel vehicle.

Measuring and Calculating Vibrations

At LuK we evaluate the performance of a torsion damper by both measuring and calculating the torsional vibrations. This provides insight into the vibration process and makes it possible to draw appropriate conclusions.

LuK has developed a mobile data recording system which, when installed in the vehicle, simultaneously measures the speed of the flywheel ahead of the torsion damper and of the transmission input shaft behind the torsion damper (Figure 5). Torsional vibrations are determined from the speed measurement.

Engine and transmission rpm readings are sampled at a frequency of 2000 Hz and stored digitally on magnetic tape. To check the stored data it is possible to read out the measured data to a plotter in the vehicle. The data on the magnetic tapes is then fed into a main frame computer for further processing. It is easy to determine whether the torsional vibrations are reduced, unchanged or amplified by resonance as they are transmitted into the transmission. In practice all three situations occur.

Analytical models with various degrees of freedom are available to back up calculations. Figure 6 shows a simple model which is adequate for many cases. The spring and friction components in the drawing represent the torsion damper and connect the rotating parts of the engine and the transmission with each other. The spring and damping element between the transmission and the vehicle mass represent the drive train.

We always use comparative measurements to check whether models of this kind truly describe the actual drive train vibration system. We have to use complicated analytical models only in exceptional cases. It is impossible to take all the fine details of the drive train into consideration because the vehicle manufacturers generally do not have all the necessary data, such as inertias, elasticities, and lash in the transmission and differential, u-joints, shafts, etc. As shown in the following discussion, this is not generally necessary. Given the appropriate level of experience, even simple models can provide all the necessary information the clutch manufacturer needs.

LuK uses non-linear analytical models to take into account the engine excitation and all the typical features of a torsion damper. Figure 7 shows a schematic wind-up characteristic curve. Special features include:

- 8-stage torque rate
- Pre-loaded coil springs
- Spline lash between the clutch disc and the transmission input shaft
- 4-stage hysteresis.

In addition, the following features are also possible:

- Velocity-proportionate damping
- Floating or spring-loaded friction control plate
- Two damper characteristics in series.

The same test engineer calculates and measures the torsional vibrations and evaluates the vehicle noise. This leads to an in-depth understanding of the processes involved.

Figure 8 shows how measurement and calculation mutually complement each other based on an example of neutral rattle tuning.

First the vehicle is evaluated and vibrations are measured with the current damper installed (see the upper left graph, Figure 8). The evaluation yields the speed fluctuations of the engine and of the transmission. The first step in the tuning process involves a vibration calculation. This requires the following data:

- Engine irregularity
- Mass moments of inertia of the engine, the flywheel, the clutch, the clutch disc, and the transmission

- Wind-up characteristic of the installed torsion damper
- Drag torque of the transmission
- Damping properties of the transmission oil.

The graph on the upper right shows the results of this kind of calculation. The correlation between this measurement and the calculation is usually very good, as long as the data cited above are sufficiently accurate.

At this point the calculation is repeated using an optimized wind-up characteristic for neutral idle. It is possible to find a wind-up characteristic which can be achieved and which will reduce the torsional vibrations in the transmission (Figure 8 bottom right). In the third step, a clutch disc with this wind-up characteristic is evaluated in the vehicle (bottom left).

If vehicle tests indicate that the calculated characteristic curve is still not optimal, steps 2 and 3 are repeated.

Torsion Damper Tuning for Neutral Idle Mode

In most cases neutral rattle problems are solved by lowering the natural frequency below the engine idle speed. Figure 9 shows a resonance curve for which the magnification function, that is, the ratio between the vibration amplitudes on the transmission input shaft and those on the crank shaft, is plotted schematically vs. the engine speed. This is a 1 to 1 ratio at very low speeds, meaning that the torsional vibrations on the transmission input shaft are exactly equal to those of the engine. When speed approaches the natural frequency, the magnification function and the torsional vibrations on the transmission input shaft take on high values. Therefore we must definitely avoid this speed range. The vibration isolation range begins at speeds above about 1.5 times the natural frequency. Torsional vibrations of the transmission input shaft become smaller to the extent by which the operating speed exceeds the natural frequency. This can be achieved with very low torque rates of 0.1 to 0.6 Nm/degree. When the damper is operating above the natural frequency, torsional vibrations can be quite effectively isolated from the transmission.

However, we have difficulties with this kind of flat neutral idle characteristic because the transmission drag torque is noticeably higher at low temperatures due to the fact that the transmission fluid is more viscous, as shown in Figure 10. Of course, the amount of drag torque also depends upon the kind of transmission in the vehicle. For instance, a 5-speed transmission has a higher drag torque than a 4-speed transmission.

Figure 11 shows a schematic damper characteristic with an idle stage. Initially, one would anticipate that the torsion damper would vibrate symmetrically about the zero position. Actually the operating range is displaced in the drive direction corresponding to the transmission drag torque which is dependent on the temperature of the transmission fluid.

If the drag torque is greater than the torque capacity of the idle stage, the torsion damper will vibrate in the transition range between the idle stage and the drive stage. Each vibration cycle then "bumps" on the drive stage. This is measured as a sudden strong acceleration in transmission input and is acoustically noticeable as gear rattle. Figure 12 shows the measurement and calculation of the torsional vibrations described above, which can be observed at average temperatures.

One can clearly see the sharp speed rise as a result of this "bumping." At lower temperatures, the entire operating range will lie in the range of the drive stage and no "bumping," with its high acceleration peaks, will occur.

This behavior can be simulated by the computer by continuously increasing the drag torque. Figure 13 shows the computed result of the vibrations in the transmission with increasing drag torque or decreasing transmission fluid temperature. When the transmission is hot, torsional vibrations in the transmission are minimal. When the transmission is cold, there is no vibration isolation because the torsion damper is vibrating in the drive stage. Nevertheless there is generally no audible gear rattle because the viscous fluid provides sufficient damping. The area in the middle of the temperature range where bumping on the main stage occurs rarely spans more than 10°C in practical cases. Consequently it is easily overlooked during tuning, but can be localized with sufficient accuracy using the vibration calculation.

The longer the idle stage of a multi-stage torsion damper is, the greater will be the deviation from a nominal linear torsion damper characteristic. Then phenomena occur, particularly in diesel vehicles, that are generally observed for non-linear vibration systems.

We know from vibration theory that the resonance curve tips to the right for progressive spring characteristics, as shown in Figure 14. For purposes of comparison, the graph at the top of the illustration again shows the resonance curve with a linear characteristic. In the case of the non-linear characteristic (bottom graph) the system can assume two stable vibration conditions, shown here as Points 1 and 2, in the speed range above the natural frequency. Point 1 is the result of a smooth clutch engagement with low vibration amplitudes in the transmission. At Point 2 clutch engagement is abrupt and leads to resonance.

Figure 15 compares these two vibration conditions. At Point 1 (on the left) the speed fluctuations of the transmission input (dotted line) are clearly lower than those of the flywheel. At Point 2 (right-hand graph) the vibrations are very much greater. The correlation between the vibration measurement and calculation is good.

The unfavorable vibration condition at Point 2 and the resulting strong transmission noise can be prevented by using an additional friction control device effective only at high vibration amplitudes (Figure 16). This is why many torsion dampers for diesel vehicles have friction control plates. This device eliminates the overhang in the resonance curve if the friction values and the operating angle are correct.

Torsion Damper Tuning for the Drive Mode

In drive mode, that is, when driving under load, the goal of torsion damper tuning is to reduce the amplitude of the vibrations input into the transmission, just as was the case for neutral idle rattle tuning.

In passenger cars it is practically impossible to move the natural frequency below the operating speed in the way we can do it for the neutral idle mode. This would require a torque rate under 1 Nm/degree, a rate which would yield a damper wind-up angle which is not feasible. Hence torsion damper tuning in the drive mode is only capable of suppressing resonance peaks through friction or by displacing them into ranges where they are less audible.

Figure 17 illustrates the effects of resonances based on the example of a front-wheel drive vehicle with a 4-cyl. engine.

The top graph plots subjective noise ratings with respect to engine speed. Significant noise peaks are registered at about 900 and 1750 rpm. Simultaneous torsional vibration measurements were taken on the transmission input shaft. They are plotted as speed fluctuations with respect to the engine speed. They also show two peaks (middle graph), and the envelope curve corresponds to the subjective noise rating.

The bottom graph in Figure 17 compares vibration measurements and their related calculations for characteristic engine speeds. We see resonance of the 2nd order at about 1750 rpm. At half this engine speed, about 900 rpm, resonance of the 4th order appears, causing the transmission to vibrate at twice the engine frequency. Lower torsional vibrations and consequently lower transmission noises are observed at engine speeds between these two points. We don't enter the overcritical range, that is the vibration isolation range, until we reach speeds over 2000 rpm.

Current torsion damper tuning uses computers to study the entire characteristic field of speed fluctuations with relationship to engine speed, hysteresis and torque rates. Figure 18 shows this kind of overview, in which the speed fluctuations of the input shaft are plotted for different torque rates and hysteresis combinations. At low hysteresis values, we can see that the resonance speeds are between 1000 and 4000 rpm for all the illustrated torque rates. It is easy to recognize that it is practically impossible to move the resonances out of the driving range. If we increase the hysteresis, the torsional vibration of the transmission approaches that of the engine, until we have a quasi-rigid performance. In many cases this represents the best possible condition for the drive mode.

This quasi-rigid performance could lead to the assumption that we don't need any torsion damper at all, that a rigid clutch disc is adequate.

However, this is not the case because a "rigid" clutch disc, together with the transmission input shaft, possesses a torque rate. Depending upon the vehicle, this rate is 100 to 600 Nm/degree, which is not really rigid. In general this produces resonances at higher speeds, leading to high amplitudes due to the lack of damping and consequently causes strong transmission rattle. For the most part vibration performance corresponds to the lower left-hand graph with the high damper torsion rate.

In summary, we have determined that, using conventional torsion dampers, it is not possible to achieve effective vibration isolation for drive mode over the entire relevant engine speed range.

Torsion Damper Tuning for Coast Mode

In contrast to drive mode, in which the strongest gear rattle generally occurs at engine speeds below 2000 rpm, coast rattle usually occurs at engine speeds far above 2000 rpm, although in both cases the computed resonance speeds are the same. We have shown in Figure 2 that in drive mode the angular acceleration, and consequently the vibration excitation vs. the engine speed, is practically constant. As illustrated in Figure 19, the angular acceleration decreases significantly in coast mode with reduced engine speed. Under 2000 rpm there is no longer any strong vibration excitation, which could lead to resonance. Hence coast noises usually are limited to high engine speeds.

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As a result, experience has shown that it is usually satisfactory to reduce the resonance speed below about 2000 rpm. We can achieve this with damper torque rates of about 10 Nm/deg. Since the torque capacity of the coast side doesn't have to be as high as the torque capacity of the drive side, we rarely run into difficulties here.

Summary

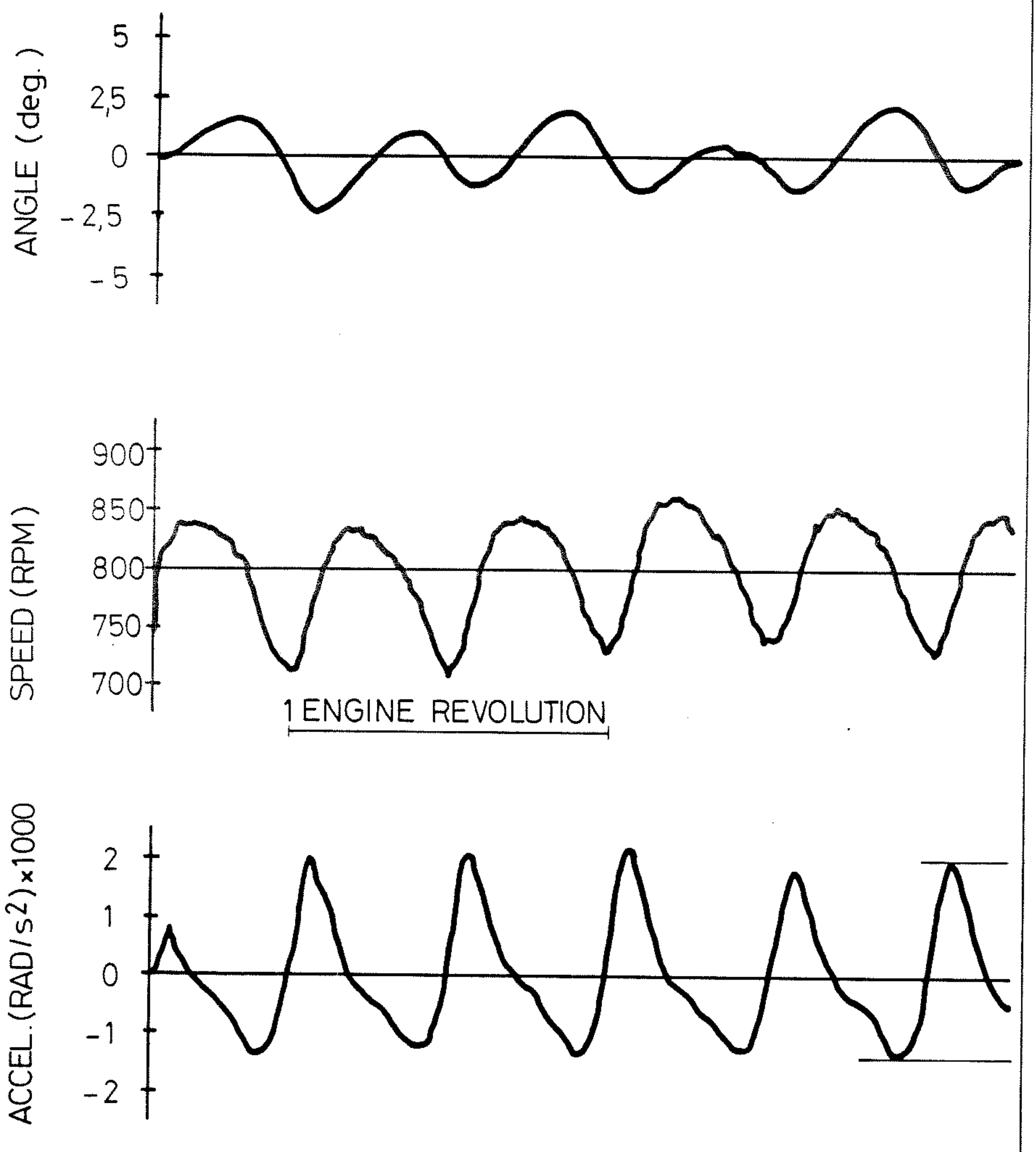
In neutral idle mode it is almost always possible to isolate the torsional vibrations from the transmission using an appropriate idle stage in the torsion damper. Because engine excitation, transmission design, moment of inertia, drag torque and noise insulation vary considerably, the idle characteristic of the torsion damper must be optimized for each vehicle model.

For coast mode, weak excitation at low engine speeds and low coast torques usually make it possible to find an acceptable solution to gear rattle.

Vibration isolation in drive mode is only partially successful because it isn't practical to build torsion dampers that will shift the resonance speeds to a range below the idle speed.

Consequently optimizing the torsion damper for drive mode must be limited to shifting resonances and damping them through friction. It is often impossible to achieve satisfactory vibration isolation at low engine speeds with conventional torsion dampers.

The following presentation will discuss the dual mass flywheel, which makes it possible to shift resonance speeds to very low engine speeds even in the drive mode. Consequently it opens up a new avenue for eliminating transmission rattle.



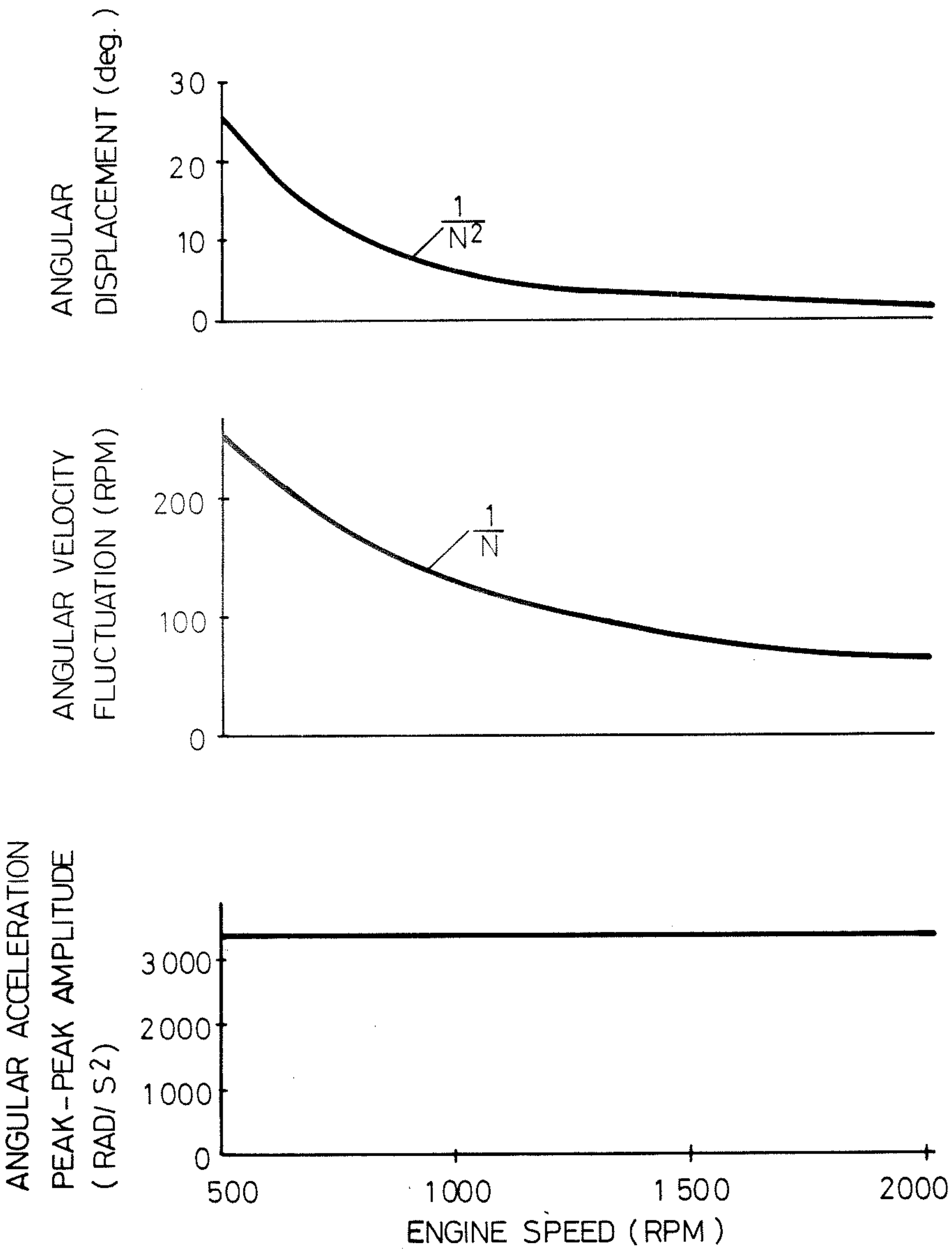
$T = J\dot{\omega}$

T = TORQUE

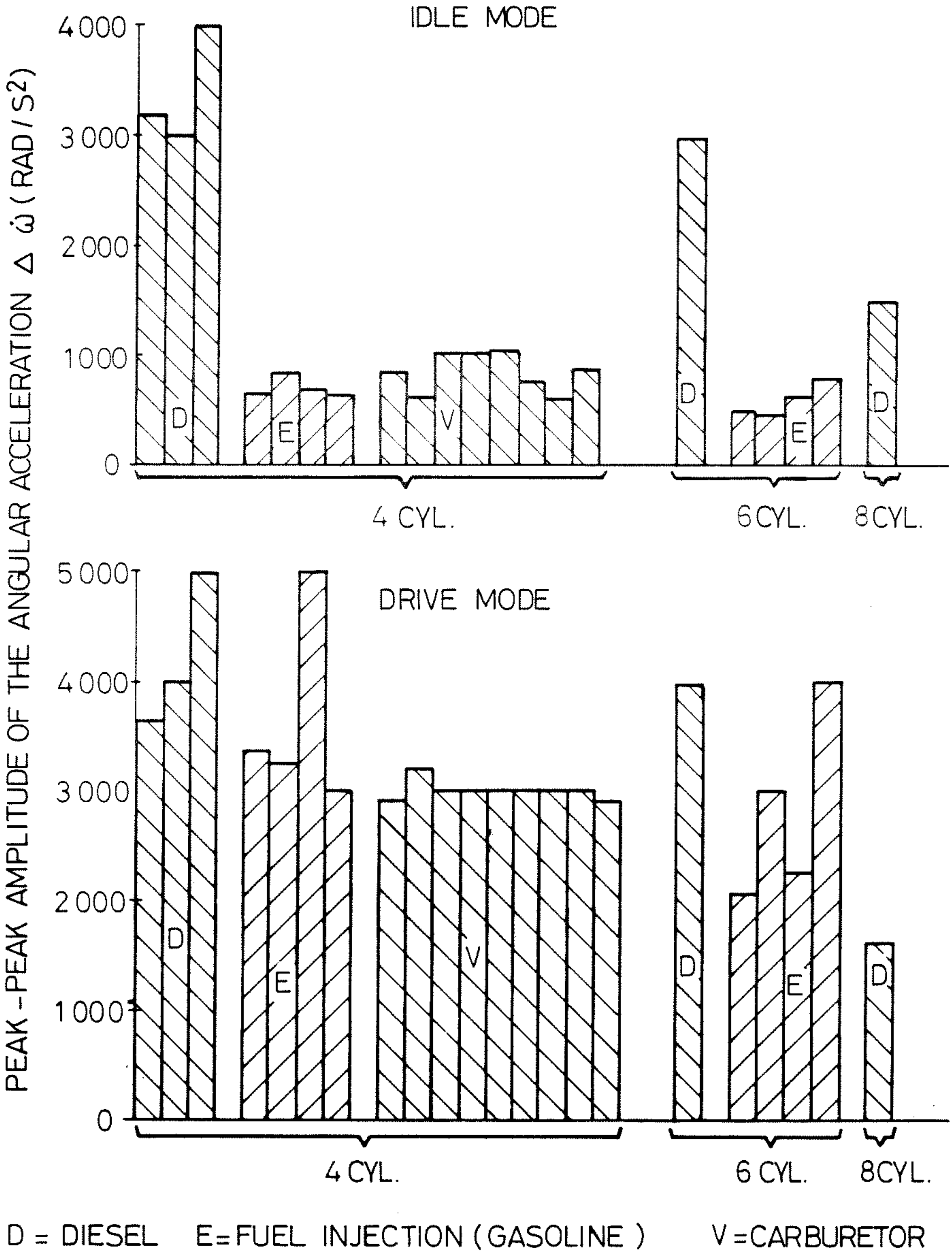
J = MOMENT OF INERTIA

$\dot{\omega}$ = ACCELERATION

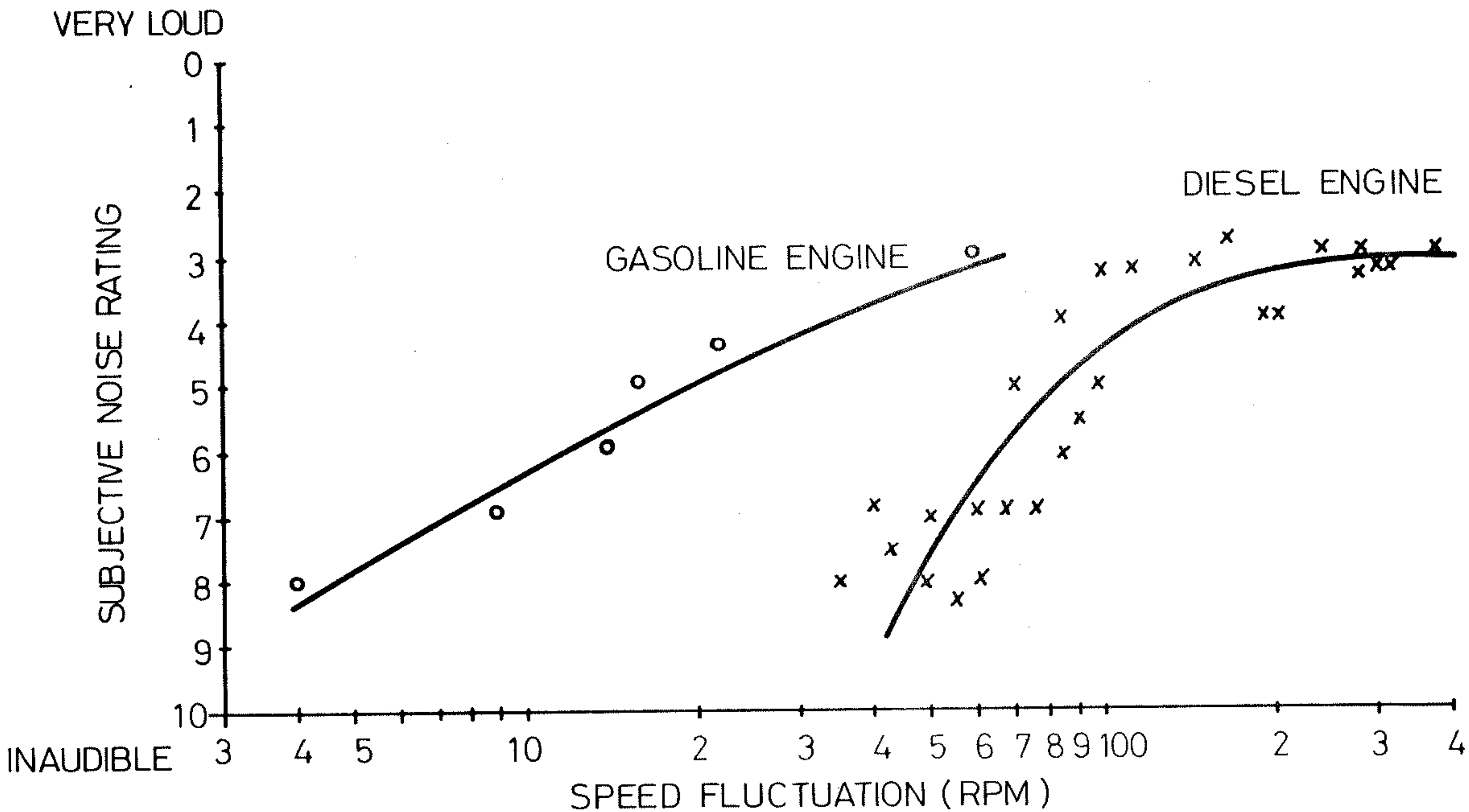
01 01 86	TORSIONAL VIBRATION OF THE CRANK SHAFT



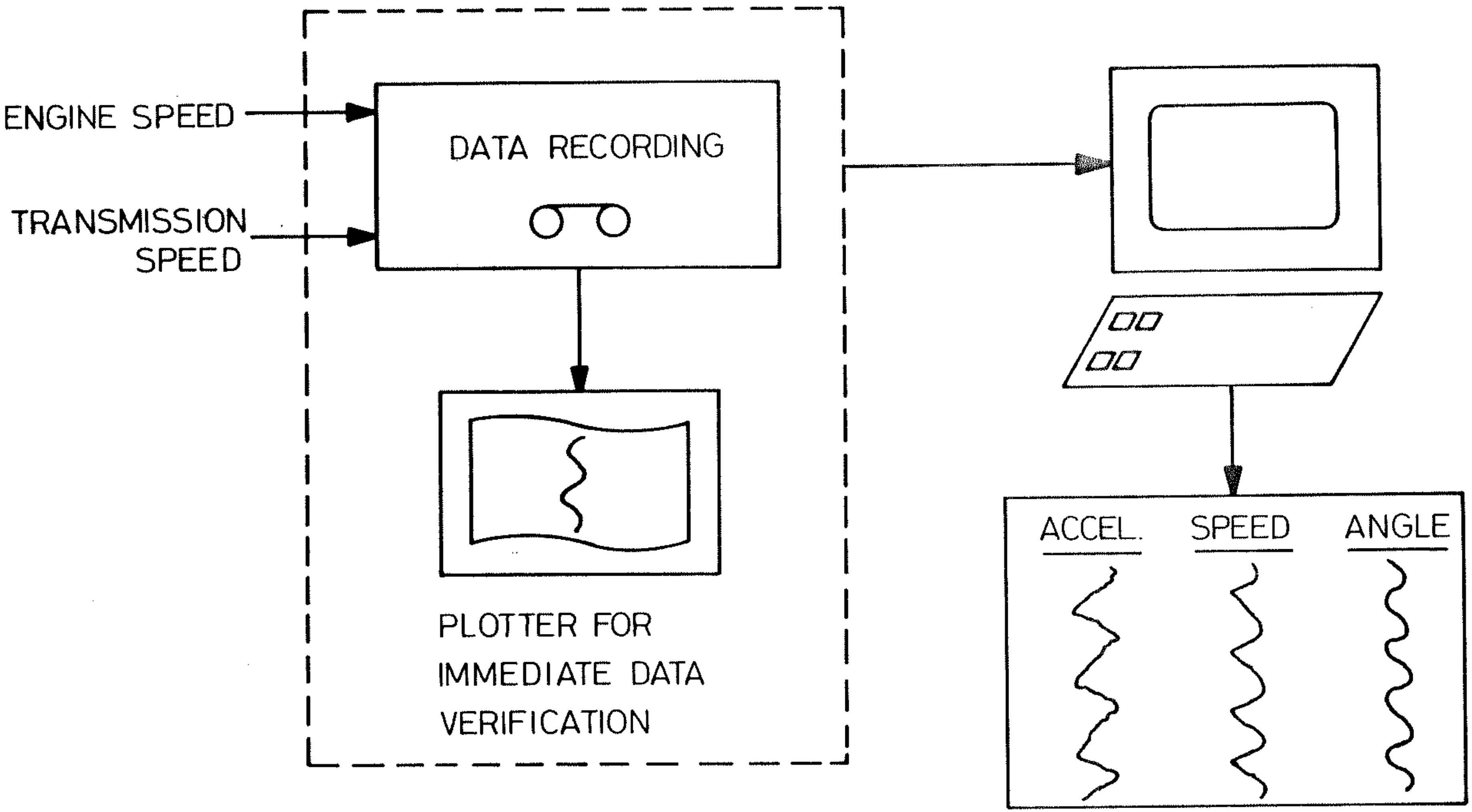
02 01 86	ANGULAR DISPLACEMENT, ANGULAR VELOCITY FLUCTUATION AND ANGULAR ACCELERATION VS. ENGINE SPEED	LUK



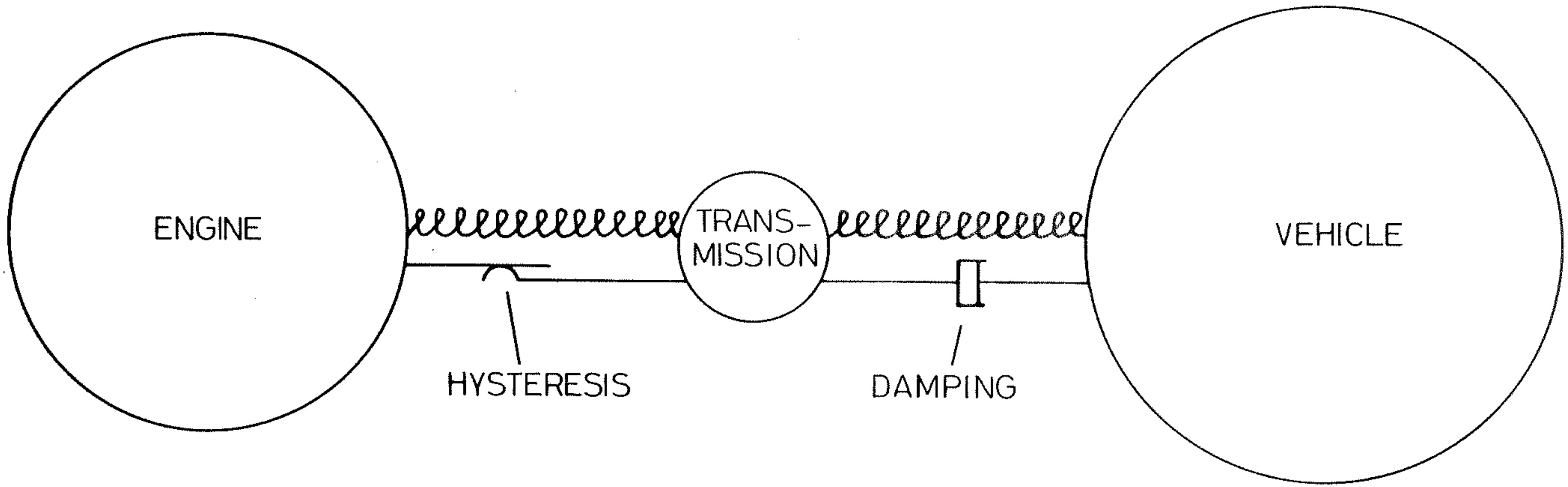
03 01 86	TORSIONAL EXCITATION FOR VARIOUS ENGINES	LUK



04 01 86	EFFECT OF SPEED FLUCTUATION ON THE NOISE RATING



05 01 86	DATA RECORDING SYSTEM	LUK



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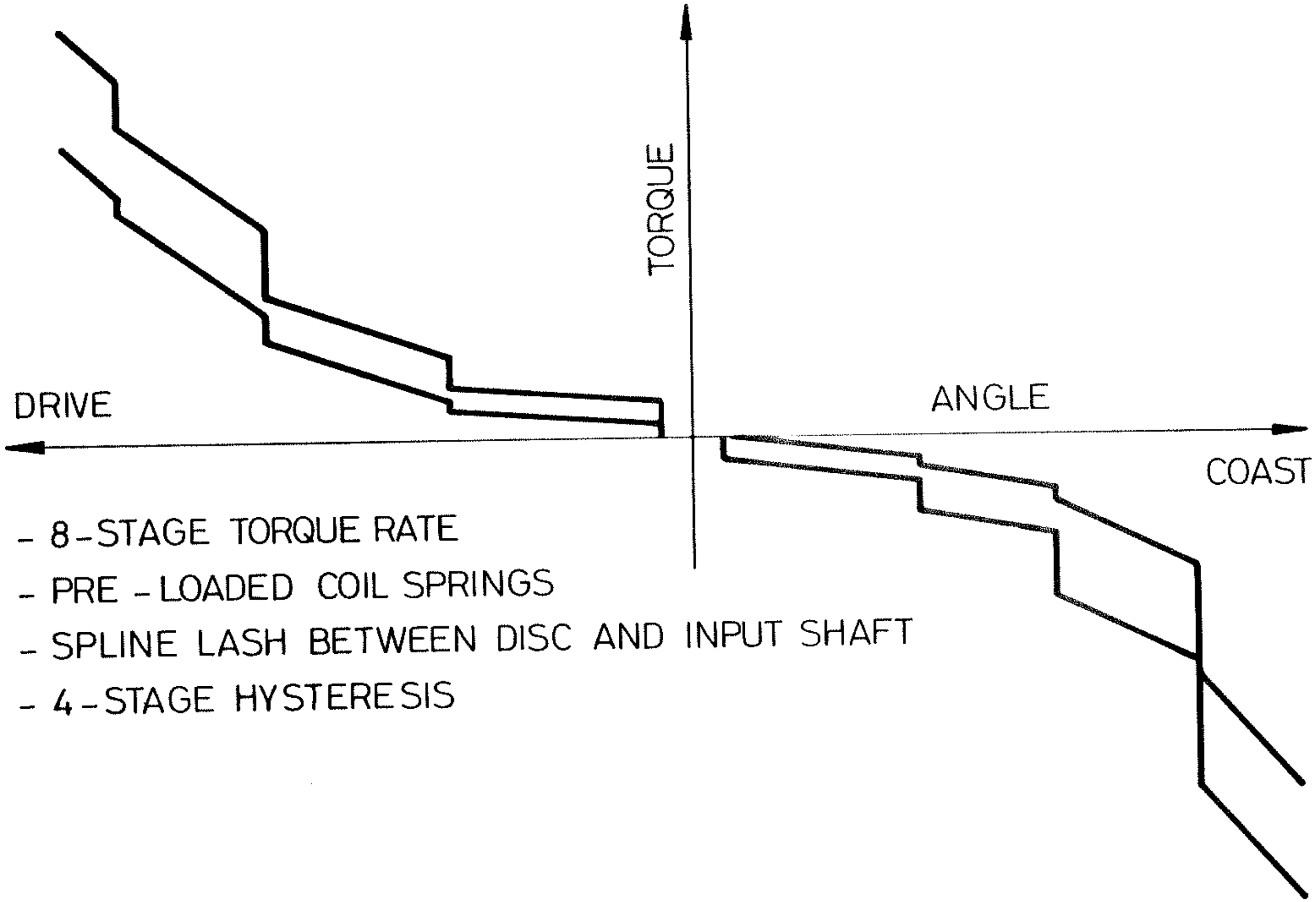
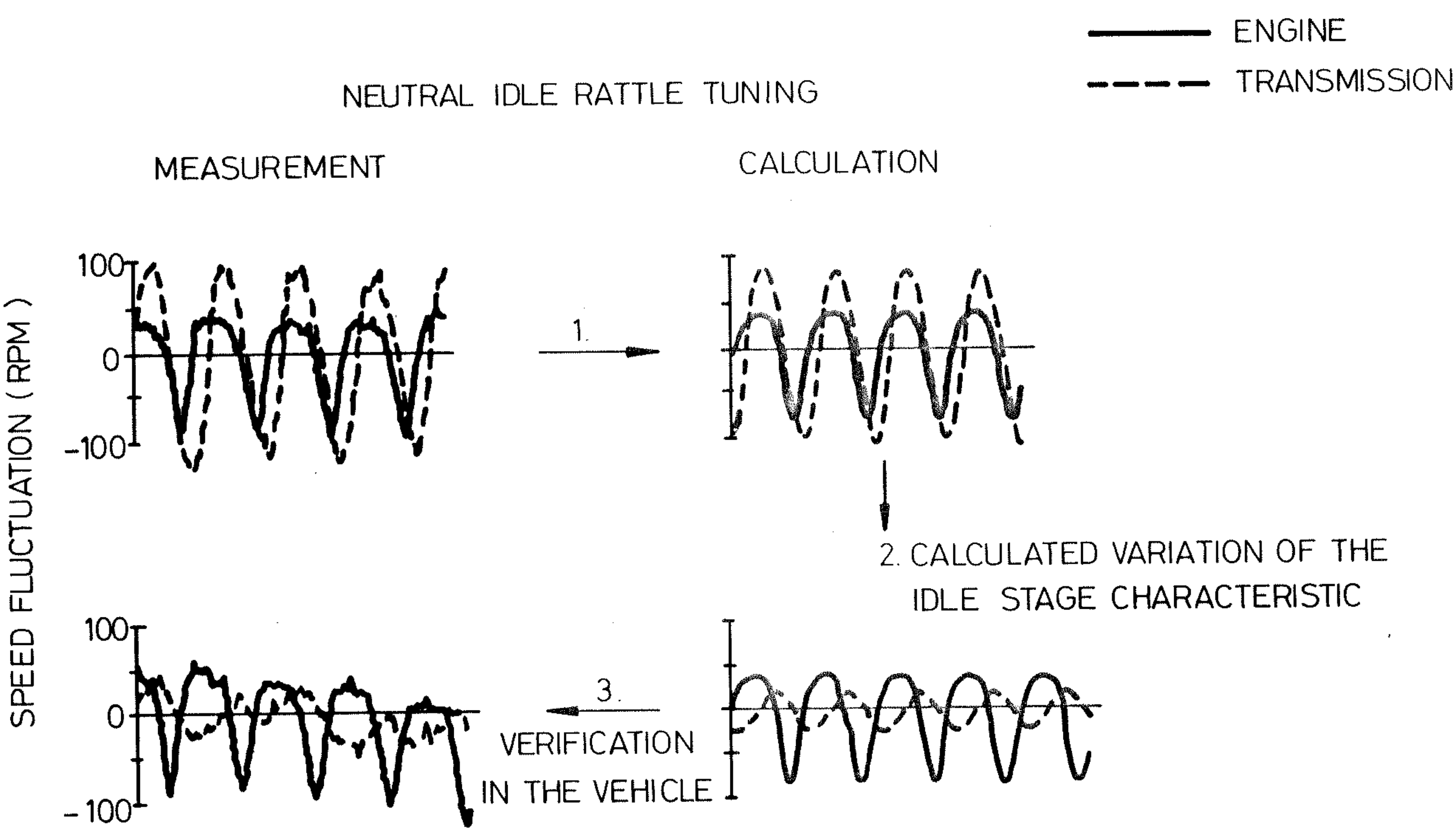
ANALYTICAL MODEL

LuK



- 8-STAGE TORQUE RATE
- PRE-LOADED COIL SPRINGS
- SPLINE LASH BETWEEN DISC AND INPUT SHAFT
- 4-STAGE HYSTERESIS

07 01 86	TORSION DAMPER CHARACTERISTIC CURVE



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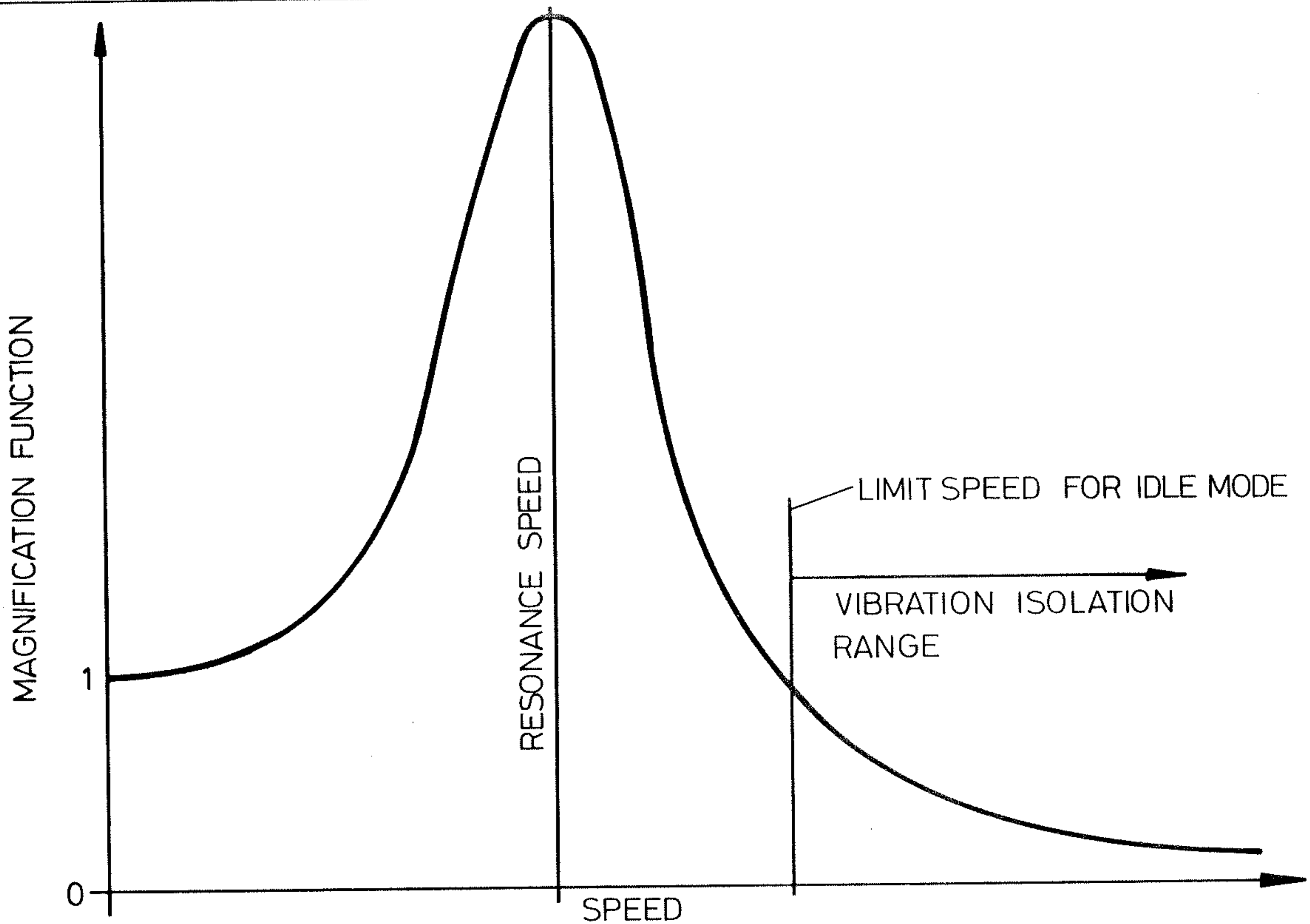
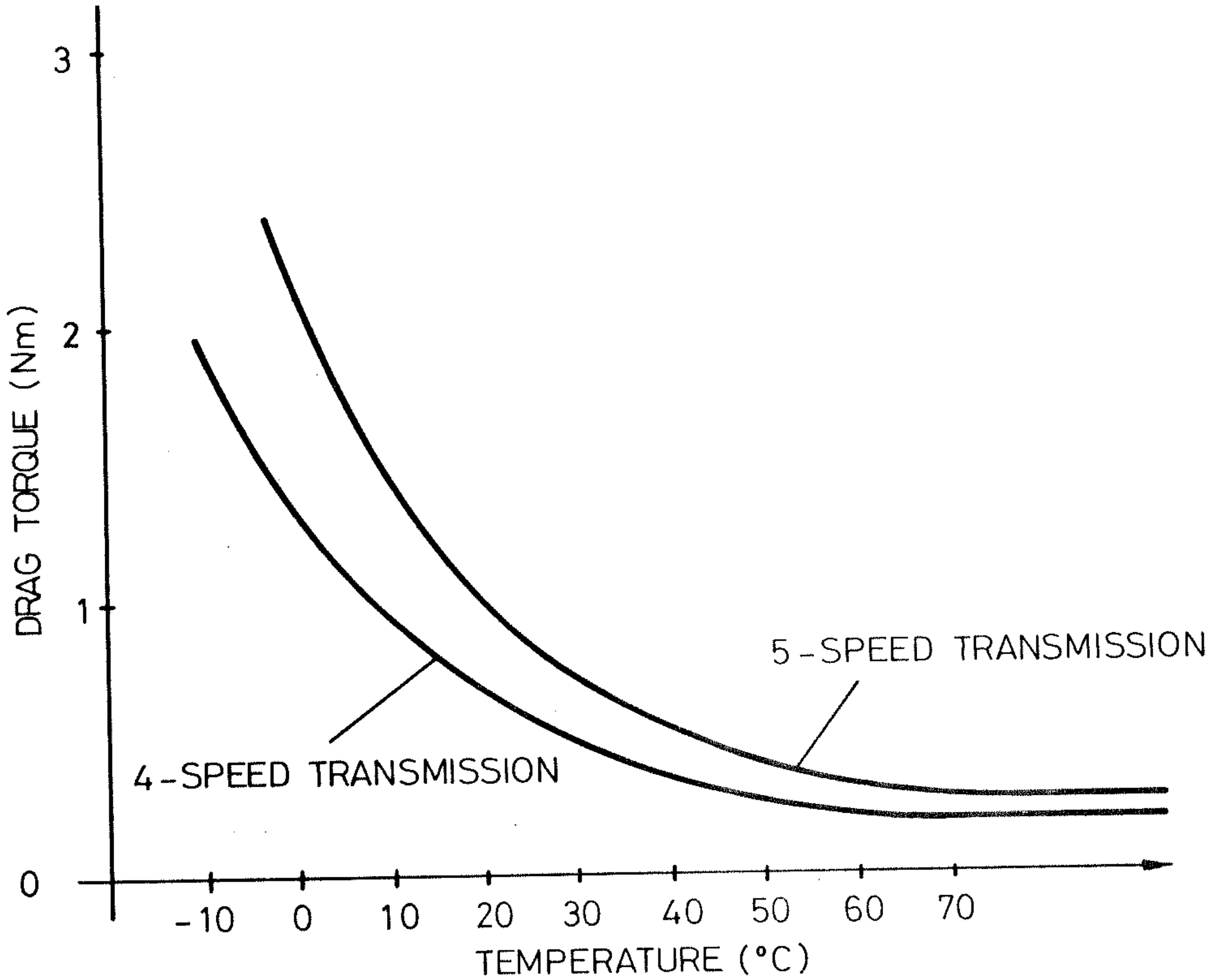
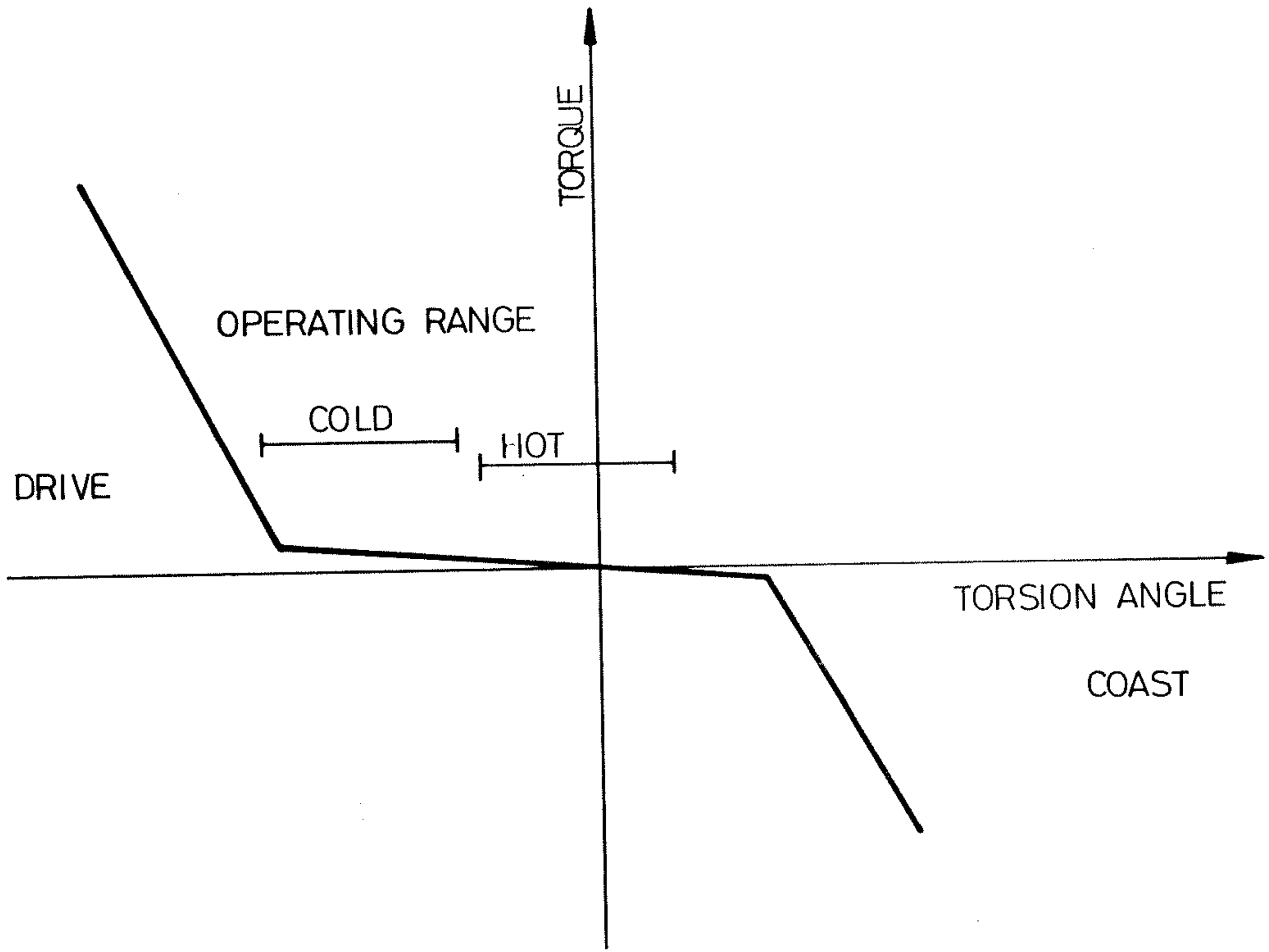
INTERACTION OF VIBRATION MEASUREMENT AND CALCULATION



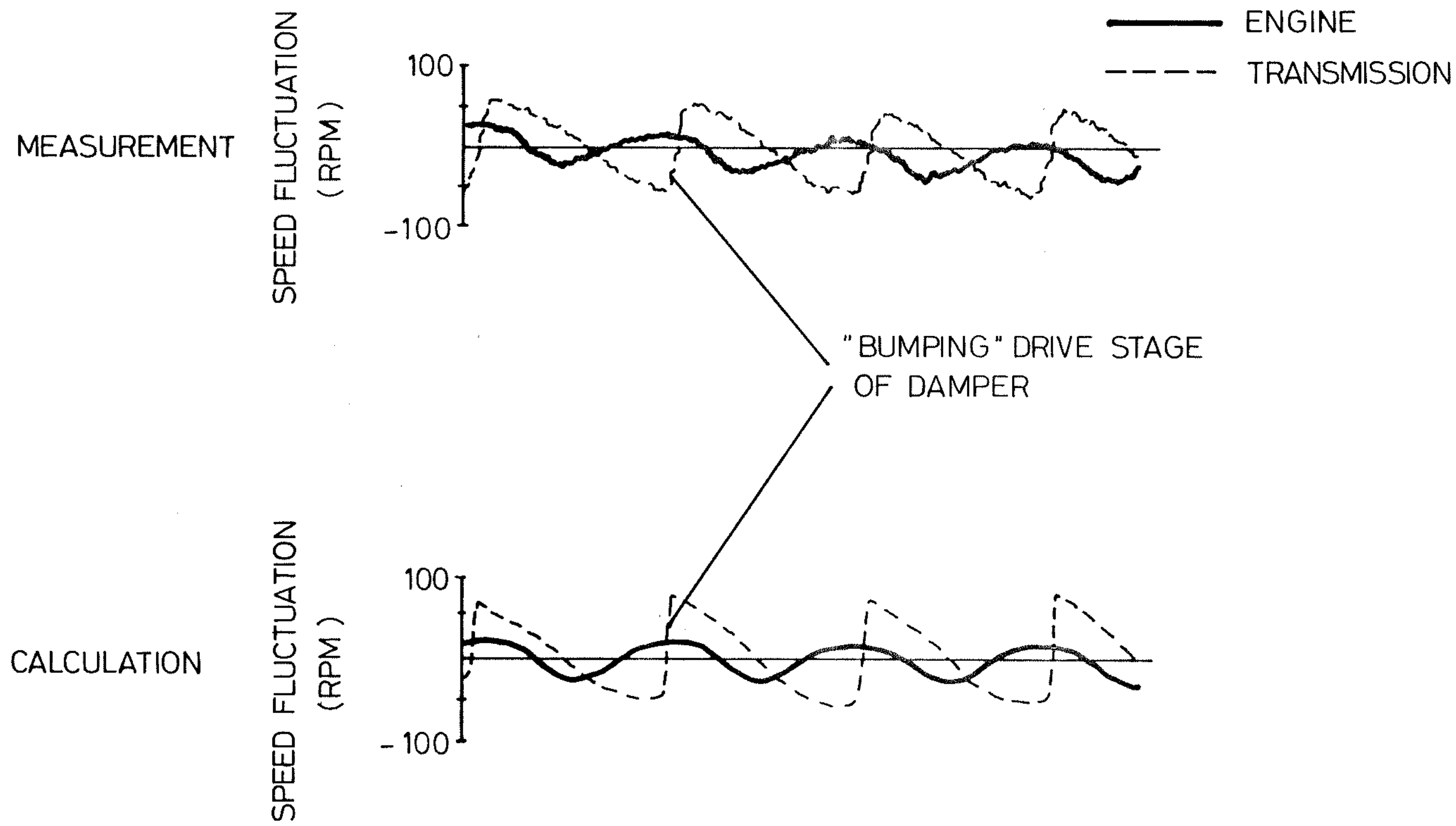
09 01 86	RESONANCE CURVE



10 01 86	DRAG TORQUE – TEMPERATURE RELATIONSHIP



11 01 86	OPERATING RANGE OF TORSION DAMPER WITH HOT AND COLD TRANSMISSION FLUID



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"BUMPING" DRIVE STAGE OF DAMPER

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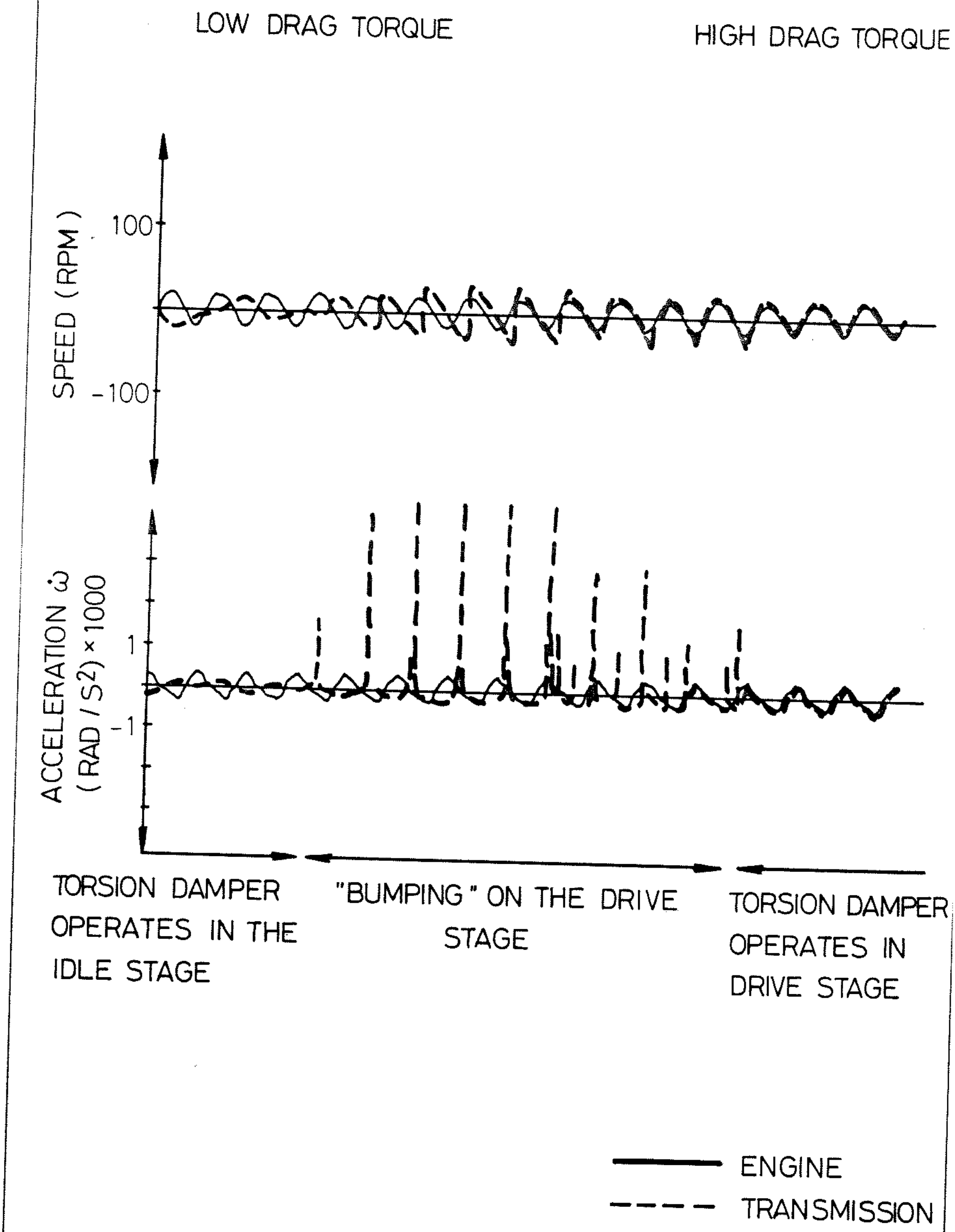
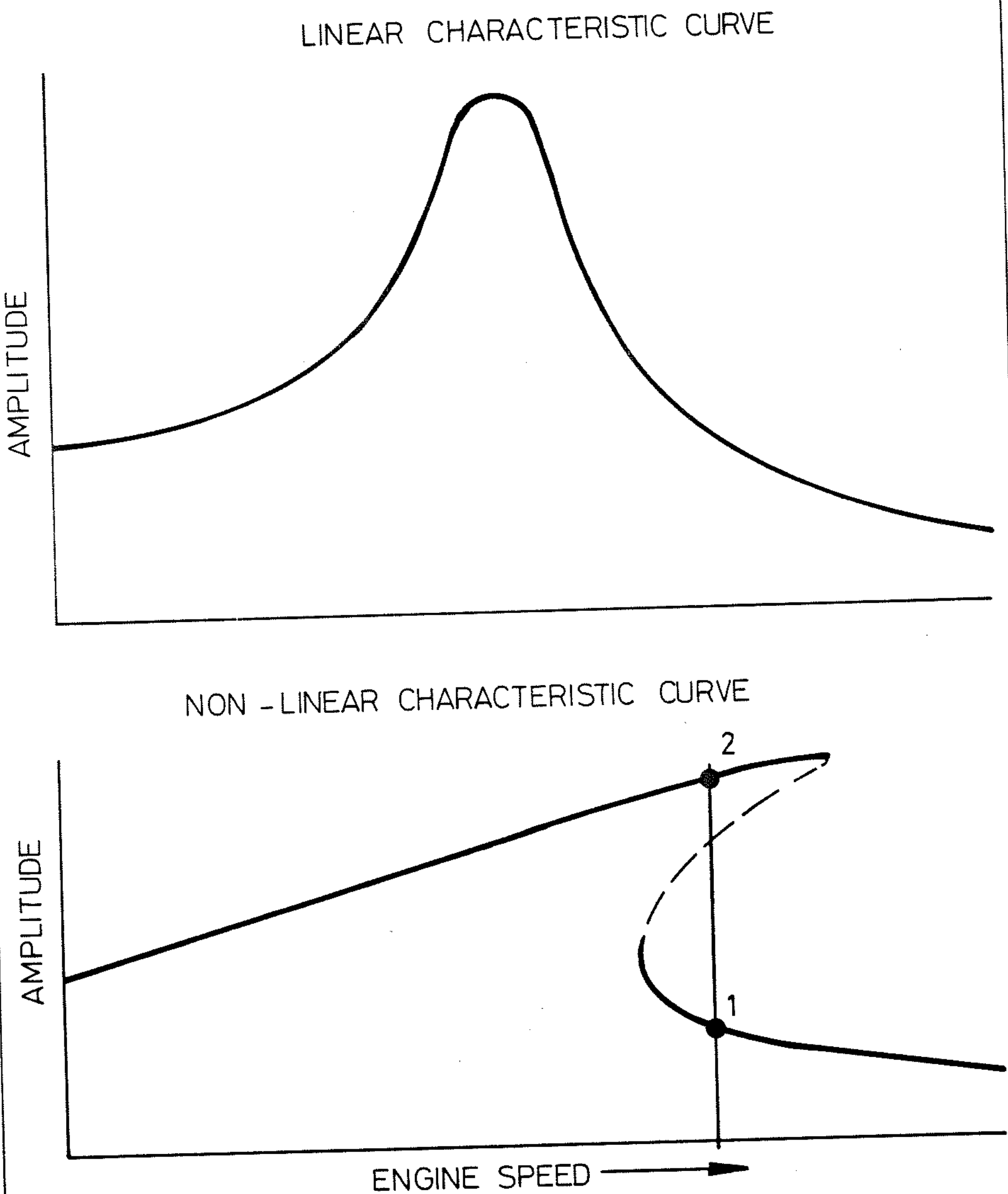


13 01 86	"BUMPING" ON THE DRIVE STAGE	LUK



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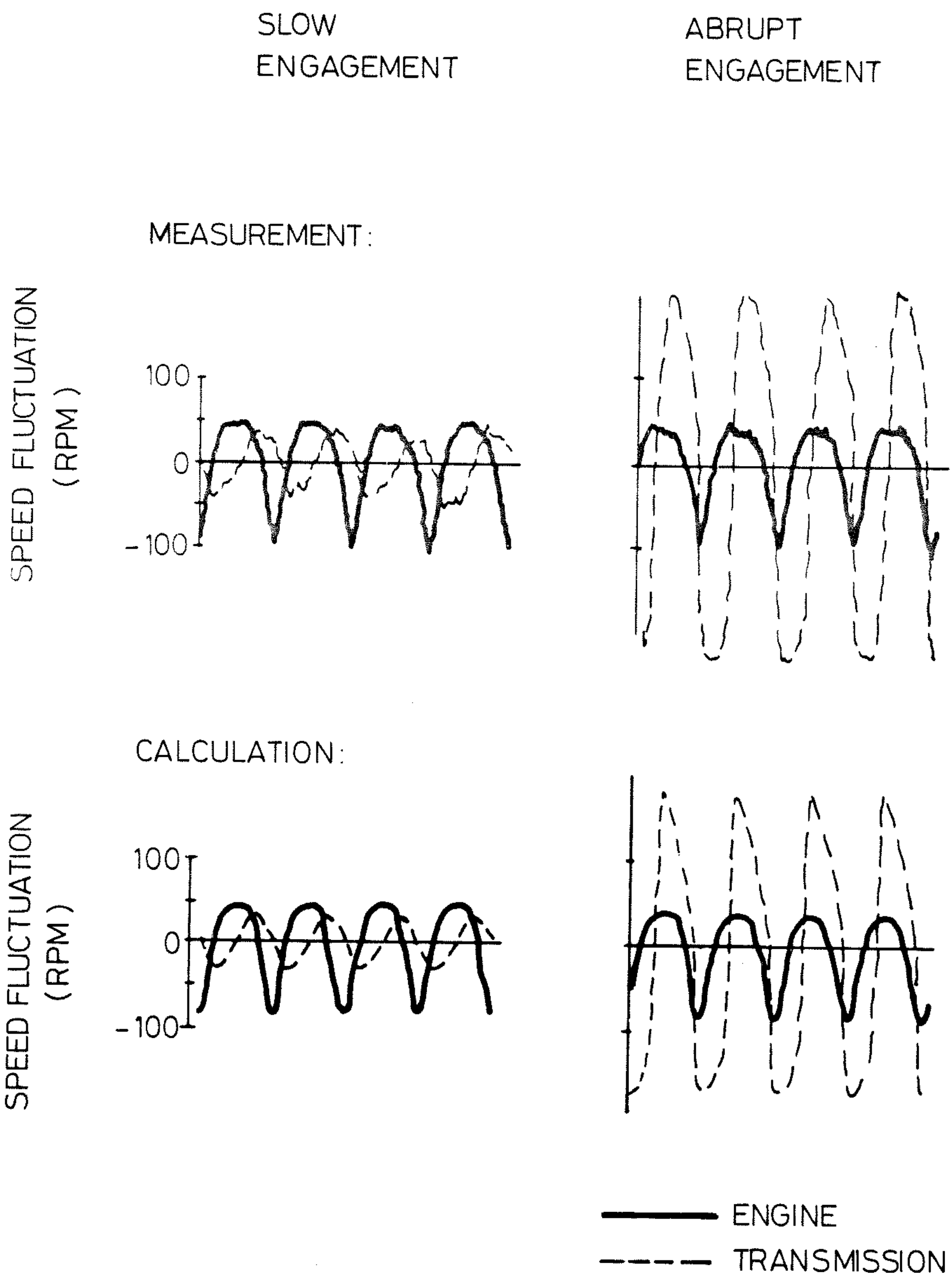
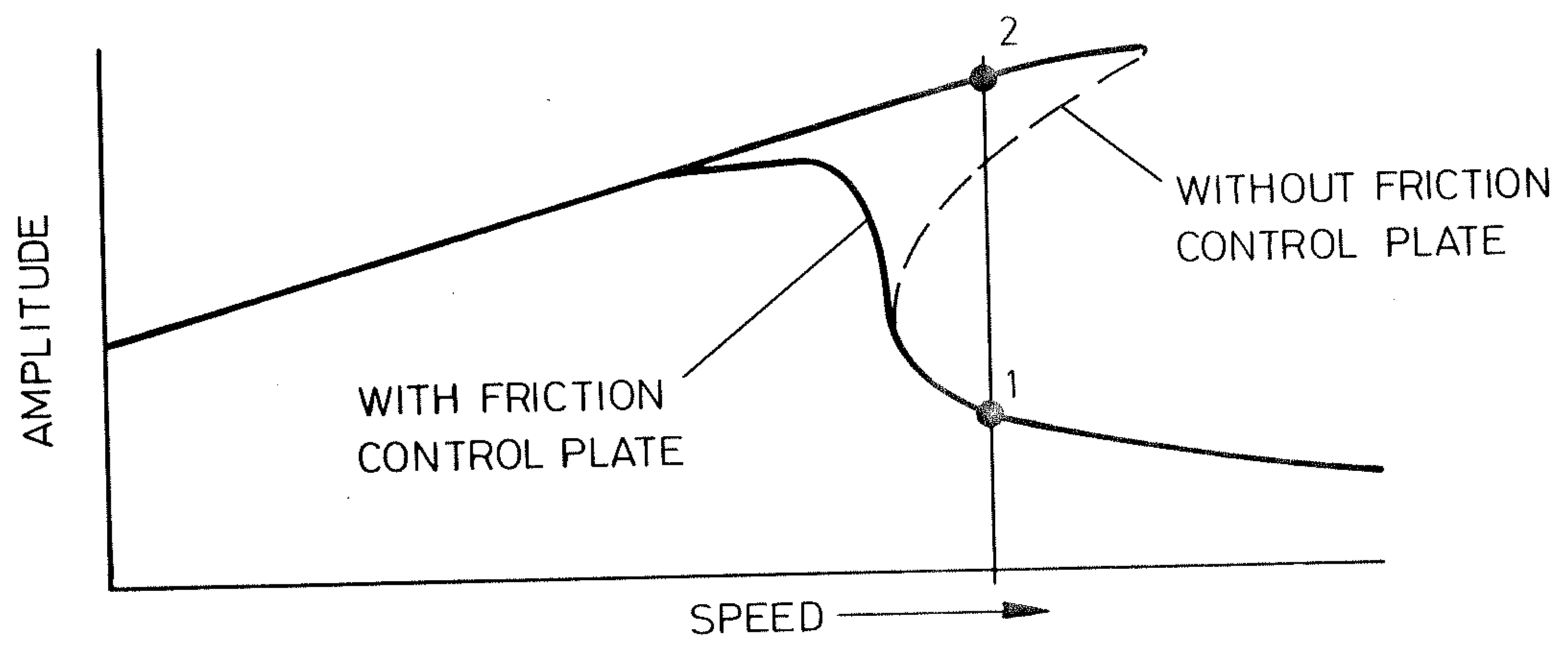
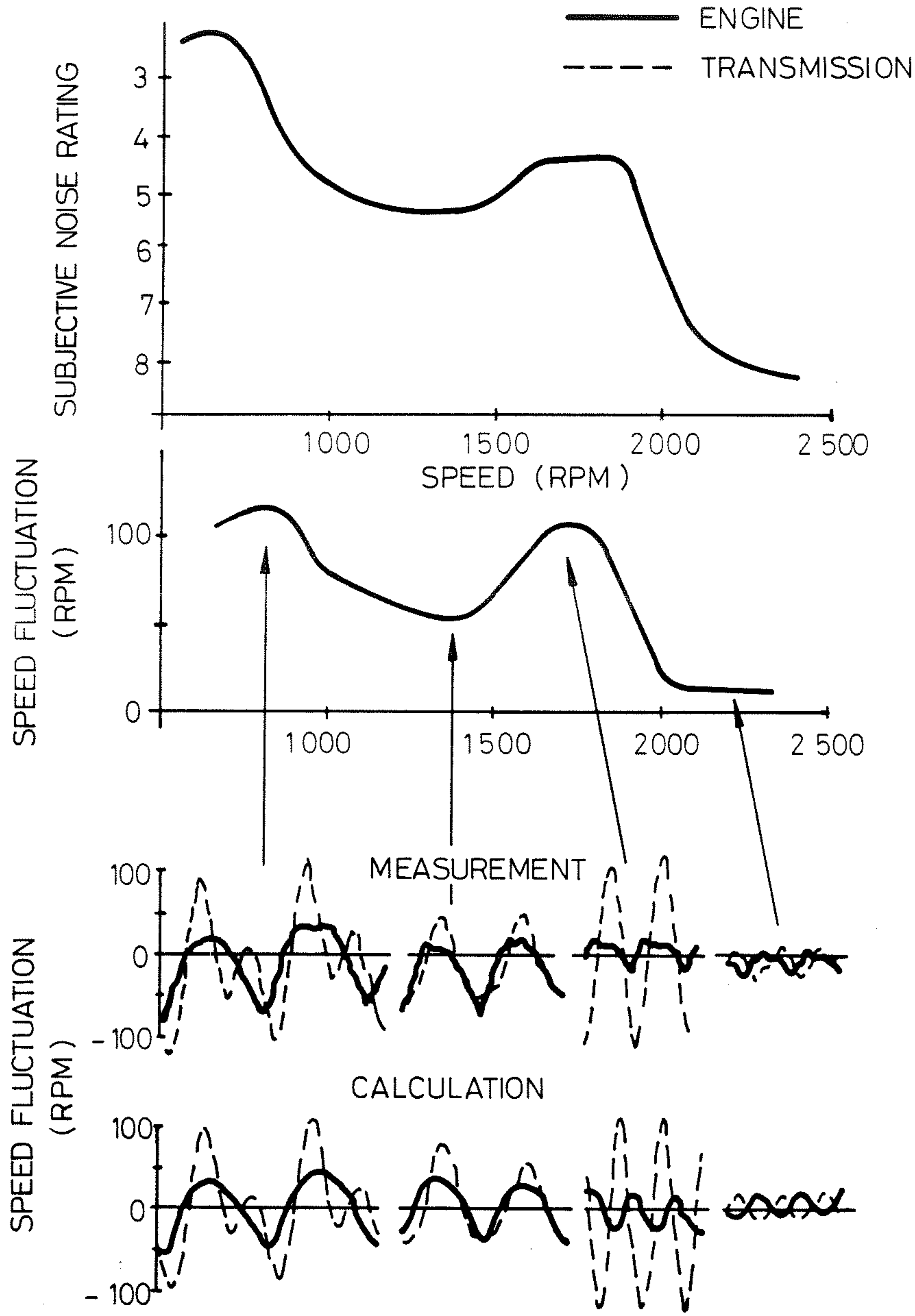
RESONANCE CURVES FOR LINEAR AND NON - LINEAR CHARACTERISTICS



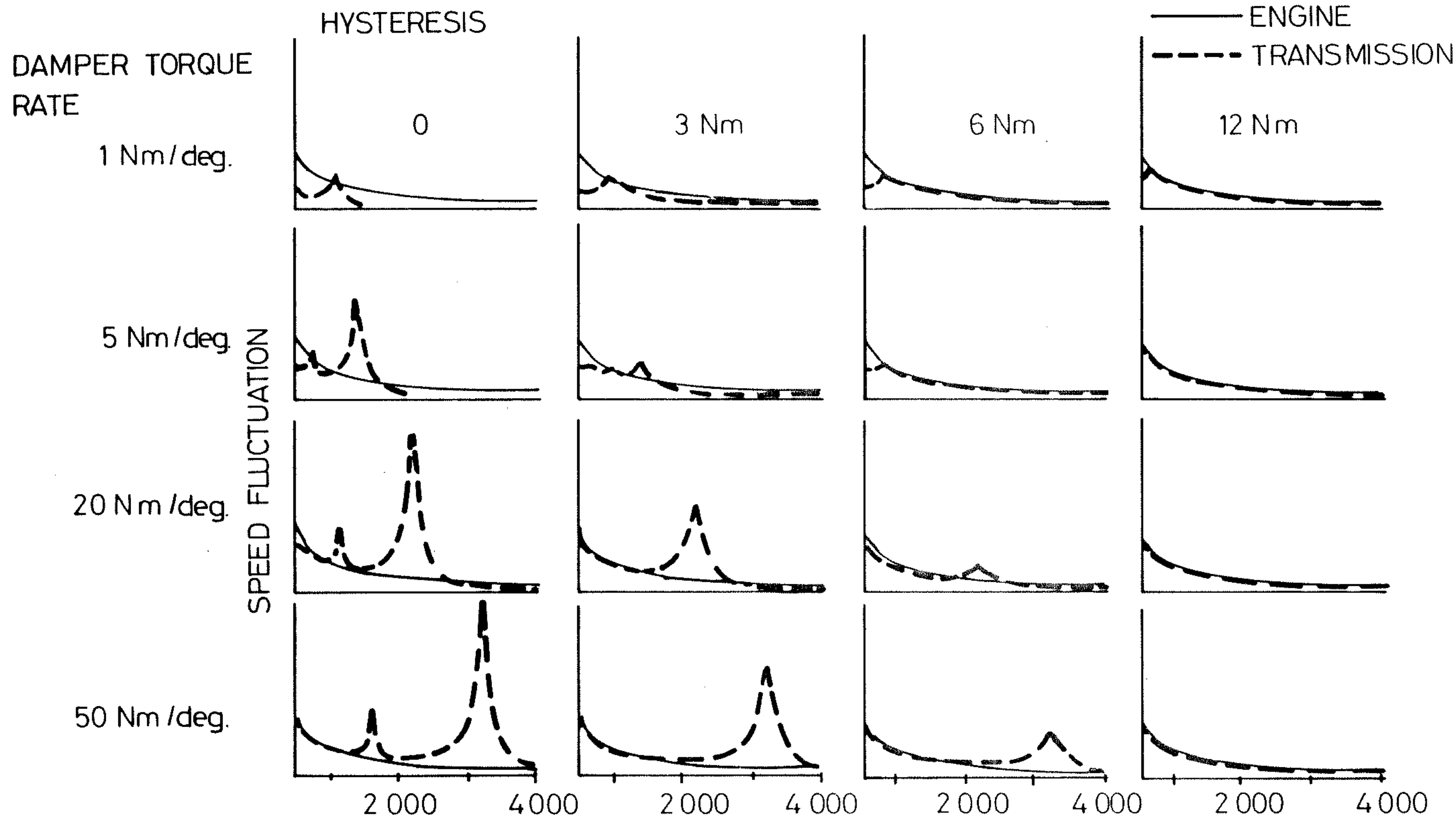
15 01 86	TORSIONAL VIBRATIONS FOR NON - LINEAR CHARACTERISTICS



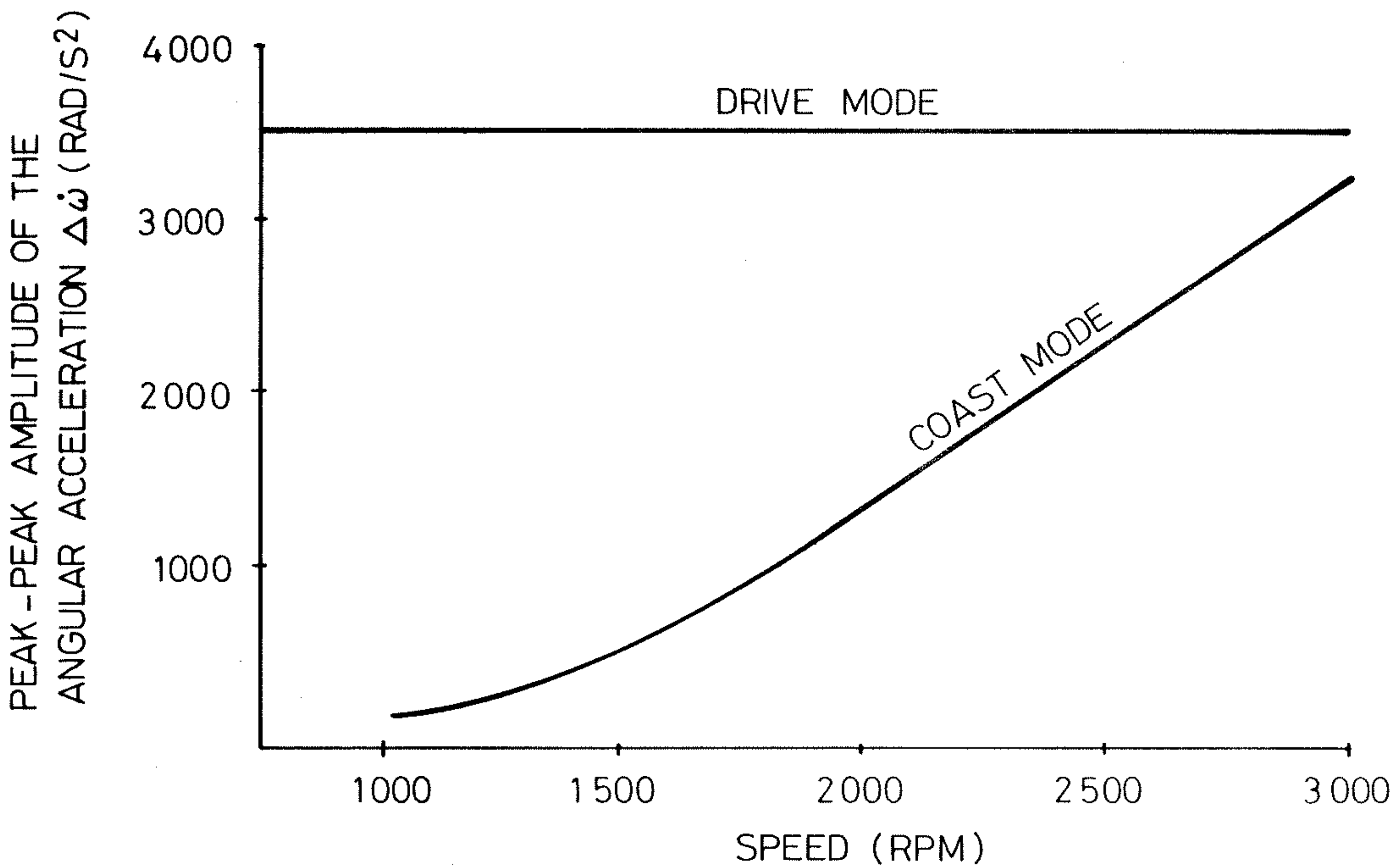
16 01 86	RESONANCE CURVES WITH AND WITHOUT FRICTION CONTROL PLATE	LUK



17 01 86	NOISE RATING AND VIBRATION AMPLITUDE IN DRIVE MODE	LUK



18 01 86	SPEED FLUCTUATIONS IN RELATIONSHIP TO THE TORQUE RATE, THE HYSTERESIS, AND THE ENGINE SPEED	LUK



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COMPARISON OF ENGINE EXCITATION IN DRIVE AND COAST MODES

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THE DUAL MASS FLYWHEEL

DR.-ING. L.F. SCHULTE

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THE DUAL MASS FLYWHEEL

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THE DUAL MASS FLYWHEEL

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.

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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.

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 fly-wheel, 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.

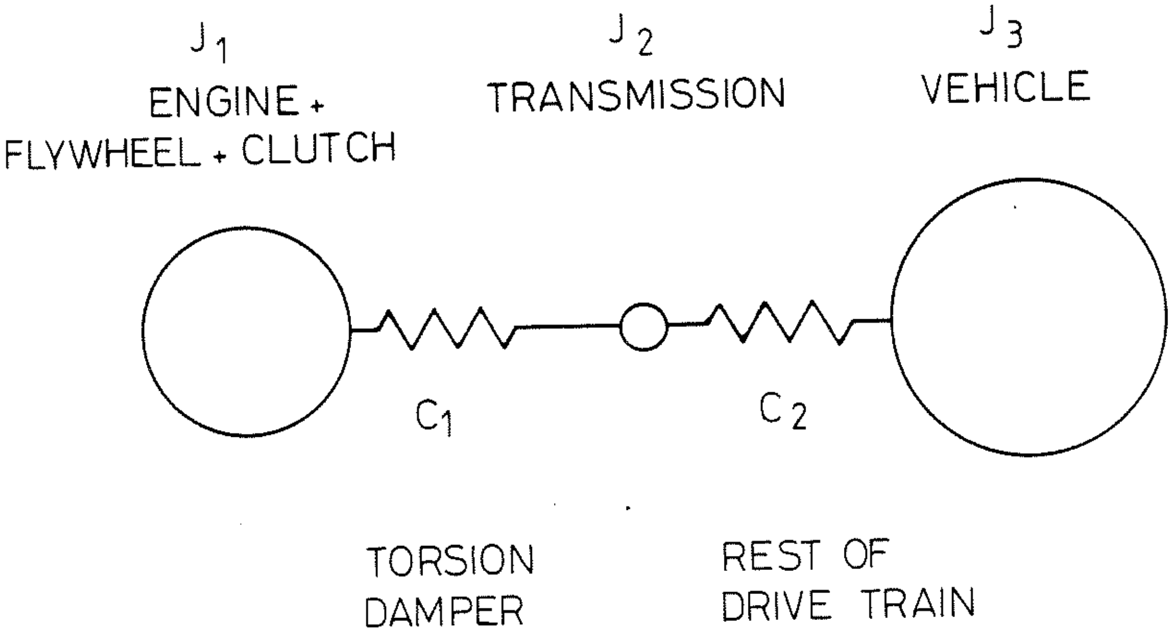
Based on several years' experience we are convinced that the introduction of the dual mass flywheel can be used to eliminate existing transmission rattle problems, in spite of the need to optimize the design.

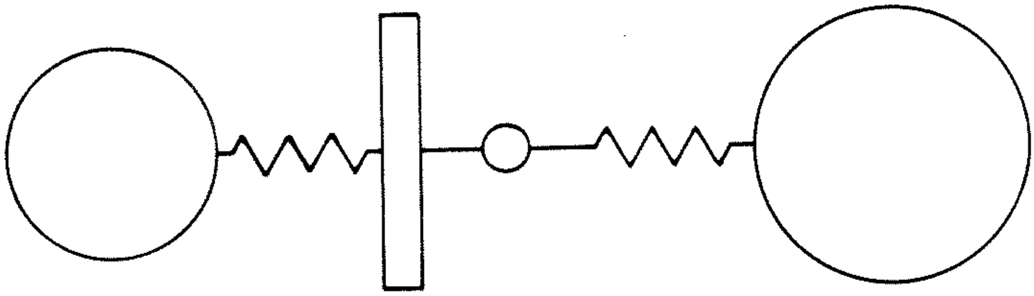
In contrast to fluid couplings, these improvements can be made without significant loss in efficiency. On the contrary, it appears that fuel consumption will be improved because it will be more comfortable to drive at lower engine speeds.

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.

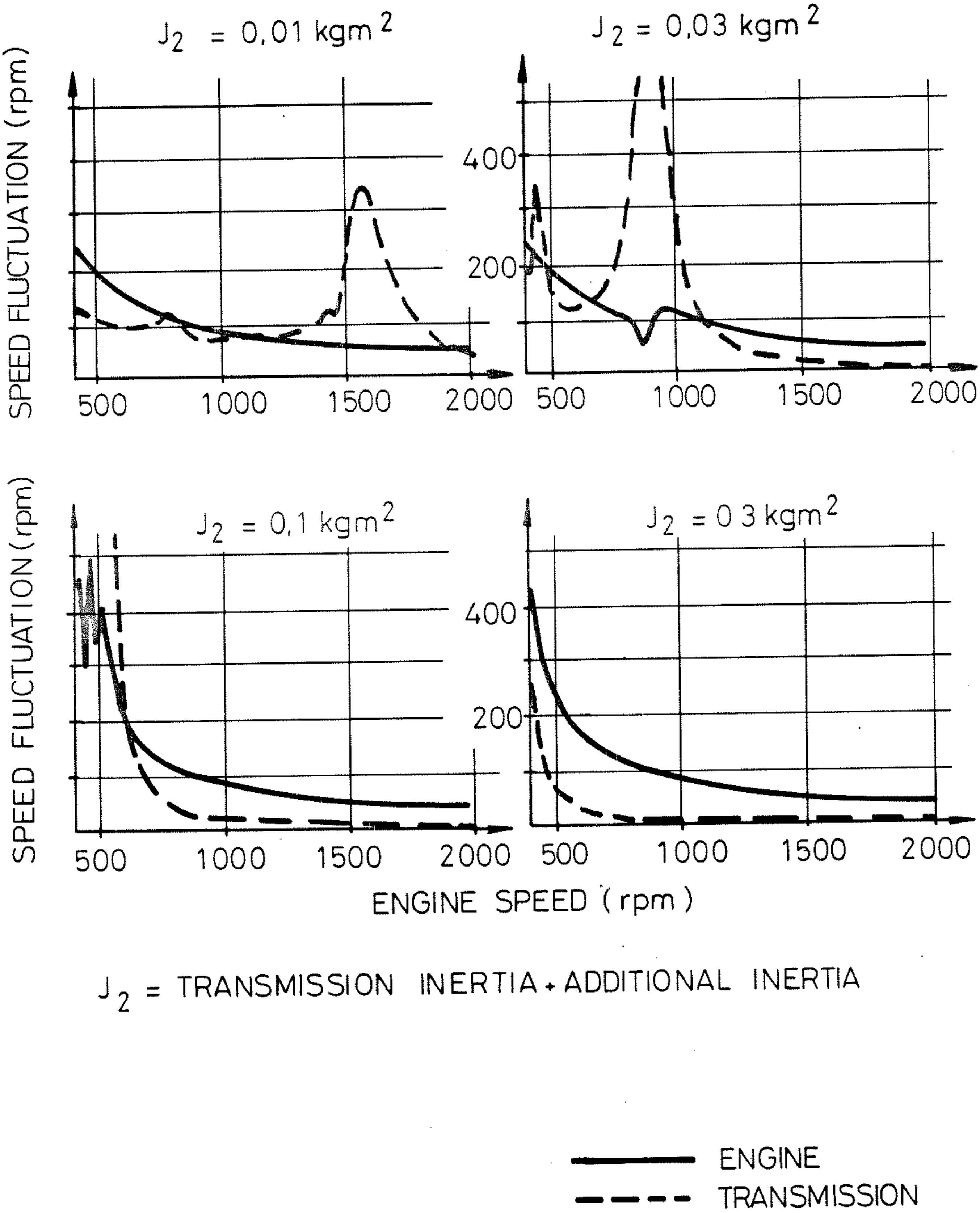
Initial understanding of the design also leads us to believe that the introduction of the dual mass flywheel would help to reduce the "sophistication" of the transmission, improve shift-ability and increase transmission service life.





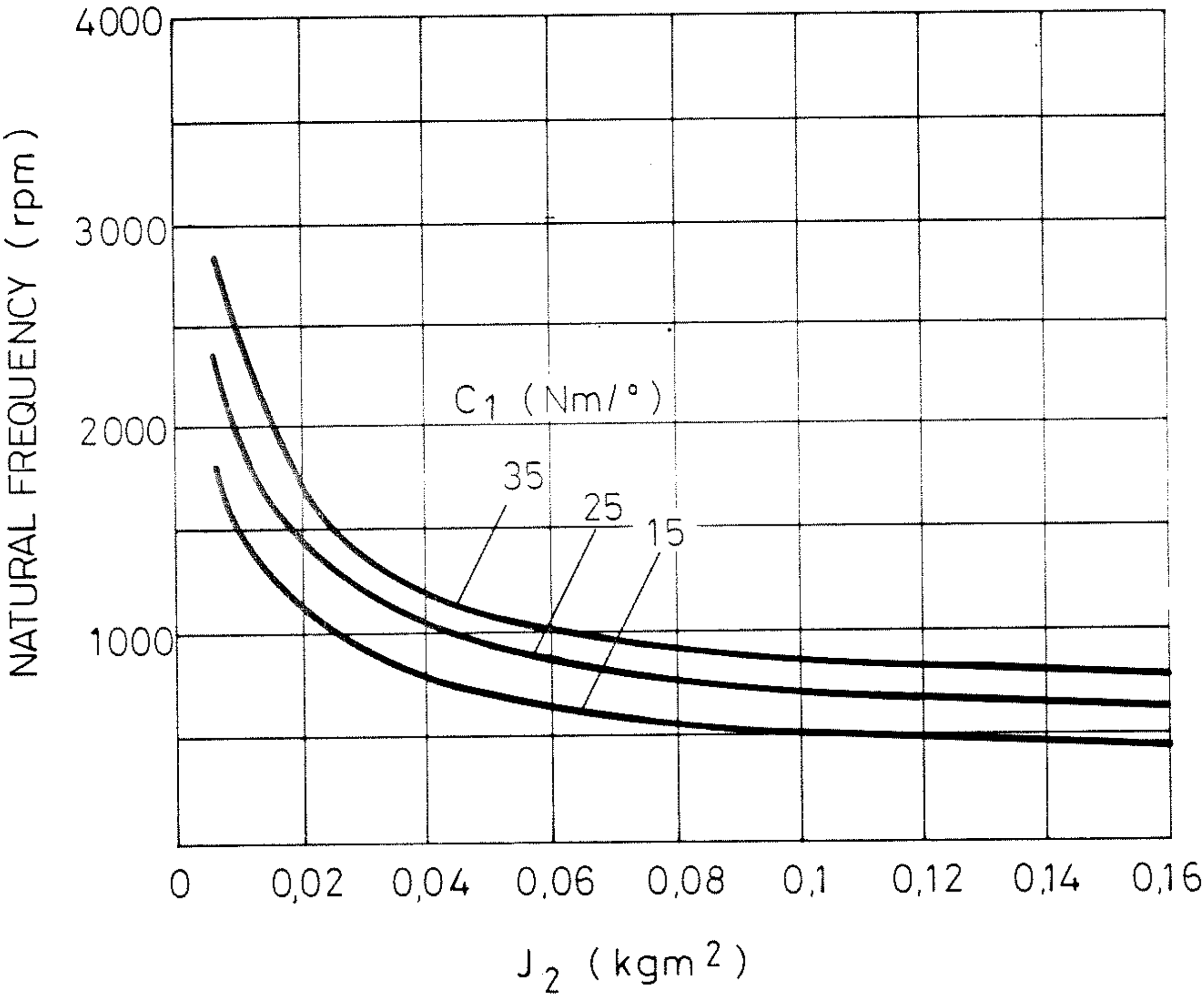
ADDITIONAL INERTIA
ON TRANSMISSION
INPUT SHAFT

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



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

LUK

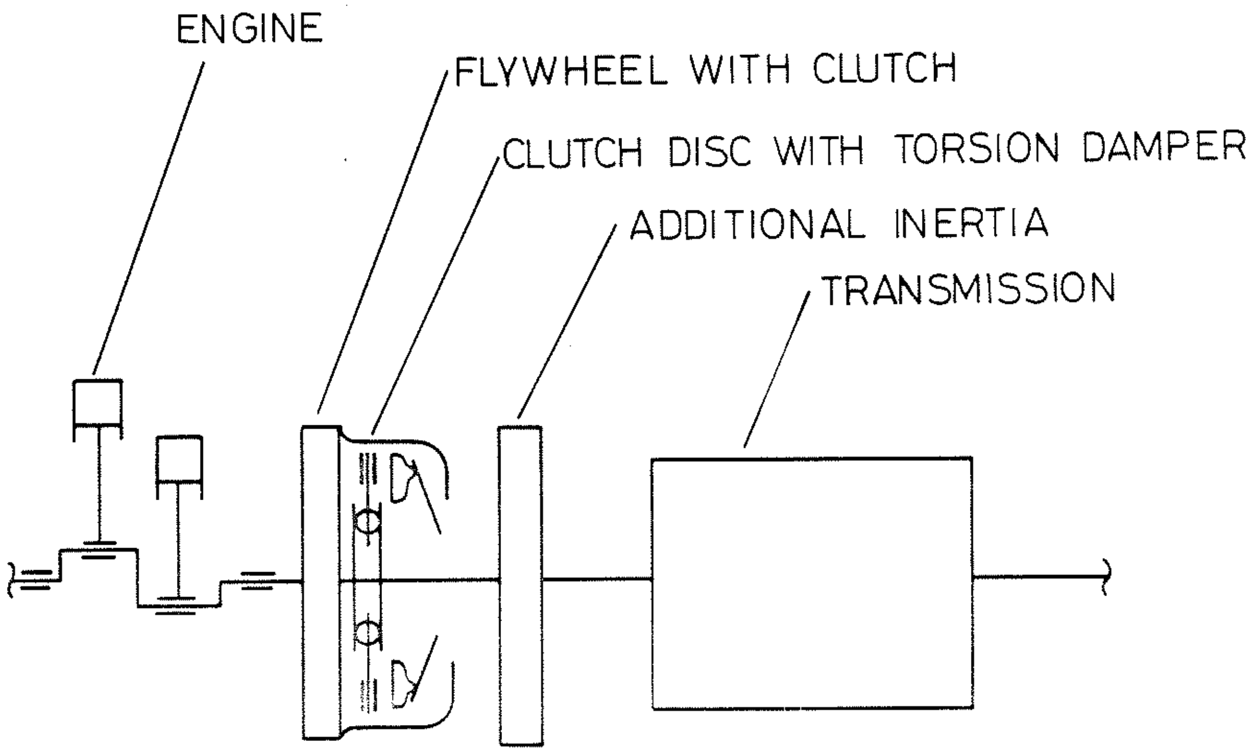


C_1 = DAMPER TORSION RATE

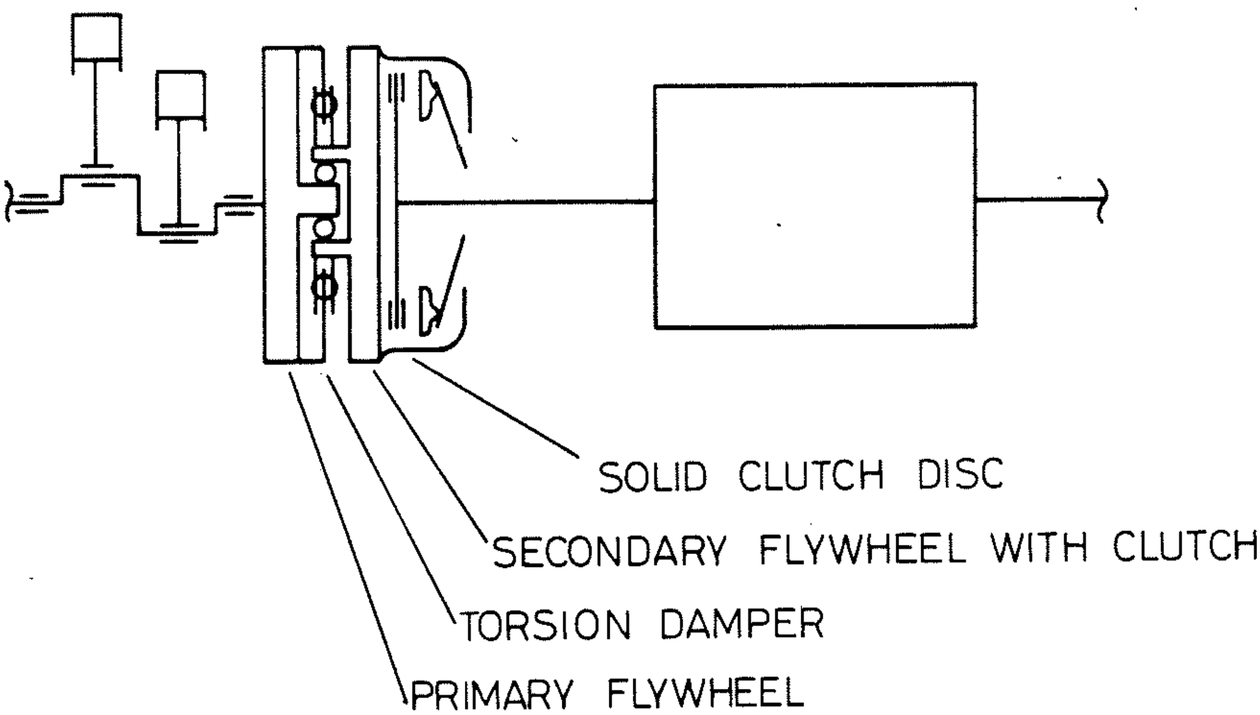
J_2 = TRANSMISSION INERTIA + ADDITIONAL INERTIA

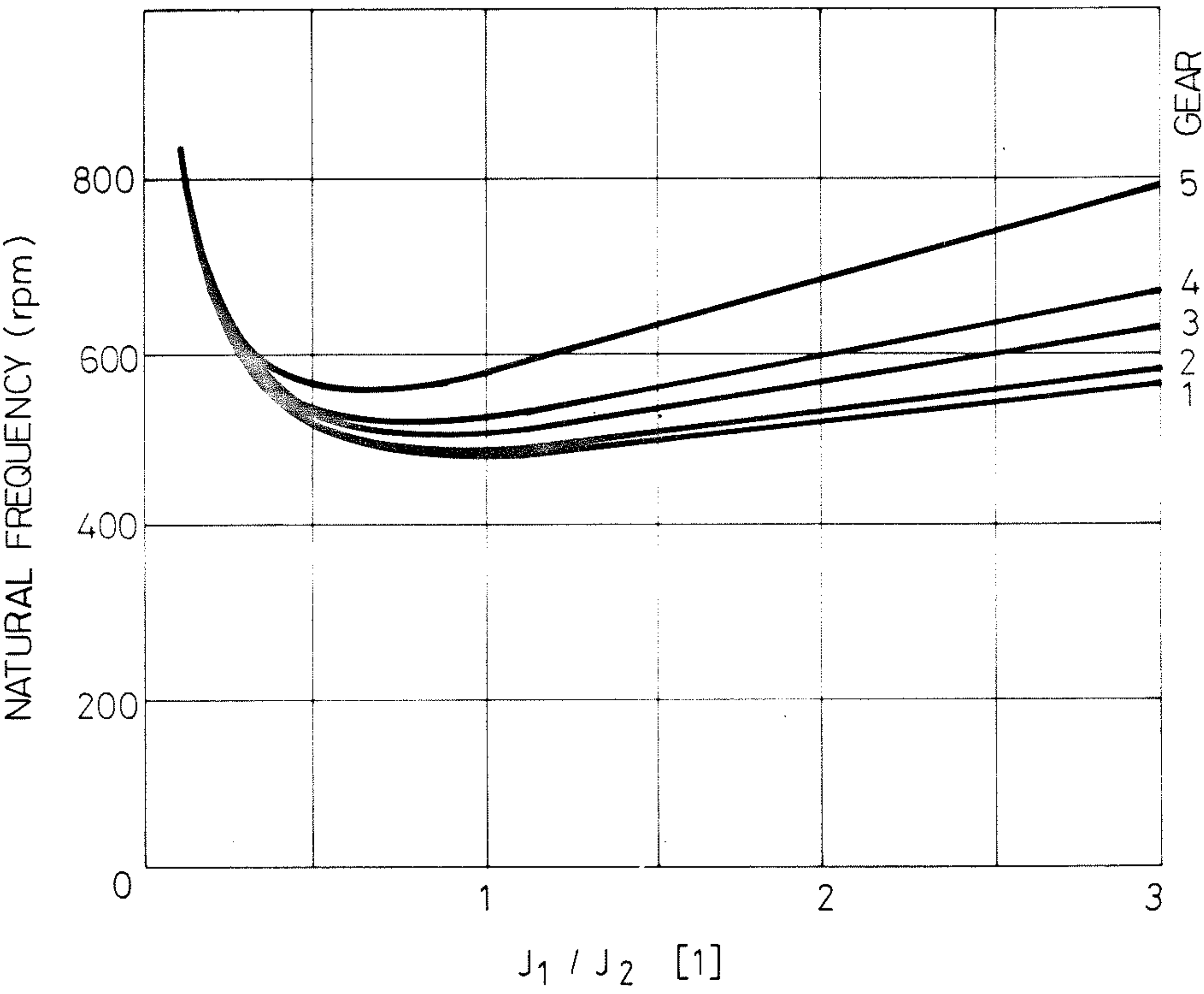
03 02 86	NATURAL FREQUENCY DEPENDENT UPON THE TORSION RATE AND THE INERTIA	LUK

ADDITIONAL INERTIA ON THE TRANSMISSION INPUT SHAFT



DUAL MASS FLYWHEEL





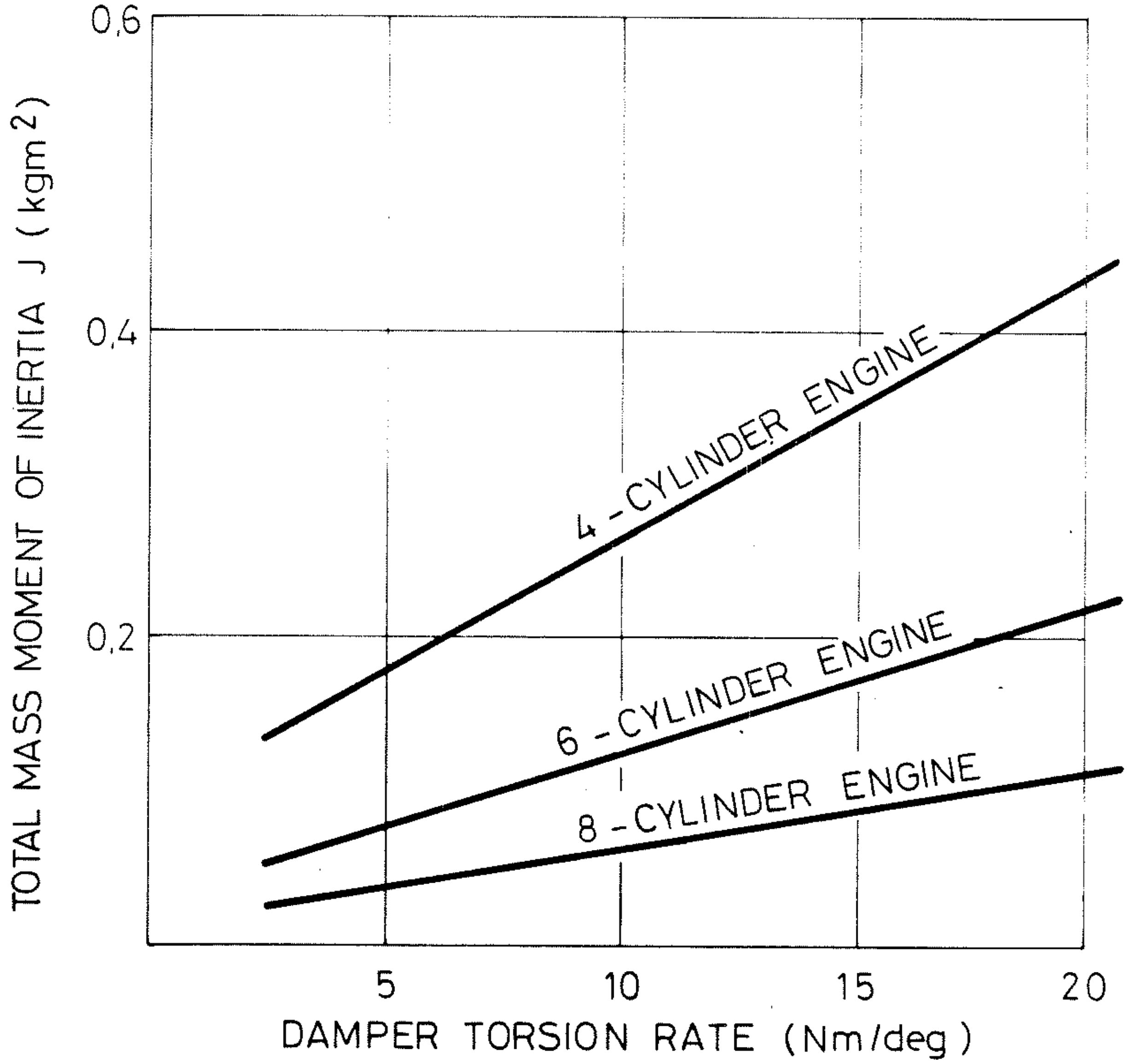
J_1 = MASS MOMENT OF INERTIA OF THE COMPONENTS AHEAD OF THE TORSION DAMPER

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

$J_1 + J_2$ = CONSTANT

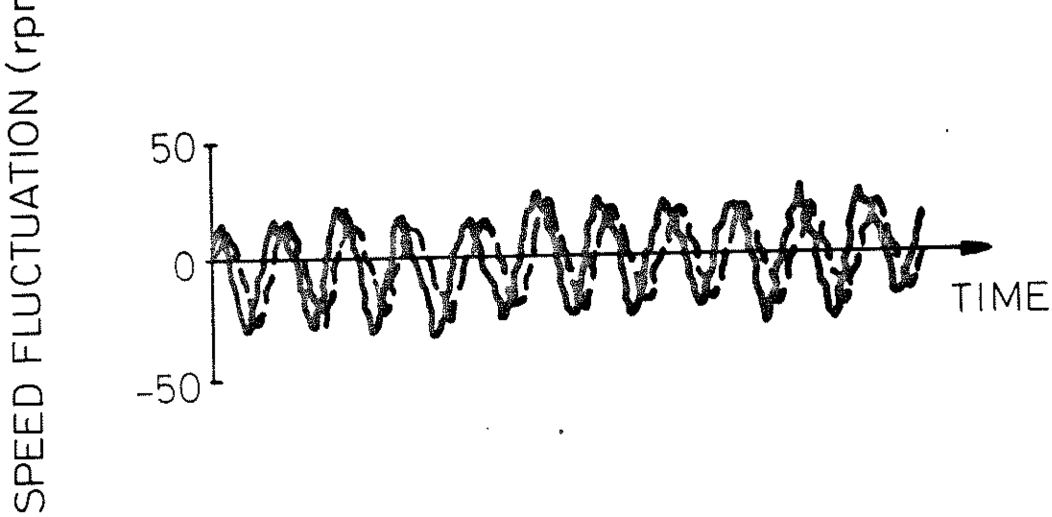
05 02 86	EFFECT OF INERTIA DISTRIBUTION ON THE NATURAL FREQUENCY	LUK

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.)

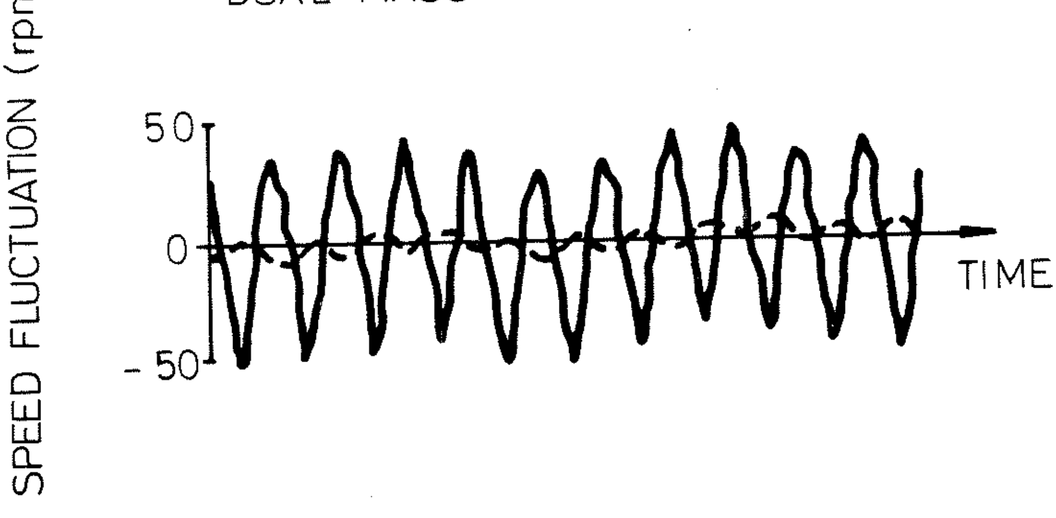


06 02 86	TOTAL REQUIRED MASS MOMENT OF INERTIA

CONVENTIONAL CLUTCH DISC



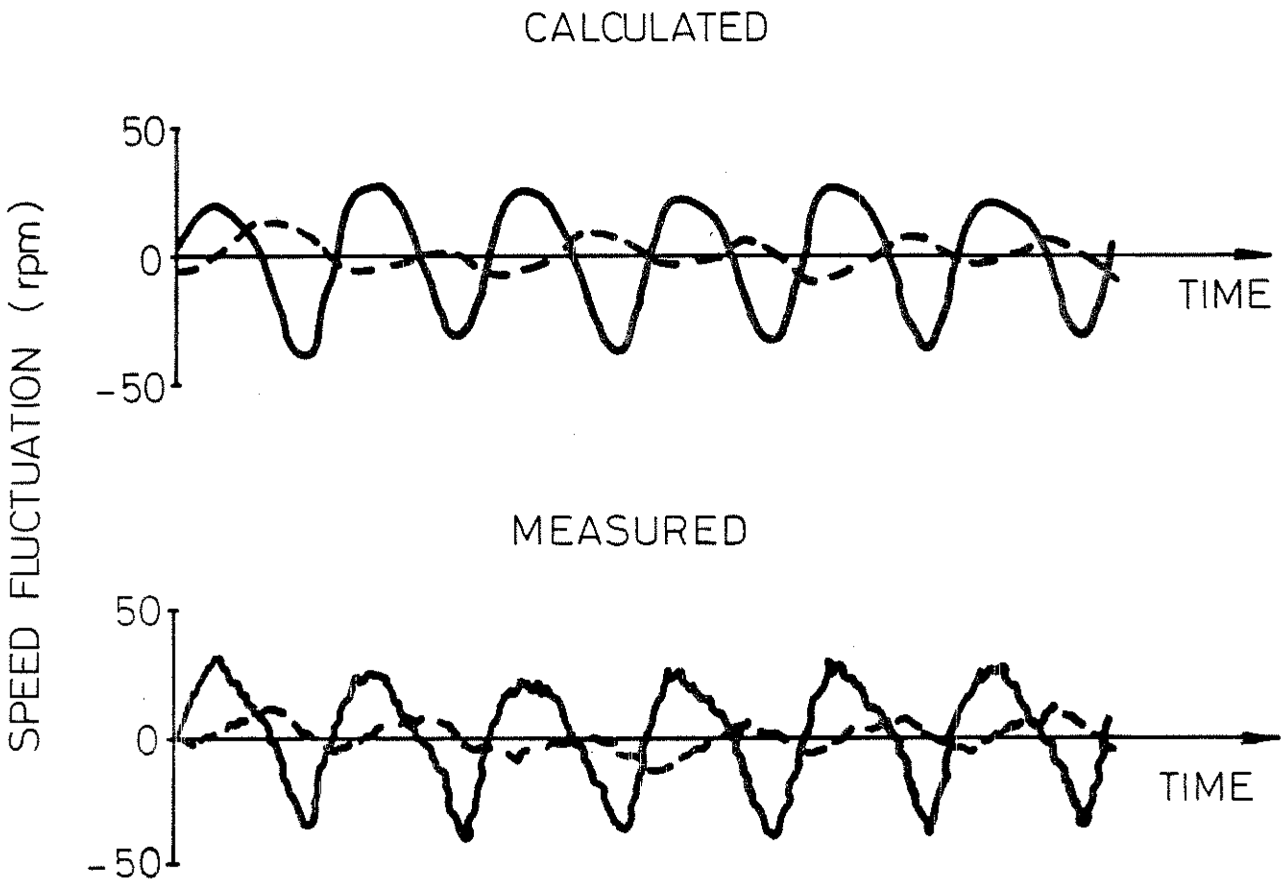
DUAL MASS FLYWHEEL



ENGINE SPEED 800 RPM

——— ENGINE
– – – TRANSMISSION

07 02 86	MEASUREMENT OF TORSIONAL VIBRATIONS IN DRIVE MODE	LUK



$J_1 = 0{,}140\ \mathrm{kgm}^2$

$J_2 = 0{,}120\ \mathrm{kgm}^2$

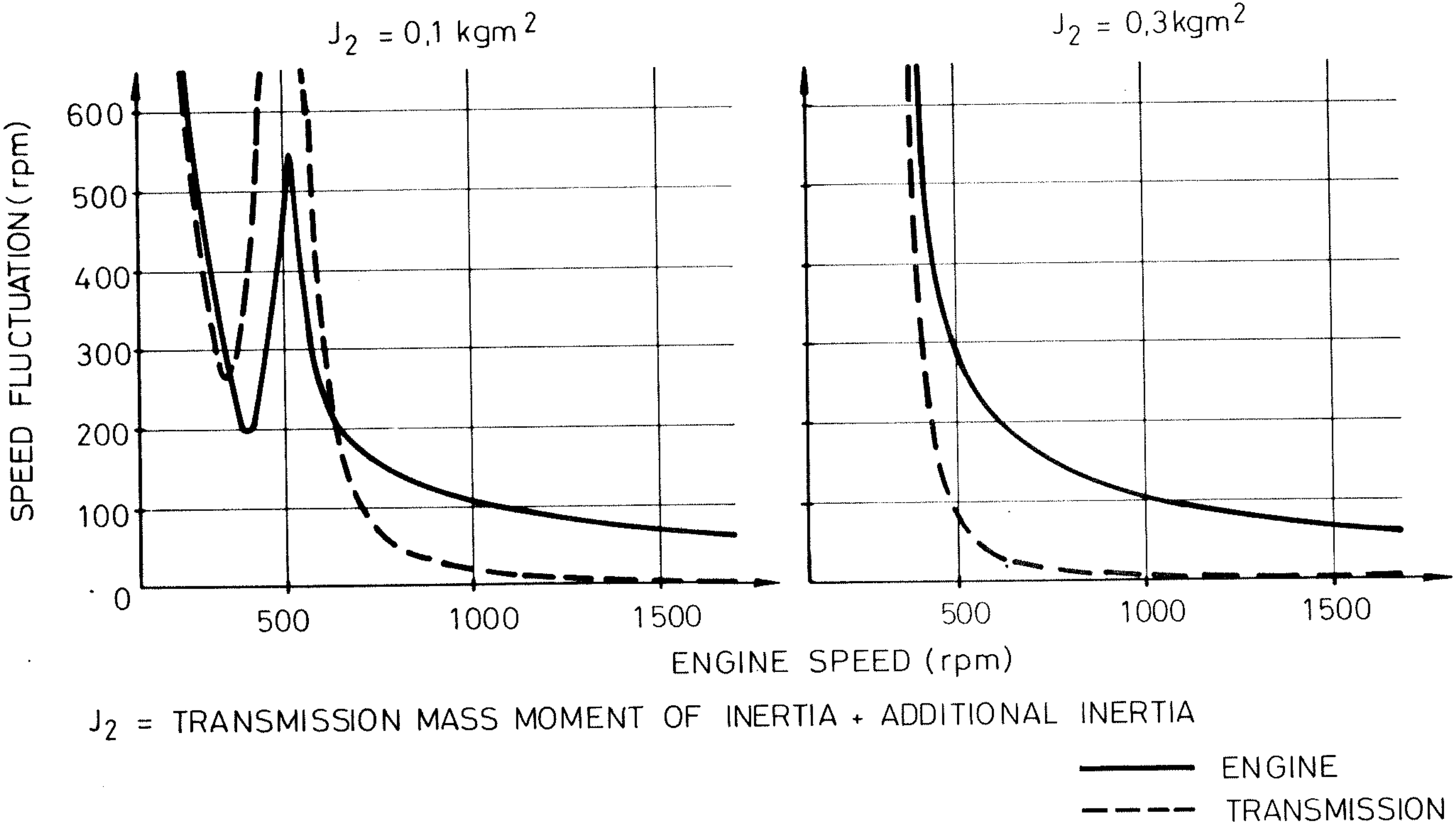
$C_1 = 3{,}3\ \mathrm{Nm/°}$

Hy. = 10 Nm

——— ENGINE

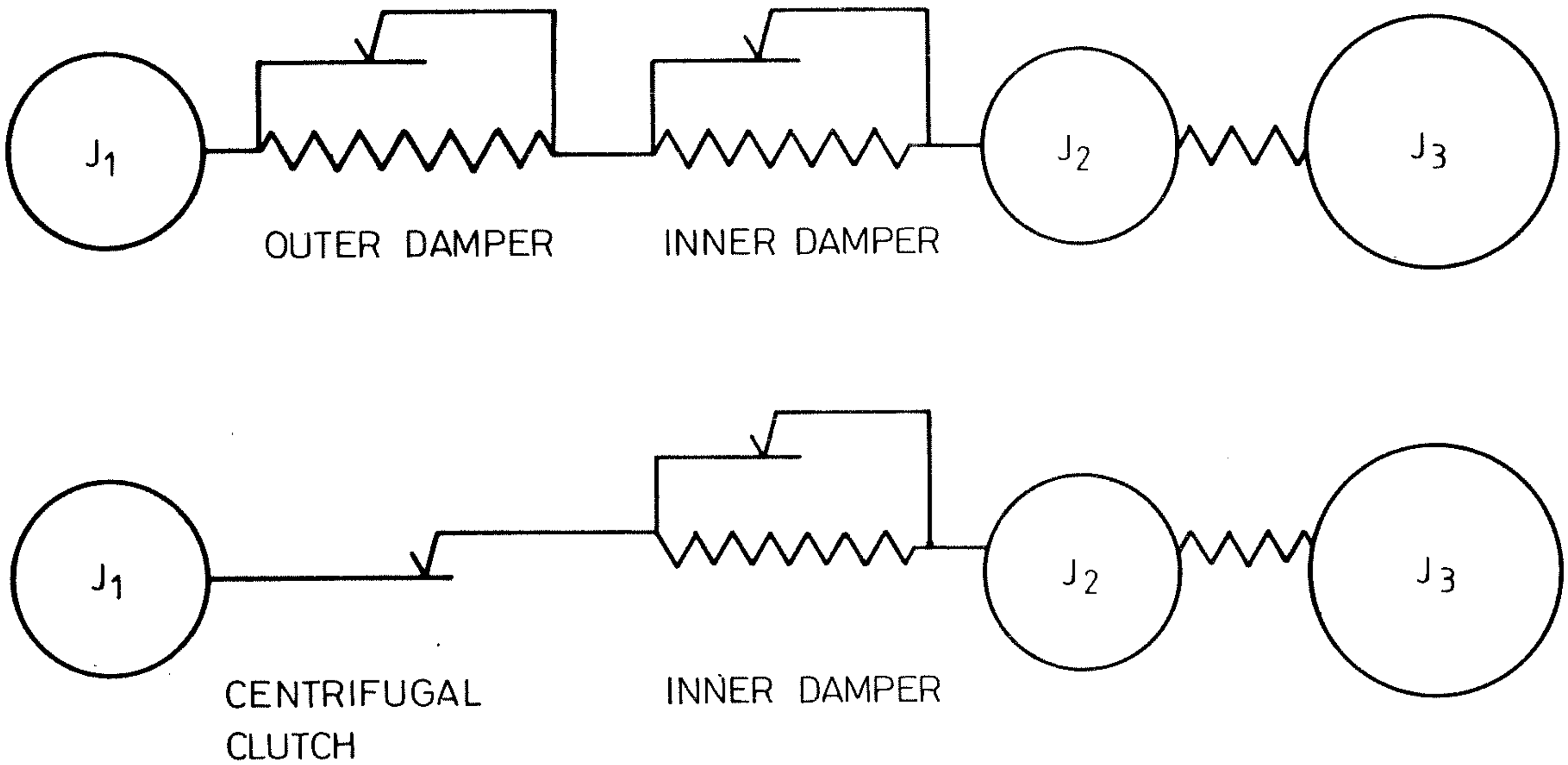
- - - - TRANSMISSION

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



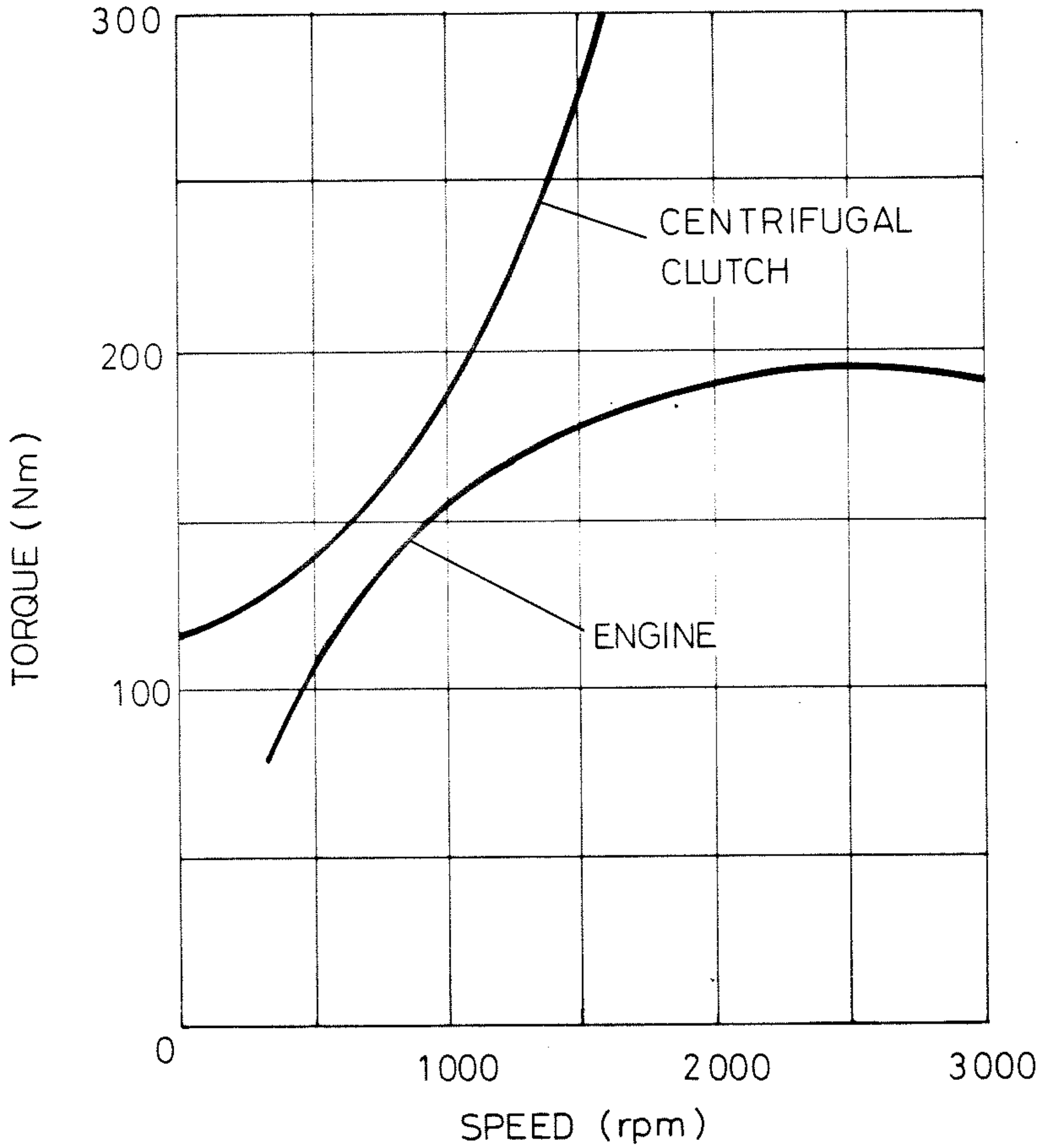
J_2 = TRANSMISSION MASS MOMENT OF INERTIA + ADDITIONAL INERTIA

09 02 86	RESONANCE CURVE



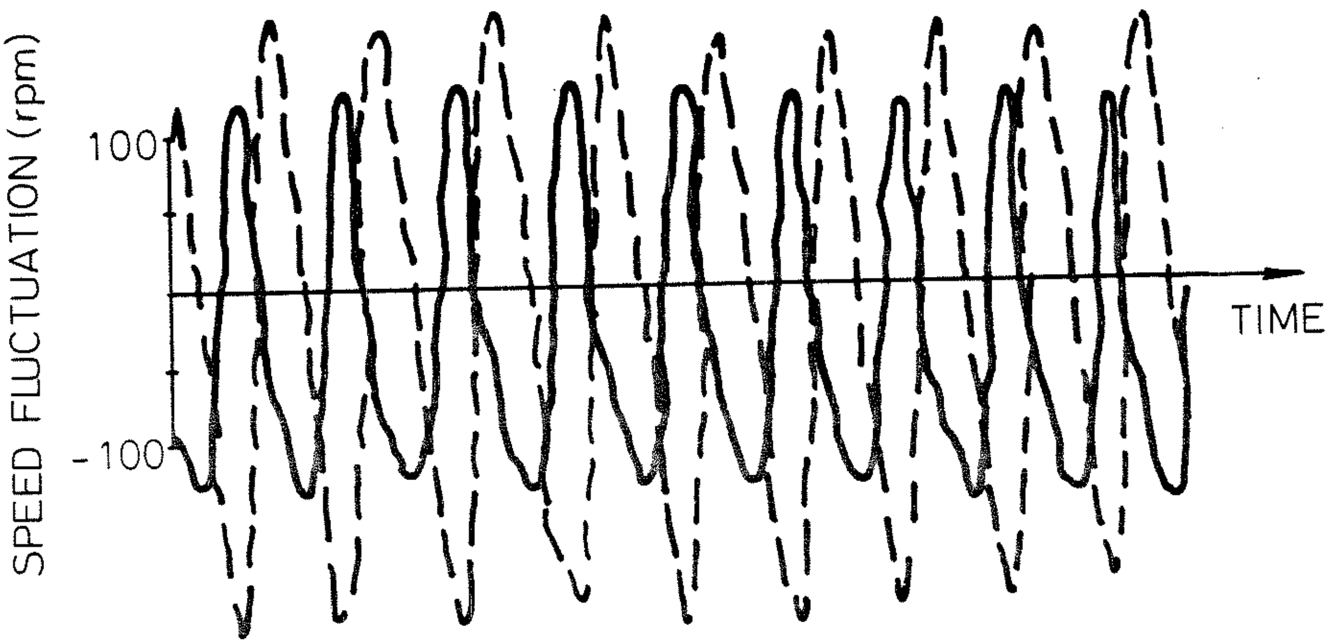
10 02 86

DAMPER ARRANGEMENT IN THE DMFW

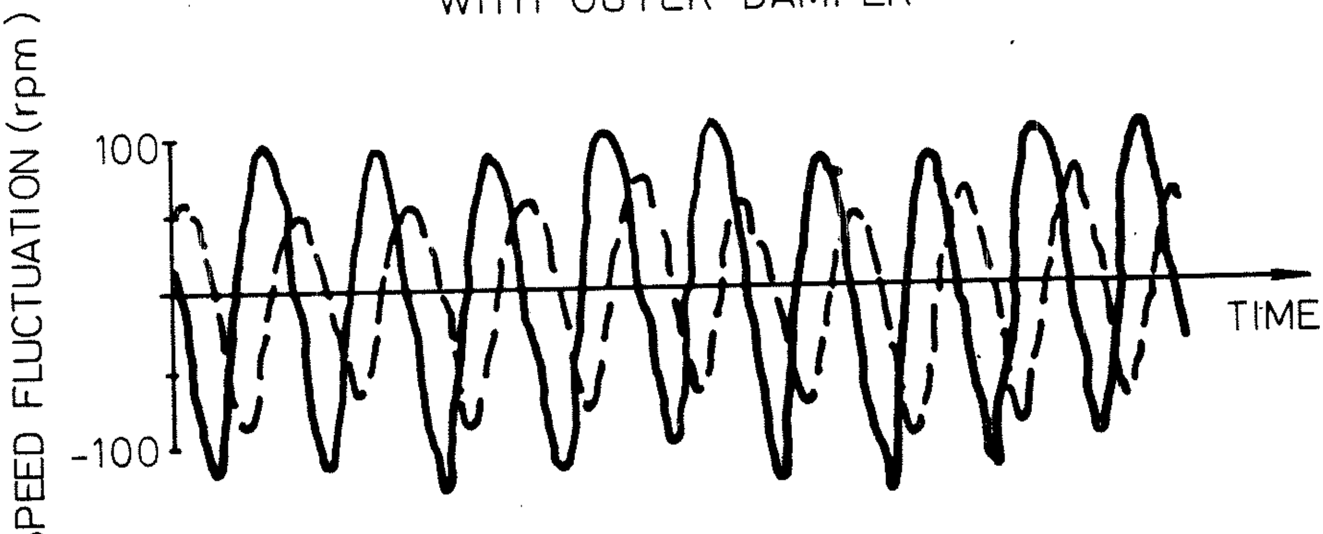


11 02 86	TORQUE CURVE OF ENGINE AND CENTRIFUGAL CLUTCH	LUK

WITHOUT OUTER DAMPER



WITH OUTER DAMPER



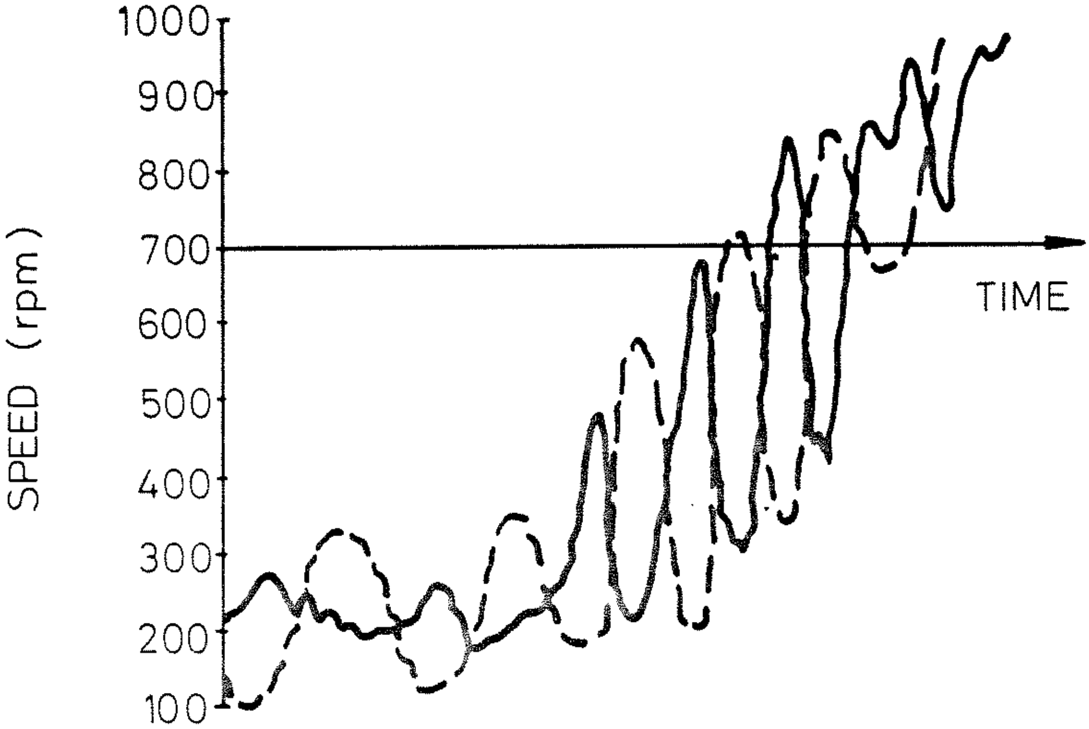
ENGINE SPEED 400 RPM

——— ENGINE

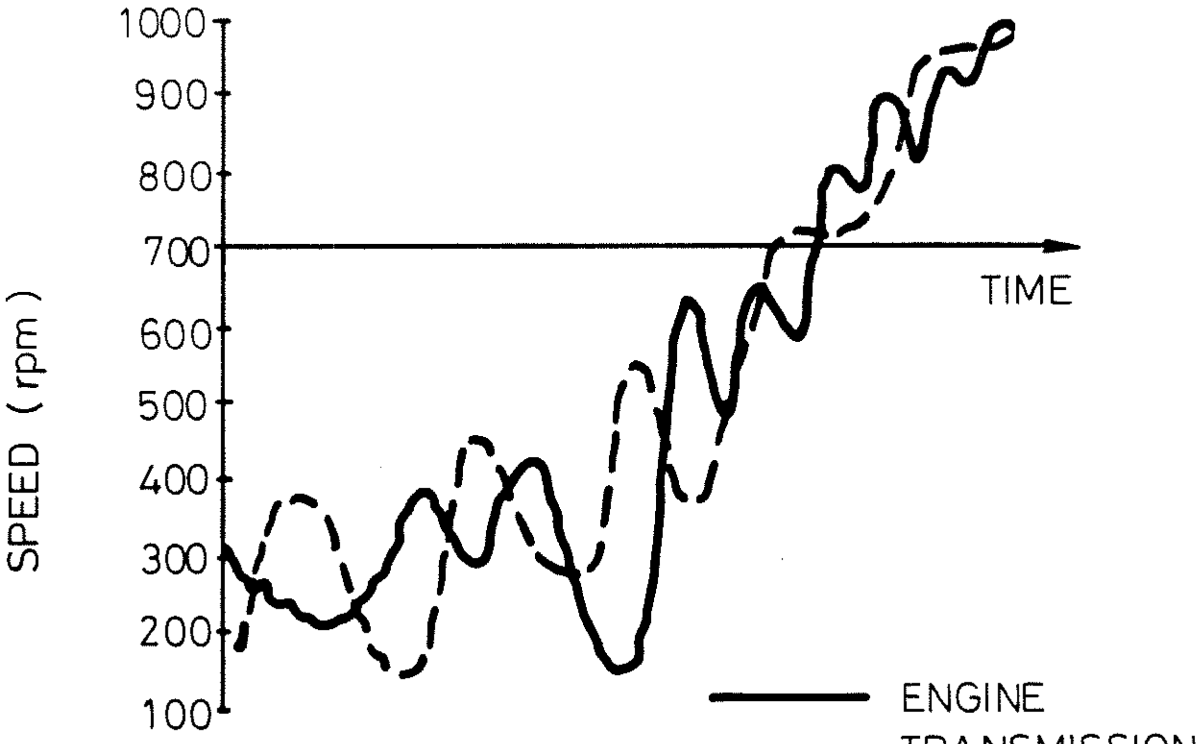
- - - - TRANSMISSION

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

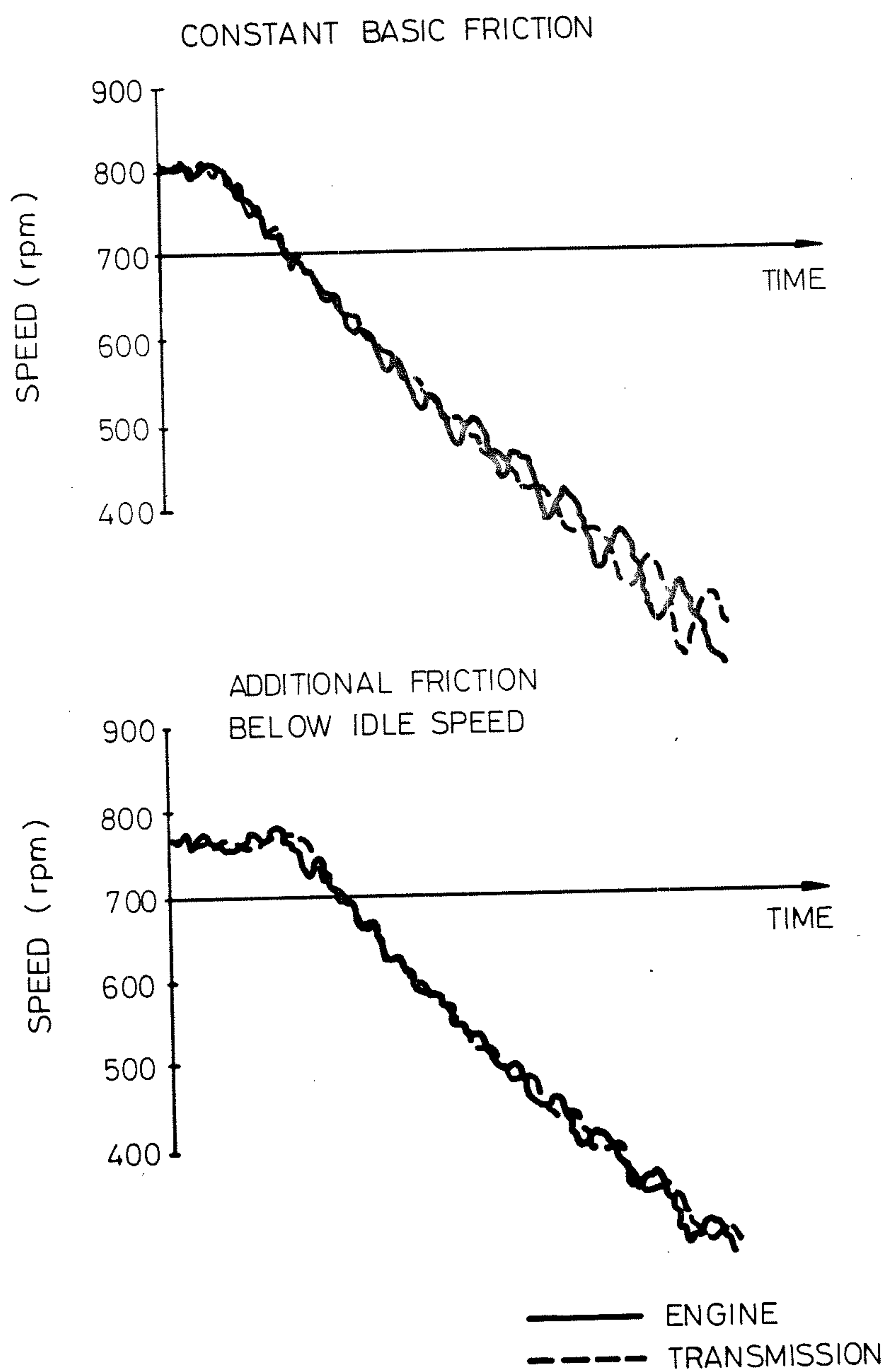
WITHOUT OUTER DAMPER



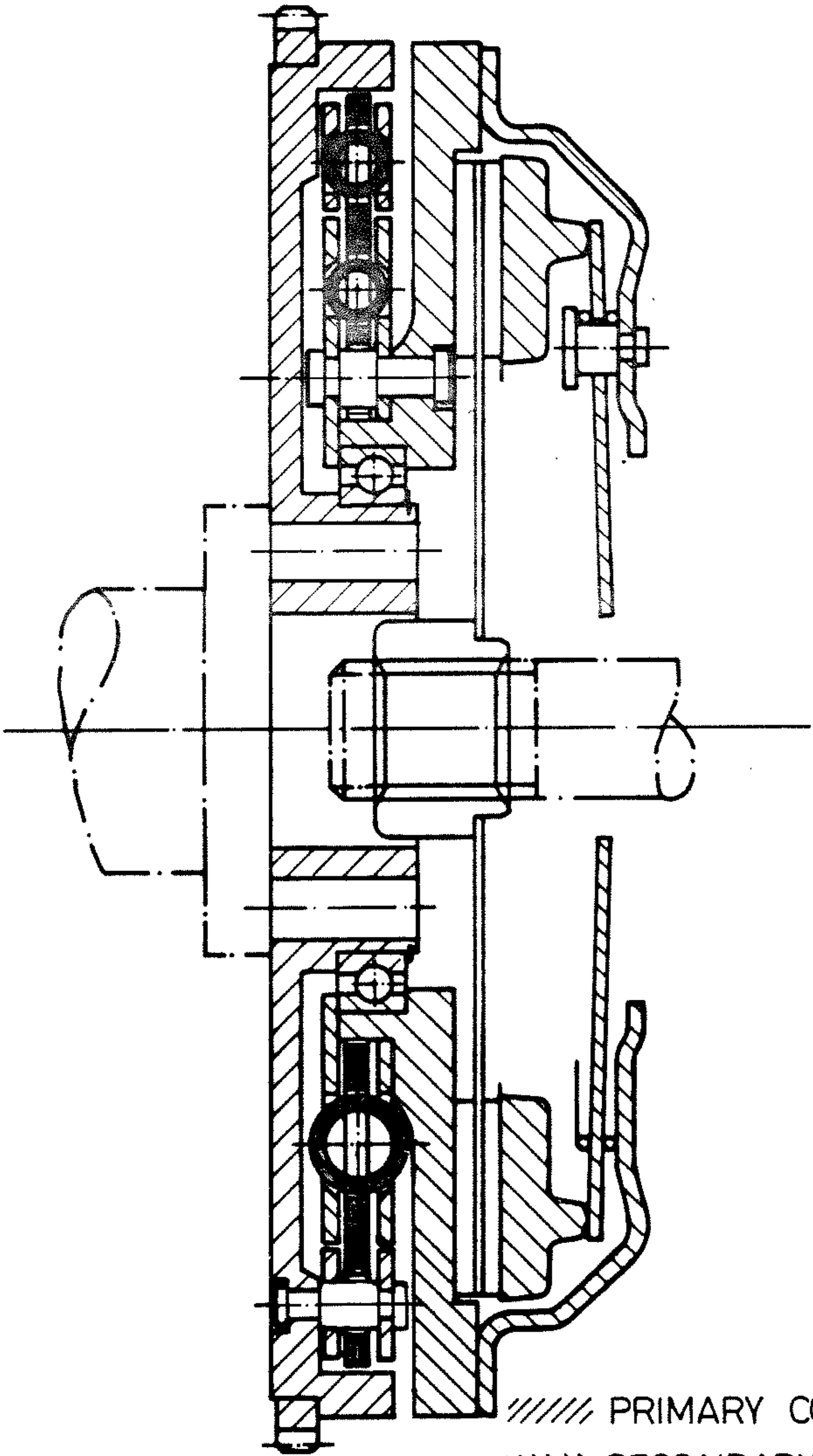
WITH OUTER DAMPER



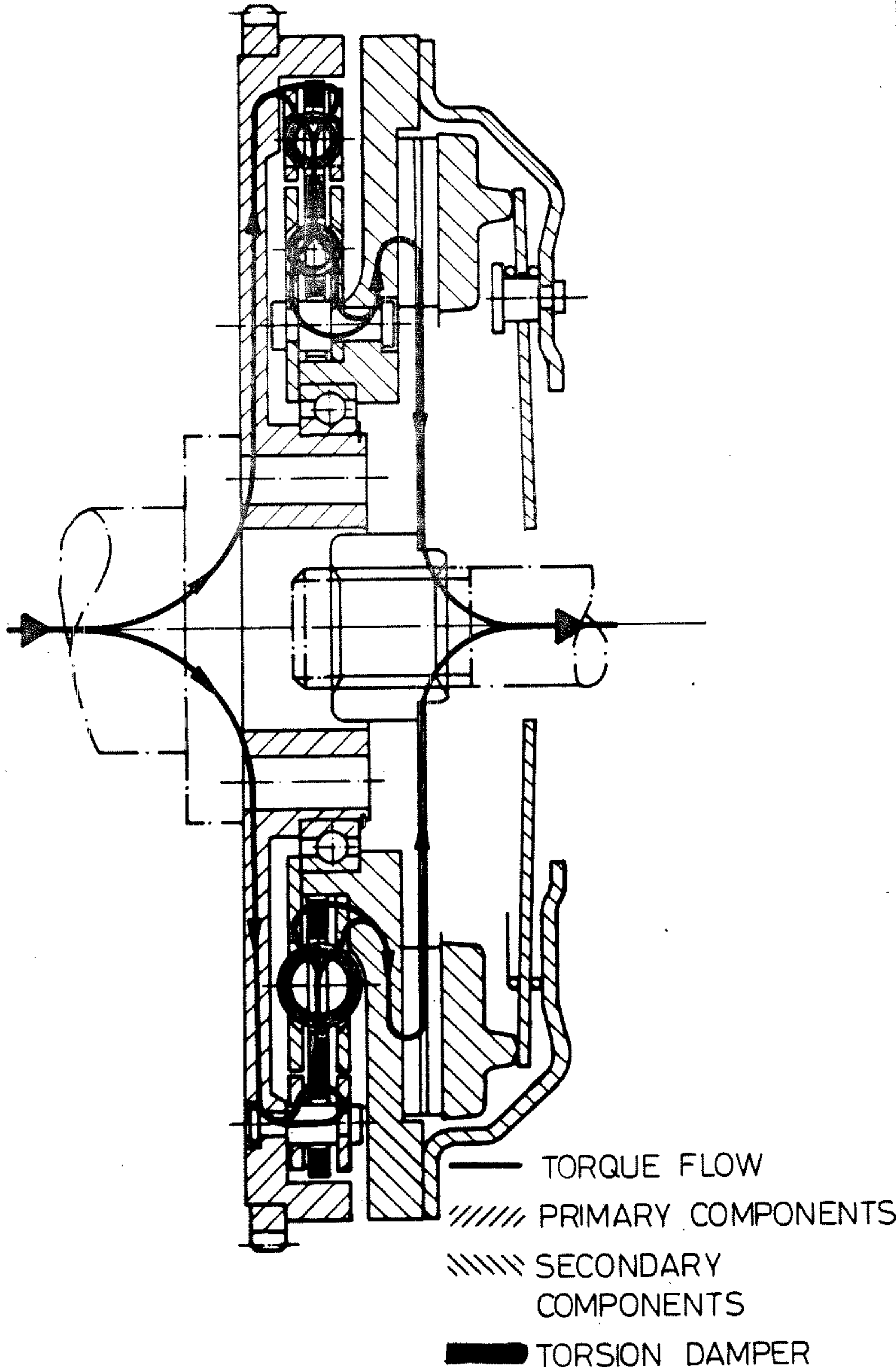
13 02 86	MEASUREMENT OF ENGINE START-UP WITH THE DUAL MASS FLYWHEEL	LUK



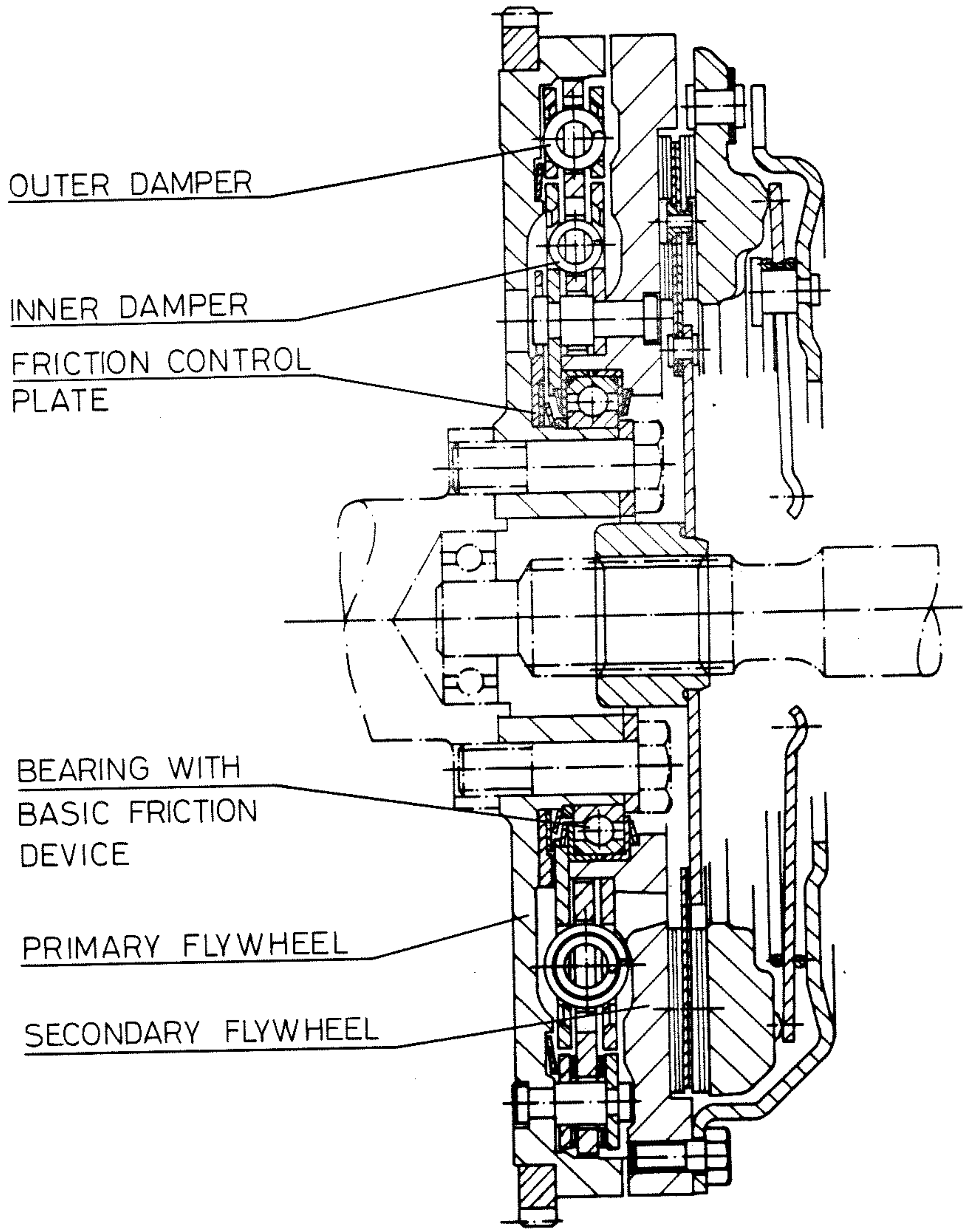
14 02 86	MEASUREMENT OF ENGINE SHUT-OFF WITH THE DUAL MASS FLYWHEEL	LUK



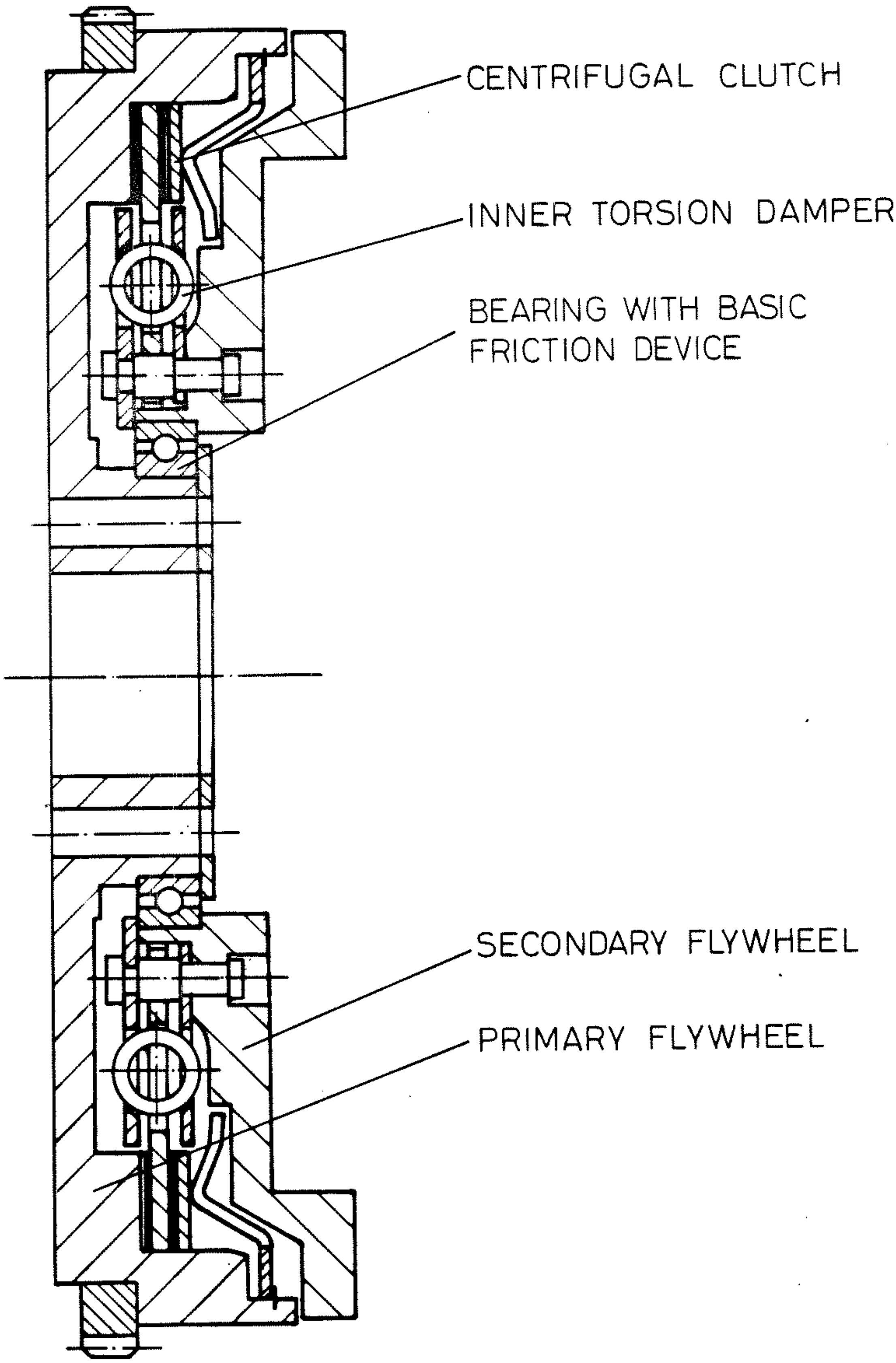
15 02 86	BASIC DUAL MASS FLYWHEEL	LUK



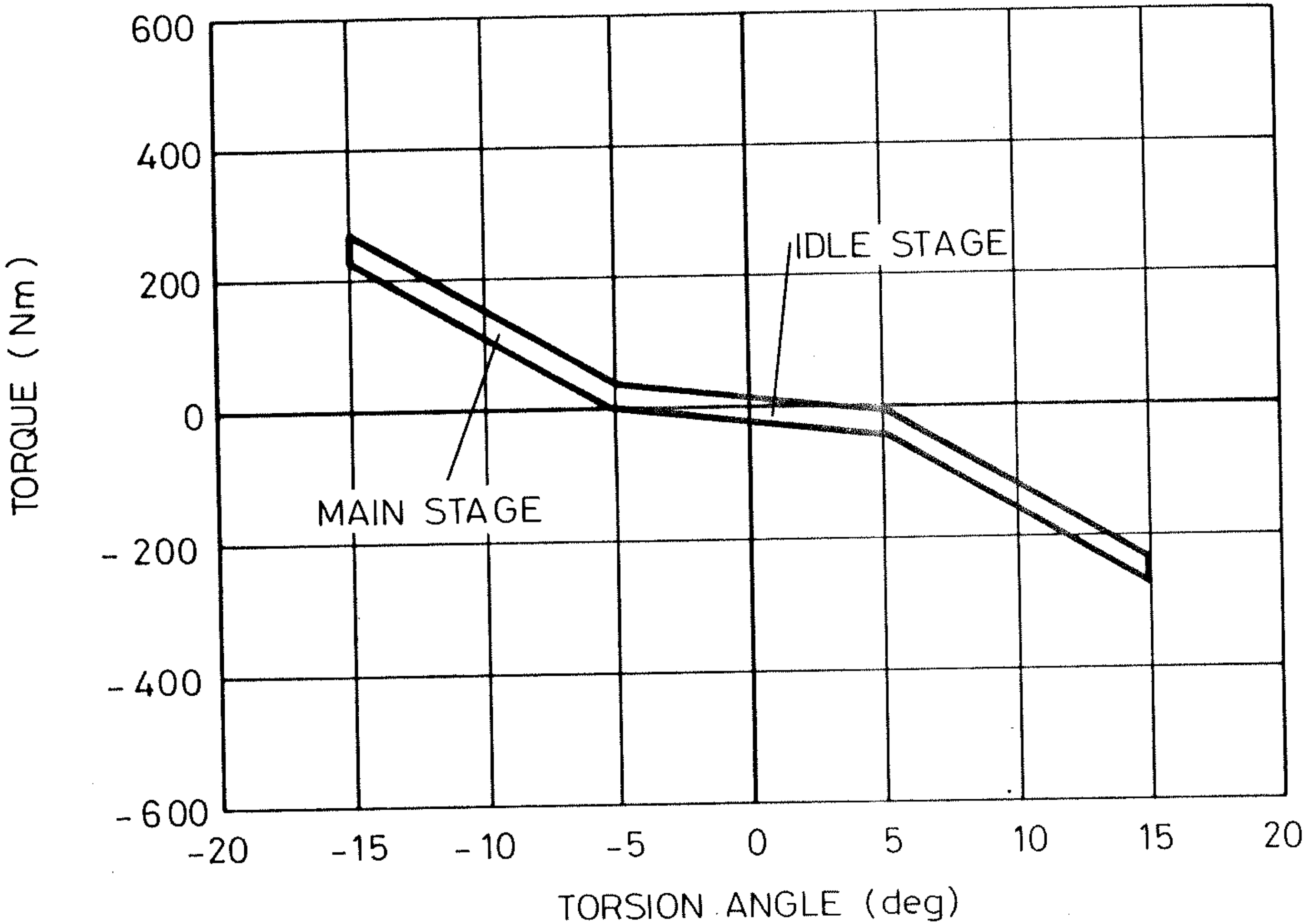
16 02 86	TORQUE FLOW THROUGH THE DUAL MASS FLYWHEEL	LUK



17 02 86	DUAL MASS FLYWHEEL WITH SERIAL DAMPER

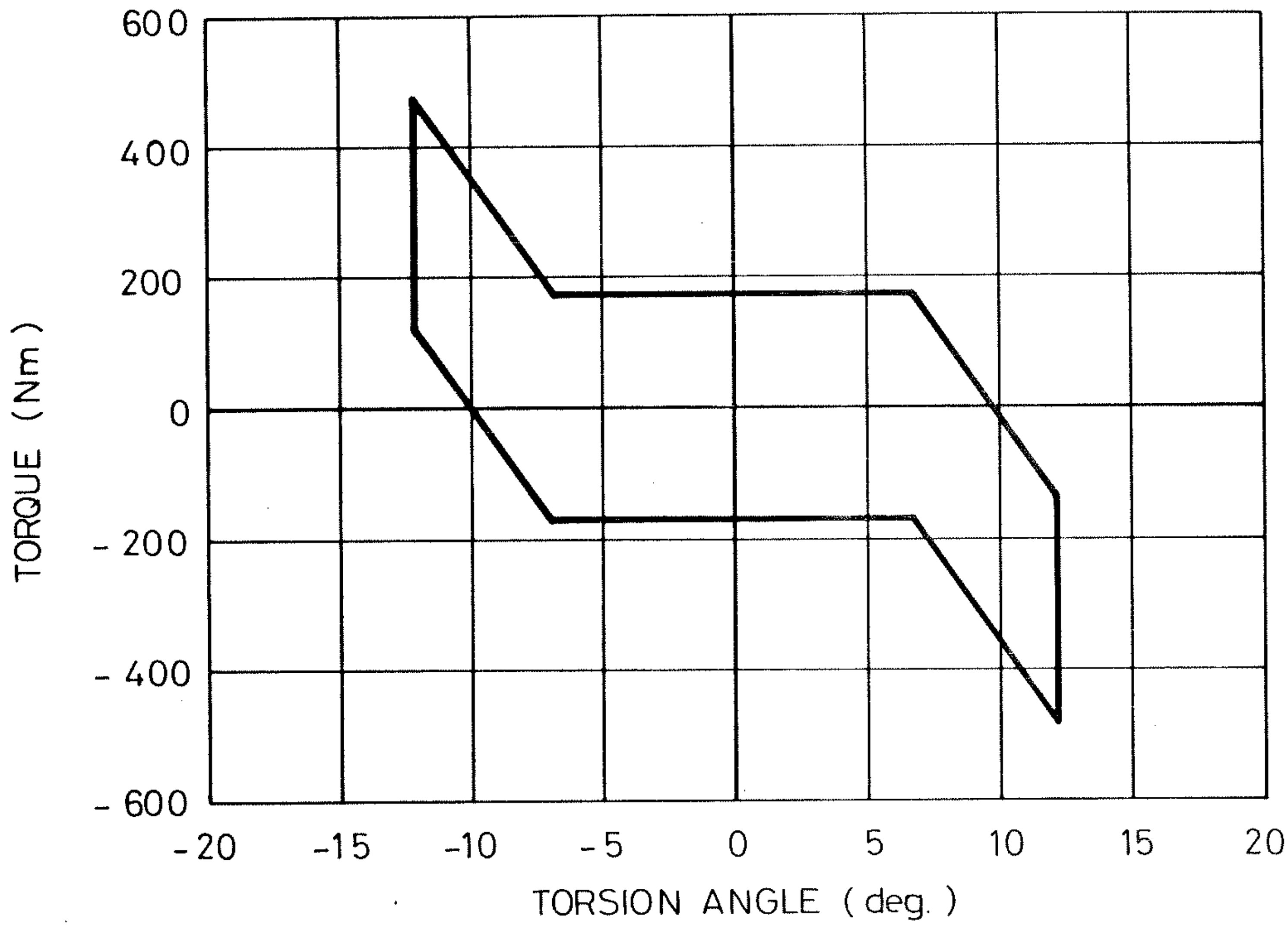


18 02 86	DUAL MASS FLYWHEEL WITH CENTRIFUGAL CLUTCH	LUK



19 02 86	WIND UP CURVE OF THE INNER DAMPER

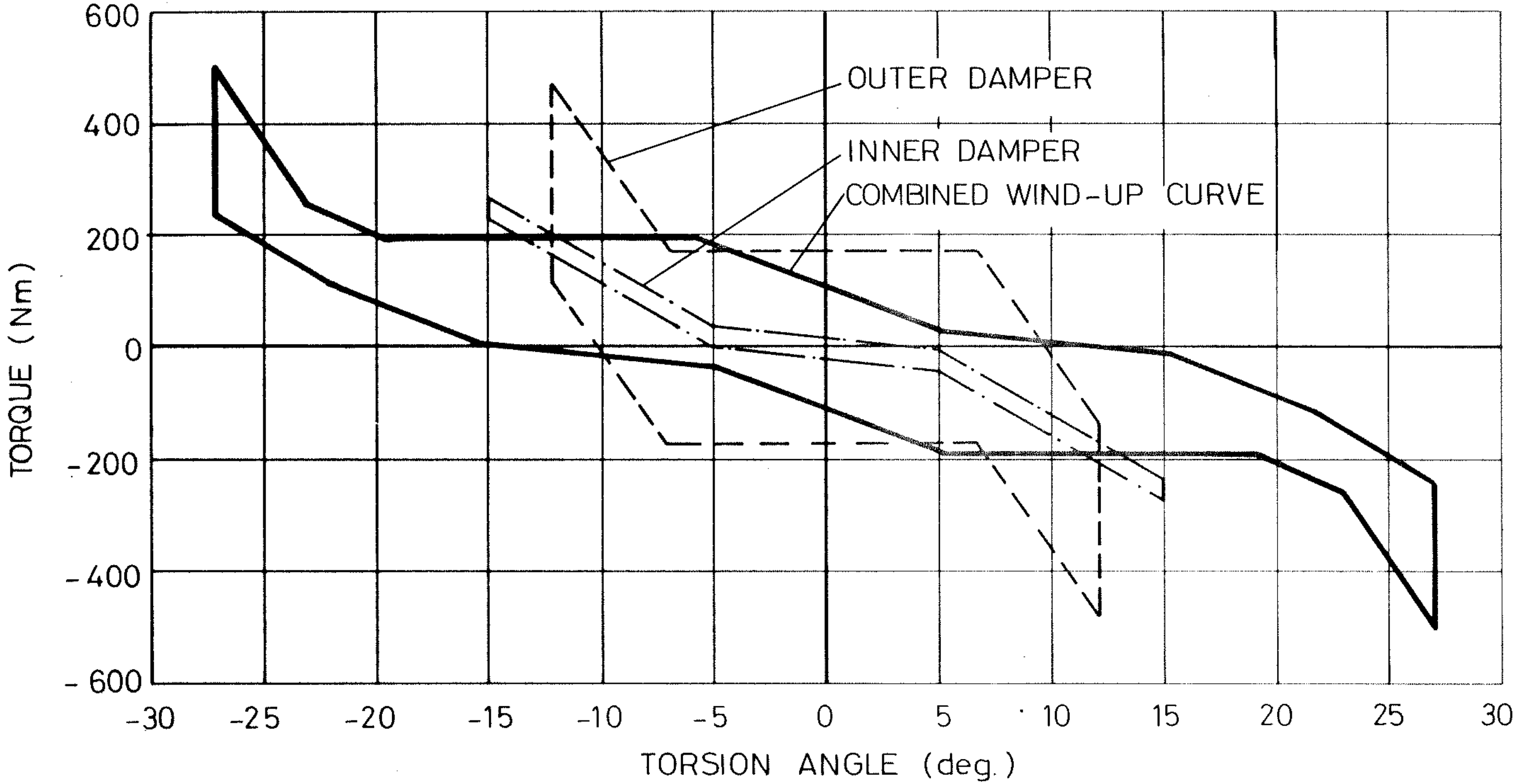
LUK



20 02 86

WIND - UP CURVE OF THE OUTER DAMPER

LuK



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

Kupplungen
Clutches
Embrayages
Embragues

RELIABILITY AND SERVICE LIFE OF TRACTOR CLUTCHES

DIPL.-ING. KARL KECK

APRIL 1986

1 0006/5/01.86/S+S

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RELIABILITY AND SERVICE LIFE OF TRACTOR CLUTCHES

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RELIABILITY AND SERVICE LIFE OF TRACTOR CLUTCHES

1. Introduction:

Reliability is a product's ability to fulfill its required functions under specified conditions over a certain period of time.

This presentation concerns itself with the **reliability** of a tractor component, specifically the main drive clutch.

Aside from **reliability** we have to take into consideration other competing characteristics affecting the total vehicle. In addition to the omnipresent problem of cost, these characteristics include primarily:

- Operating comfort
- Maintenance of necessary package dimensions
- Universal vehicle use.

The last characteristic, "universal vehicle use" makes it difficult to specify target values for the design because we have insufficient data on clutch stresses and their distribution over the course of time.

However, we can't specify reliability until we know the stress distribution during practical application.

So far as reliability is concerned, this presentation is limited to dry friction clutches, which are used almost exclusively in Europe. I will cite necessary and possible improvements.

On the other hand, the sections pertaining to function and load do not depend on the clutch type. These values also apply generally for the development of other kinds of clutch assemblies in the event that total vehicle design prevents us from achieving the required reliability with a dry friction clutch.

2. Clutch Functions, Failure Modes

The following three functions are required of the clutch (Figure 2); failure of any of these features will result in the clutch having to be repaired:

- **Connecting two shafts mechanically in order to transmit the engine torque to the vehicle**

- **Separating two shafts mechanically in order to permit gear shifting in the transmission**

- **Isolating two shafts in a vibrational sense in order to eliminate transmission noise.**

Reliability is based on the frequency with which the clutch has to repaired and its service life up to that point in time. For example, we can use a Weibull analysis to determine reliability.

Only in some cases can we use objective criteria for evaluating clutch failure or inability to perform its functions. In these cases we can prove the cause of failure by measuring functional charateristics, such as facing wear, breakage or setting of spring components, broken linkage components, etc. In other cases, particularly during the warranty period, clutches are removed for subjective reasons, that is, the user either rejects the functional characteristics from the outset, or he complains that something has changed that can't be documented with clutch measurements.

LUK Kupplungen Clutches Embrayages Embragues

The following discussion will describe the three cited clutch functions and related types of failure.

2.1 Separating Two Shafts

Mechanical separation and vibrational isolation appear to be of secondary importance to proper clutch design. Generally speaking mechanical separation requires that the slip torque M_R be a clearly defined function of the pedal travel, with the limiting condition $M_R = 0$, which means that given enough clearance, the clutch disc will run free. The mechanical-geometrical relationships involved must not change significantly when affected by temperature, corrosion, contamination and the presence of coolant.

This is hardly the case, as proven by warranty claims on clutches with less than 1000 operating hours. About 50% of clutch repairs can be attributed to this type of problem. People not only complain about incomplete disengagement, but also about the amount of operating force needed for disengagement, and the failure of linkage components.

As far as vibrational isolation goes, tractor operators actually don't complain much about such problems. This is not to imply that people who drive tractors have less sensitive ears than drivers of passenger cars. The real reason is that the vehicle parameters allow us to provide a solution to the problem using a relatively simple torsion characteristic. Figure 3 shows this kind of 2-stage characteristic curve with a ± 4 degree wind-up angle in the first stage and a total wind-up angle of 18 degrees, which is almost always appropriate for 4 and 6 cylinder engines. The most important factor here seems to be that the first stage can be relatively stiff because of the high moment of inertia of the transmission components. Thus, the first stage eliminates both idle and drive rattle; the latter is fairly typical in tractors if they are driven at excessively low speeds in the higher gears.

Another advantage is that there is usually sufficient package space available in tractor clutches for installing a sturdy damper design. Facing wear and prevalent temperatures dictate a much larger clutch than is required for passenger cars with the same engine torque.

2.2 Connecting Two Shafts

The connection between two shafts ceases to function when the friction facings are completely worn. In dry clutches this wear is an unavoidable result of the friction work performed during the connecting cycle when the shafts are under load and rotate at different speeds.

During the service life of the clutch up to total wear, the resulting moment of the friction must be greater than the engine torque transmitted under all operating conditions--in the case of tractor clutches by a factor of about 1.4. The temperatures involved cause friction coefficient variation of about a factor of 3 and exert a dominating influence on the moment of the friction.

Therefore, two basically different types of failure affect the connecting function. If we plot failure due to total wear according to a Weibull distribution, we get a line that curves to the right for the clutch, as well as for other wear items. As an example Figure 4 shows a sample evaluation of 30 double clutches with 310 mm diameter for tractors in the 50 to 80 kW (70 to

110 hp) range from the Middle European area. All the clutches have been repaired due to complaints of difficult gear shifting. Consequently there is a certain probability that we are dealing with an accidental selection so far as stress is concerned. The curve indicates a minimum service life of about 1000 hours.

The failure curve for "slippage due to facing wear" indicates maximum anticipated clutch reliability. All the other causes for failure overlap with this curve and decrease the overall reliability.

Clutch repair for slippage <u>before</u> the wear reserve has been used up currently account for about 50% of all complaints, almost independent of the clutch model and the absolute number of complaints registered.

However, insufficient torque transmission results in considerably increased friction work and a corresponding increase in wear, making it impossible to determine the actual cause of failure.

Of course, failure as a result of "slippage due to facing wear" would not be particularly important if the curve were mostly or entirely displaced outside the vehicle service life range, which is determined by the design of other components, such as the engine. This is not the case for a tractor with a vehicle service life of about 10,000 hours; however, it could hold true in the future for passenger cars using non-asbestos facings, as shown in Figure 5. In this case we evaluated 100 clutches with 230 mm diameter from a mid-sized vehicle with a turbo-charged diesel engine. All of these clutches were returned because of chatter problems. Consequently in this case there is once more a certain probability that we are dealing with an accidental selection so far as load is concerned. The reliability curve again bends to the right typical of wear parts. The minimum service life amounts to 70,000 kilometers. Since the average service life for a passenger car in this class is about 120,000 kilometers, failure on account of "slippage due to facing wear" only plays a role in 15% of all vehicles.

3. Operating Conditions

Operating conditions and stress on the clutch vary much more for a tractor than they do for a street vehicle. Even an individual tractor has a far broader stress range due to its use with a wide variety of implements, under variable weather conditions and operation by several persons. The stress range for an individual tractor expands considerably if we look at all tractors for a given model. Tractors are sold all over the world today; soil conditions, field size, and environmental conditions, in addition to the type and operating frequency of implements are all different. We have derived the following approximate clutch stresses from our experience and we base our design calculations on these values.

3.1 Engagement Frequency

Approximately 0.1 to 8 per minute. These are mostly start-up cycles.

3.2 Friction Work and Temperatures

Because of a fairly constant ratio of performance to weight--at least in the case of standard tractors--it is meaningful to relate friction work and friction performance to engine performance. Figure 6 shows experience values as well as resulting temperatures for current standard dry friction clutch sizes.

a) Normal continuous load, such as plowing

- Average friction work equivalent to 0.3 to 0.6 % of rated engine performance
- Friction work per kW of engine performance per operating hour: 10,000 to 20,000 Joules
- Ambient temperatures in the bell-housing: 70 to 90° C
- Peak temperature of the friction surfaces: 130 to 150° C.

b) Extreme continuous load, such as front loader application

- Average friction performance equivalent to 2.5 to 5% of rated engine performance

- Friction work per kW of engine performance per operating hour: 90,000 to 180,000 Joules
- Ambient temperature in the bell-housing: 100 to 160° C
- Peak temperatures of the friction surfaces: 230 to 350° C.

c) Extreme short term load intervals, such as starting up with a high tractive effort

- Slip cycles in a range of about 10 seconds at full engine performance.

In this case it doesn't make any sense to specify average friction performance or friction work per kW of engine performance per operating hour, because this kind of stress would destroy a dry friction clutch within a short time given current cooling conditions. The temperature of the friction surfaces jumps by about 80° C, and between 2000 to 5000 of these cycles will completely deplete the wear reserve.

3.3 Torques Values

We have to count on peak torque values up to 3 times the maximum engine torque.

3.4 Clutch Installation

The clutch is normally installed in a completely closed bell-housing in order to prevent malfunctions caused by contamination. Sliding surfaces and bearings in the release system are even more susceptible in this regard than the clutch itself. The closed bell-housing also prevents outside air from cooling the clutch directly; cooling occurs indirectly by the air in the bell-housing, which is considerably hotter. Since we will also have to deal with closed bell-housings in the future, the following calculations are based on the temperature curve shown in Figure 7. The friction surface temperature is at least 70° C and increases in proportion to the average specific friction performance. Depending upon the clutch design and the size of the bell housing, we will find significant variation, as indicated by the shaded area in the illustration.

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The size of the bell-housing is also limited by the package dimensions of the tractor. In this regard, criteria such as wheel base, frame width, free access to the cab, use of standard production equipment, etc. all play a role. Even in larger tractors in the range from 80 to 120 kW, the bell-housing inner diameter normally limits the clutch size to a maximum diameter of 330 to 360 mm.

3.5 Release System

Because of cost, current release systems operate almost entirely on muscle power. Based on our experience, the pedal load should be limited to maximum 200 N in a new tractor and maximum 300 N during the entire operating life. Maximum pedal travel is from 150 to 200 mm in tractors with a cab, and the release systems have efficiencies between 40 and 60%.

4. Reliability Goals and Stress Specifications

Depending upon operating conditions, the clutch achieves certain reliability values in performing the functions described here. These values show considerable variation in tractor models currently equipped with LuK clutches, primarily because there is also considerable variation in the design characteristics. The shaded area in Figure 8 shows the overall reliability currently achieved, as a result of the types of failure described above.

- Average service life of 2000 to 6000 operating hours
- Failure rate at 1000 operating hours of 1 to 20%.

In this light, the current situation is more or less satisfactory.

Tractor manufacturers do not have very specific goals for clutch reliability. Nor do we have enough information on the stress distribution. However, since both factors must be specified if development work is going to be meaningful, I would like to pose the following assumptions for discussion:

4.1 Reliability Goal

As noted above, Figure 8 shows the current total clutch reliability range as a shaded area. The clutches on the right side of the area have been rated as excellent, and customers generally reject any cost increases for further improvements in these clutches. Therefore, we suggest that the bold border line on the right be viewed as an adequate goal for clutch reliability in the next generation of tractors. In the process, we are assuming that this total reliability goal will only be achieved if failures due to facing wear do not exceed the dash-dot line, which is defined by the following points:

- Minimum service life of 1000 operating hours
- 80% reliability at 4000 operating hours.

The classification of the two curves, that is the relationship of facing reliability to the total reliability, reflects current LuK experience.

4.2 Stress Specifications

Up until now LuK has based its service life calculations on specific stress values, assigning a specified amount of friction work per each kW of engine performance per operating hour:

- 100,000 Joules per kW per hour as maximum stress, which yields the minimum service life of the clutch.

- 13,000 Joules per kW per hour as minimum stress, which yields the maximum service life of the clutch.

In Figure 9 the maximum stress, assumed as a distribution, is plotted vs. the relative friction performance. It contains portions which reach a friction performance of about 11% of engine performance, accompanied by corresponding high temperatures.

We don't need to specify a distribution for the minimum stress because the relatively low temperature level has no effect on the wear performance of the facings.
These load extremes have been calculated based on field results. In our opinion they still describe the

extreme conditions for tractor clutch application with sufficient accuracy; therefore we have used them for the following calculations.

The frequency of such extreme conditions within a vehicle population is of critical importance for the slope of the reliability curve.

Past calculations have obviously underestimated the frequency of high stresses. The following discussion is based on the clutch stress distribution shown in Figure 10, which gives us a reliability curve coinciding to current experience. The extreme values of 100,000 and 13,000 Joules per kW per hour are plotted at cumulative frequencies of 0.1 and 99.9 and connected with a straight line. The average load at 50% cumulative frequency, which yields the average service life, is 22,000 Joules per kW per hour. The curve we are primarily interested in runs from the maximum value to this average value and corresponds roughly to a logarithmic standard distribution. For the sample calculation in Figure 11, we used LuK's design limits of 80 Watt/cm^2 with ceramic facings. This yielded a wear reserve of 3 mm. This would apply, for instance, to a

310 mm diameter clutch for a 75 kW (102 hp) tractor. The calculated result is represented by the solid line on the left side of the graph. The failure rate of just 4% at 1000 operating hours based on the wear alone does not appear to be at all unrealistic. For purposes of comparison the broken line on the right side of the graph shows the goal as cited previously.

5. Required Sizing for the Cited Reliability Goals

5.1 Facings

In addition to the stresses and their resulting temperatures, we need to know the wear performance of the facings in order to calculate the service life. The measured wear rate is plotted for the following facing materials in Figure 12 as a function of the temperature:

- Ceramic facings--solid line
- Organic non-asbestos facings--broken line
- Organic asbestos facings--dash-dot line

Contrary to previous experience, current results from tests with the best non-asbestos organic facings, over the total relevant temperature range, indicate that wear is lower than that of asbestos organic facings. Since asbestos facings are supposed to be taken off the market by the end of this decade, the following discussion is based entirely on the use of either non-asbestos organic or ceramic facings.

5.2 Required Sizing Characteristics

Engine Performance/Friction Surface and Wear Reserve

Figure 13 shows the characteristic sizing values, engine performance per friction surface and wear reserve, which allow us to achieve the required reliability goals under the cited conditions. These values are indicated in lines 1 and 2. In each case, the wear reserve is chosen to ensure a minimum service life of 1000 operating hours. If we require service life of 4000 operating hours with 20% failure rate, we can see, according to line 4, that non-asbestos organic facings exceed this goal by about 10%.

This results from the flatter reliability curve. Ceramic facings fall short of the goal by about 10%.

As for the cited results, I would like to add:

- This requires a relatively generous clutch design; we should size in the range of 50 to 55 Watt/cm^2 with non-asbestos organic facing material. With ceramic facings, we need to be in the range of 60 to 65 Watt/cm^2 or--if we can achieve the specified wear reserve of 5.6 mm--up to max. 70 Watt/cm^2.

- At the same time a relatively high wear reserve is required. We can only achieve values over 4 to 4.5 mm using a diaphragm spring clutch which is lever-operated or a pull-type version. Values over 6 mm are unrealistic for diaphragm spring clutches.

Figure 14 shows the engine performance ranges covered by the currently prevalent clutch sizes using the cited sizing goals. If in the future we don't have clutch package space in excess of a nominal diameter of 350 mm, it would be a good idea to abandon the use of single-disc dry friction clutches for applications above about 88 kW (120 hp) and resort to using twin plate clutches or wet clutches.

5.3 Pressure Plate Thermal Capacity

At the moment, the performance weight of the pressure plate in LuK tractor clutches is between 5 and 11 kW/kg. In order to avoid overheating and failure as a result of grossly abusive operation, it appears advisable not to exceed values of 6 to 7 kW/kg in the future.

Figure 15 indicates the average pressure plate thickness required, dependent upon the characteristic size value: engine performance per unit of friction surface. If we relate the value 6 kW/kg to the organic facings and 7 kW/kg to the ceramic facings and use the sizing targets 55 and 65 Watt/cm^2 respectively, we arrive at a pressure plate thickness of about 25 mm in both cases.

5.4 Clutch Safety Factor, Pedal Effort

As described earlier, "insufficient torque transmission" accounts for about 50% of current complaints. In spite of our goal for generous sizing, LuK recommends that, we should also continue to design with a nominal clutch safety factor of min. 2.2, based on a friction coefficient of 0.27 for organic and 0.4 for ceramic facings.

We make the following assumptions in order to determine the pedal load based on the clutch safety factor and the resulting clamp load:

- Engines have a rated speed of 2200 rpm and 20% torque rise
- 160 mm pedal travel and 60% total release system efficiency in the tractor
- Mechanical clutch efficiency of 75% using a minimum lift-off of 1.8 mm

If we size at 55 Watt/cm^2 for organic and 65 Watt/cm^2 for ceramic facings, we arrive at the pedal efforts dependent upon the engine performance as shown in

Figure 16. The values on the graph apply for the clutch and linkage system in new condition. Here we should not exceed pedal loads of 200 N because we can count on an increase of about 50% over the operating life. This increase is caused by the decreasing efficiency of the clutch and linkage system, as well as the increasing diaphragm spring load. We experience complaints when the pedal load exceeds 300 N.

The illustration shows that the assumed 60% linkage efficiency is adequate for ceramic facings over the entire performance range, with the exception of the twin plate clutch. For organic facings, on the other hand, we have to introduce improvements in the clutch and the linkage system beyond about 50 kW. Such improvements might include beaded diaphragm spring fingers in the clutch or overcenter pedal assist springs.

5.5 Mass Moment of Inertia of the Clutch Disc

We have found it relatively difficult to eliminate complaints related to poor disengagement and the resulting high shift efforts. In this regard we ought to see

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positive results from more generous designs because temperature effects, which certainly play a role, will be reduced. However, we shouldn't overlook the fact that large diameters and a high wear reserve result in discs with high moments of inertia. Figure 17 illustrates the levels of inertia we have to expect. We are assuming here that organic friction facings have a density of 2 g/cm^3, and that a third of the friction surface is covered by facings when we use ceramic facings. The graph illustrates the mass moment of inertia we have to expect with both friction materials on discs with and without torsion damper.

6. Summary

We find the reliability of the dry friction tractor clutches currently being used is not entirely adequate. We are suggesting that we meet this problem with more generous designs, simultaneously increasing the facing wear volume and decreasing thermal stress. If we can apply the defined sizing values, we can achieve the desired reliability with regard to facing wear performance and torque transmission. On the other hand, failures due to increased pedal and shift efforts cannot be entirely corrected in the clutch, and sometimes not at all. When this is the case, we have to make improvements in the interactive tractor components as well, such as the release system and the synchronization.

Our cited reliability goal continues to define the dry friction clutch as a wear part which must be changed at least once, in most tractors, during the total service life of 10,000 h. If this is not acceptable, or if the necessary package space can't be made available, then we should use a different kind of clutch. The wet friction clutch is a possibility. It can be designed in such a way that it does not need to be considered a wear part. In this case we have to look at other aspects--in addition to the manufacturing cost--such as the abuse capacity factor and disengagement performance. LuK is of the opinion, however, that we have not entirely exhausted the capabilities of the dry friction clutch.

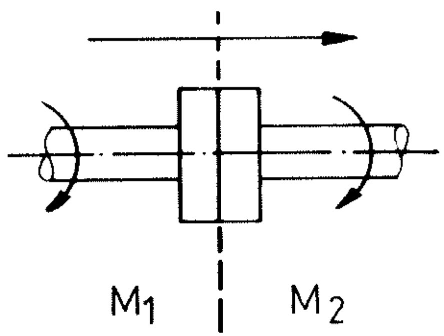
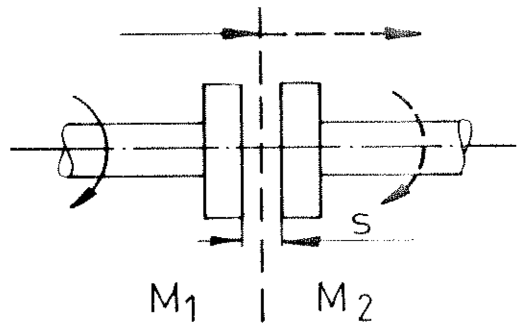
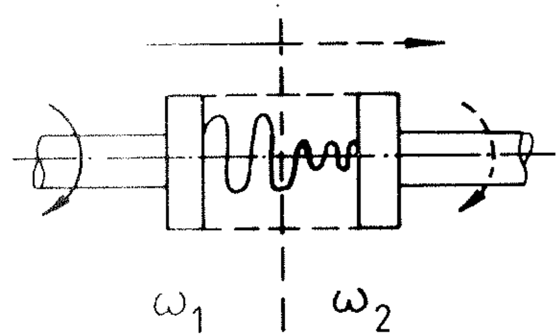
Zuverlässigkeit ist die Fähigkeit eines Produktes, die geforderten Funktionen während einer bestimmten Zeitdauer unter festgelegten Bedingungen zu erfüllen.

Affidabilita è la capacità di un prodotto di assolvere una funzione durante un tempo ben preciso in condizioni prestabilite.

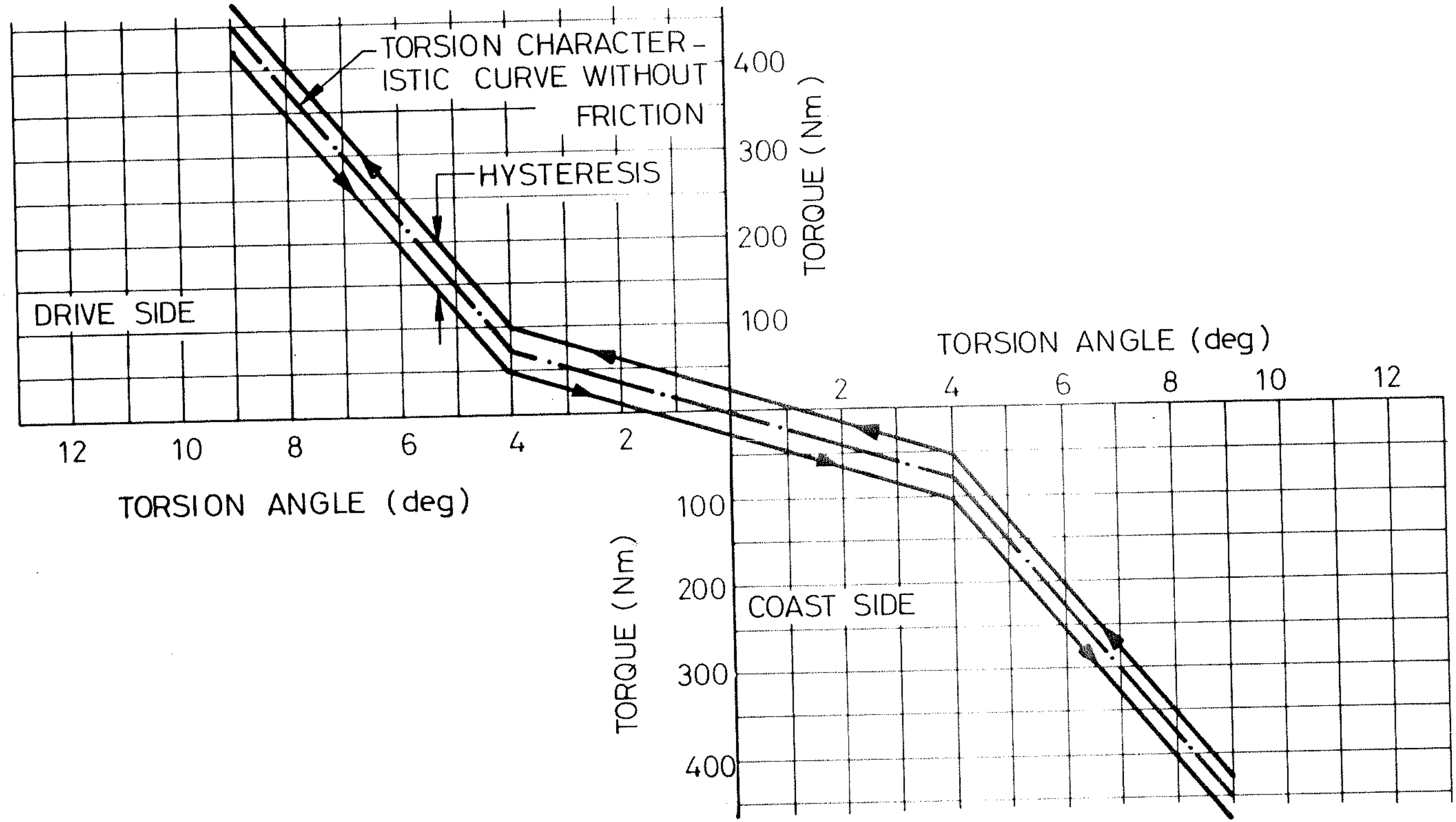
RELIABILITY IS A PRODUCT'S ABILITY TO FULFILL ITS REQUIRED FUNCTIONS UNDER SPECIFIED CONDITIONS OVER A CERTAIN PERIOD OF TIME.

La fiabilité est l'aptitude d'un produit à répondre aux fonctions exigées durant un temps déterminé pour des conditions fixées.

01 03 86	DEFINITION OF RELIABILITY	LUK

MECHANICAL CONNECTION	MECHANICAL SEPARATION	VIBRATIONAL ISOLATION
		
$M_2 = M_1$ $M_R = K \times M_1$ $> M_1$	$M_2 \overset{!}{=} f(s)$ $M_2 \approx 0 (s = s_{max.})$	$\dot{\omega}_2 = K \times \dot{\omega}_1$ $< \dot{\omega}_{max.}$

CLUTCH FUNCTIONS



TORSION CHARACTERISTIC CURVE FOR TRACTORS WITH 4 AND 6 CYL. ENGINES

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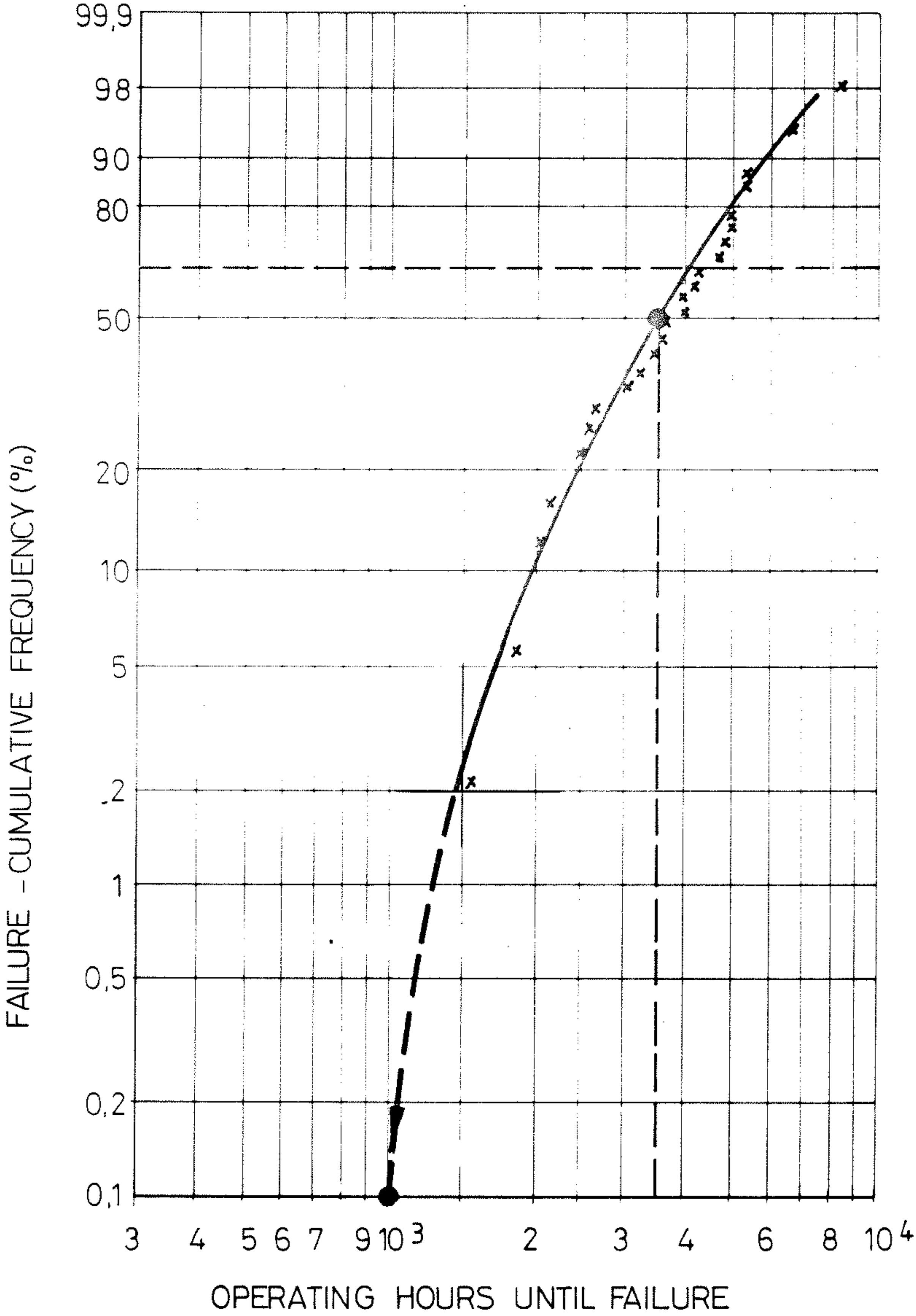
LUK



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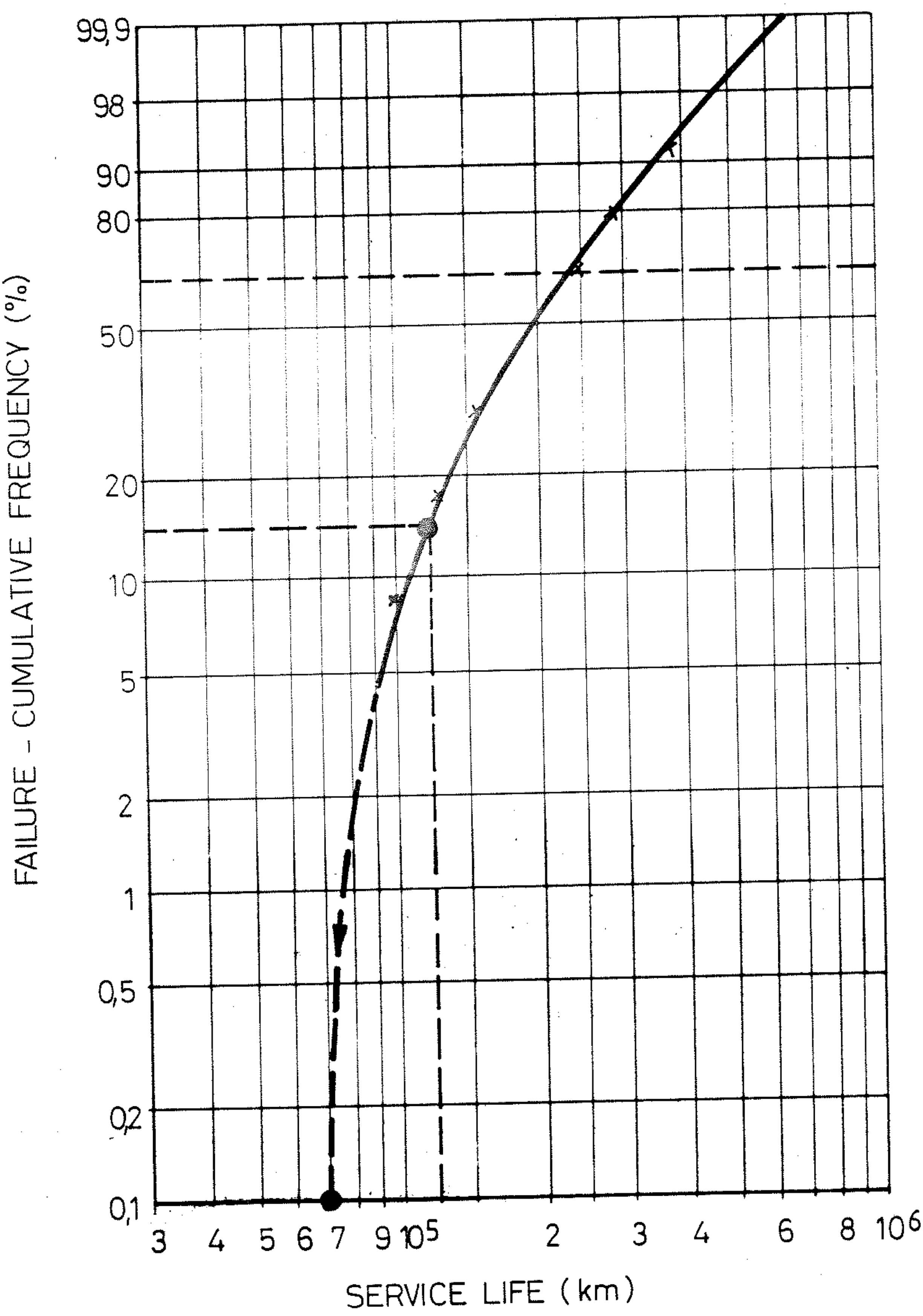
EXAMPLE OF TRACTOR CLUTCH RELIABILITY BASED ON FACING WEAR



05 03 86	EXAMPLE OF PASSENGER CAR CLUTCH RELIABILITY BASED ON FACING WEAR	LUK

	AVERAGE FRICTION PERFORMANCE / ENGINE PERFORMANCE	FRICTION WORK PER KW PER OPERATING HOUR	AMBIENT TEMPERATURE	TEMPERATURE OF FRICTION SURFACE
NORMAL CONTINUOUS STRESS	0,3 ÷ 0,6 %	10 000 ÷ 20 000 Joule / kWh	70 ÷ 90° C	130 ÷ 150° C
EXTREME CONTINUOUS STRESS	2,5 ÷ 5 %	90 000 ÷ 180 000 Joule / kWh	100 ÷ 160 °C	230 ÷ 350 ° C
EXTREME SHORT - TERM STRESS	SLIP CYCLES UP TO 10 SEC. AT MAX. ENGINE PERFORMANCE			

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STRESS ON TRACTOR CLUTCH

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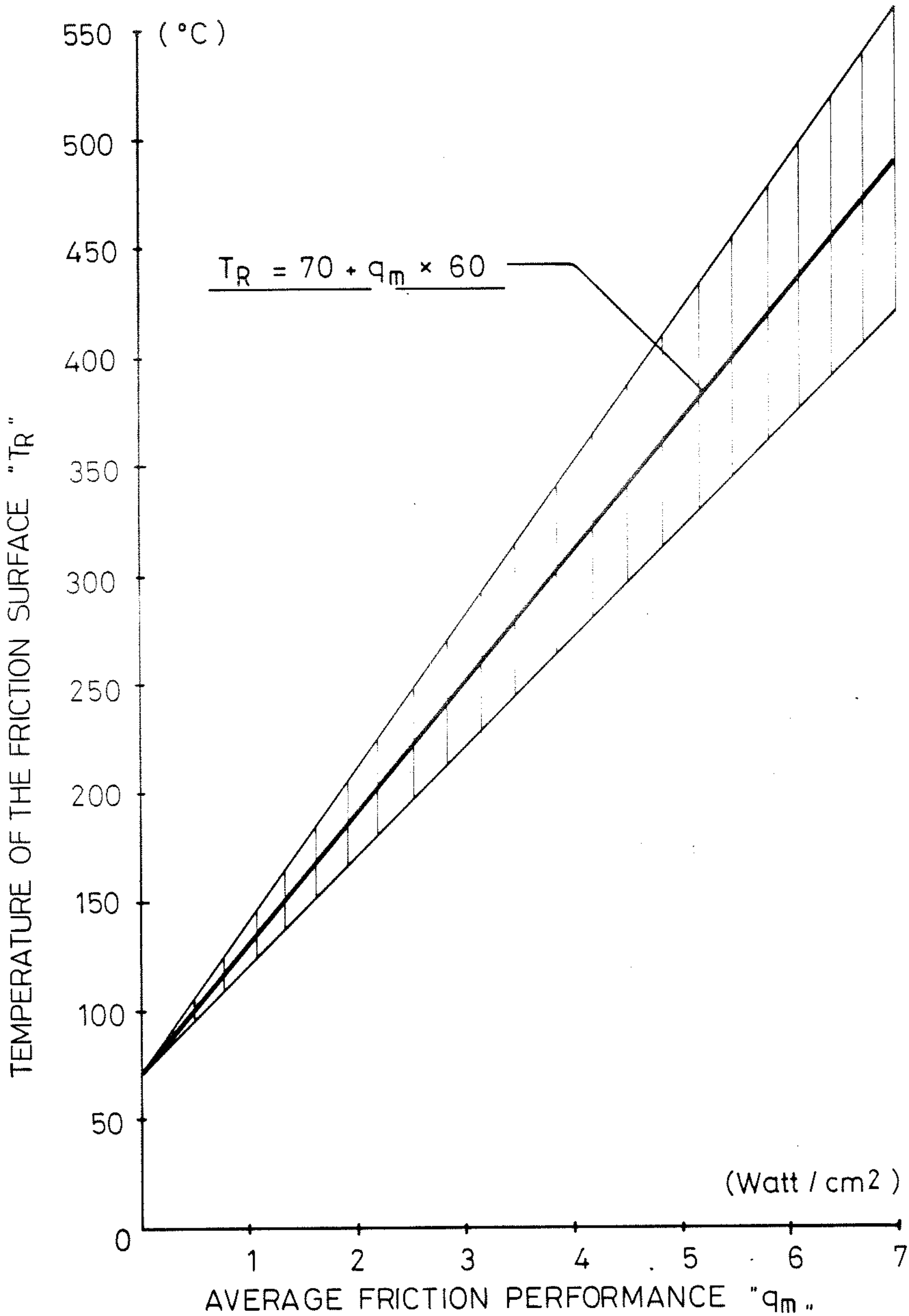
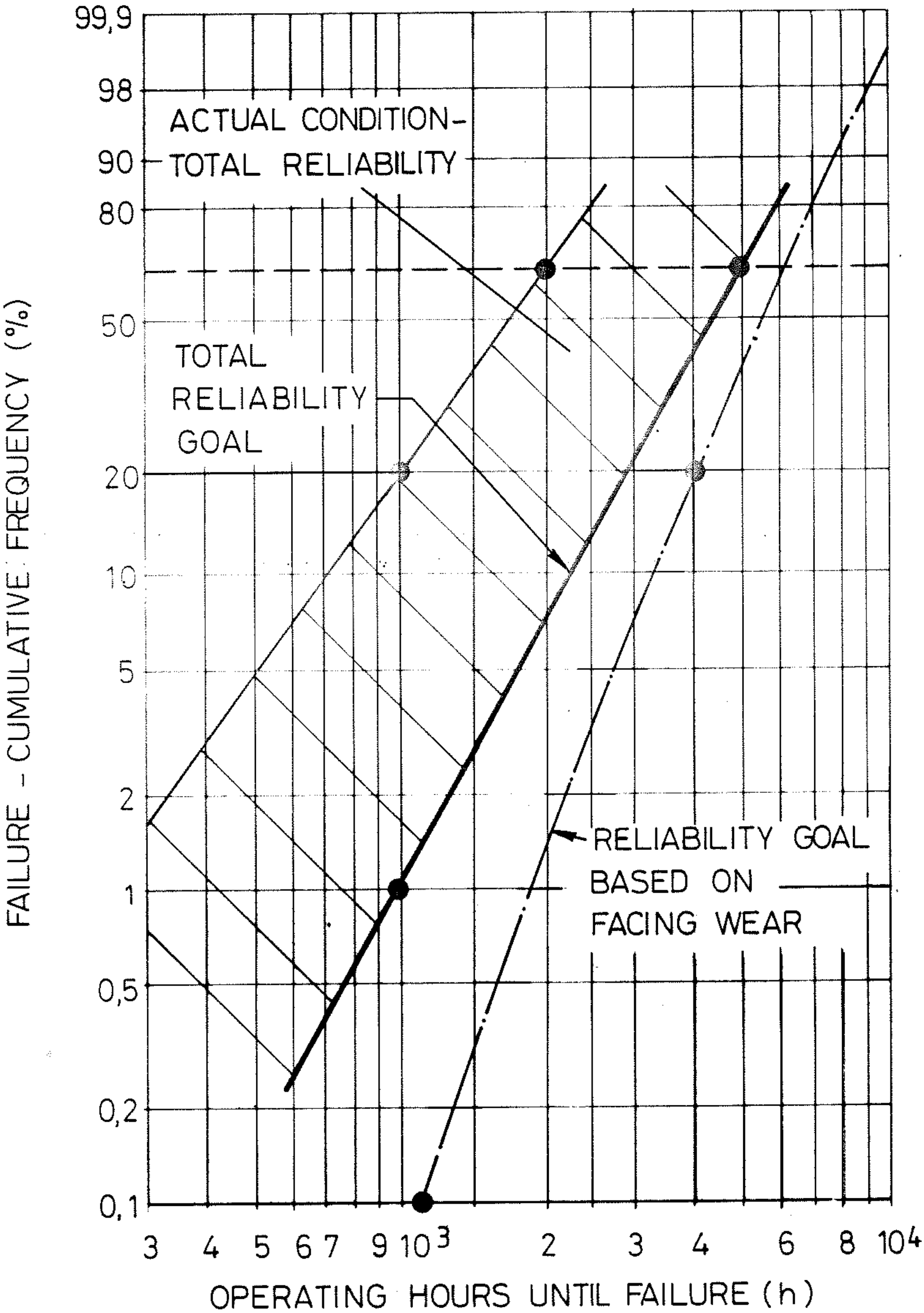
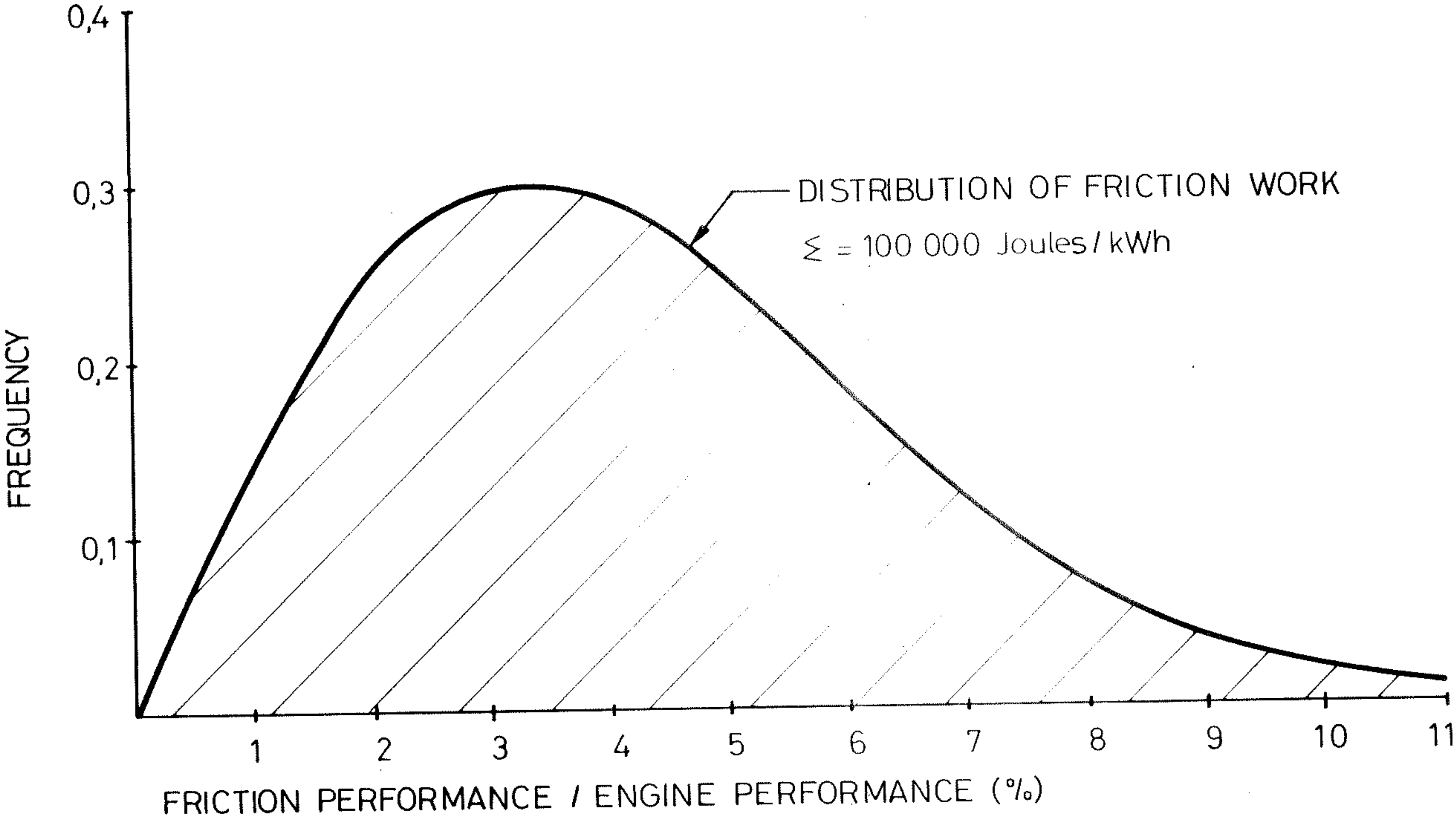


07 03 86	FRICTION SURFACE TEMP. VS. THE AVE. FRICTION PERFORMANCE WITH CLOSED BELL-HOUSING

LuK



08 03 86	RELIABILITY GOALS	LuK



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ASSUMED STRESS DISTRIBUTION FOR MINIMUM SERVICE LIFE

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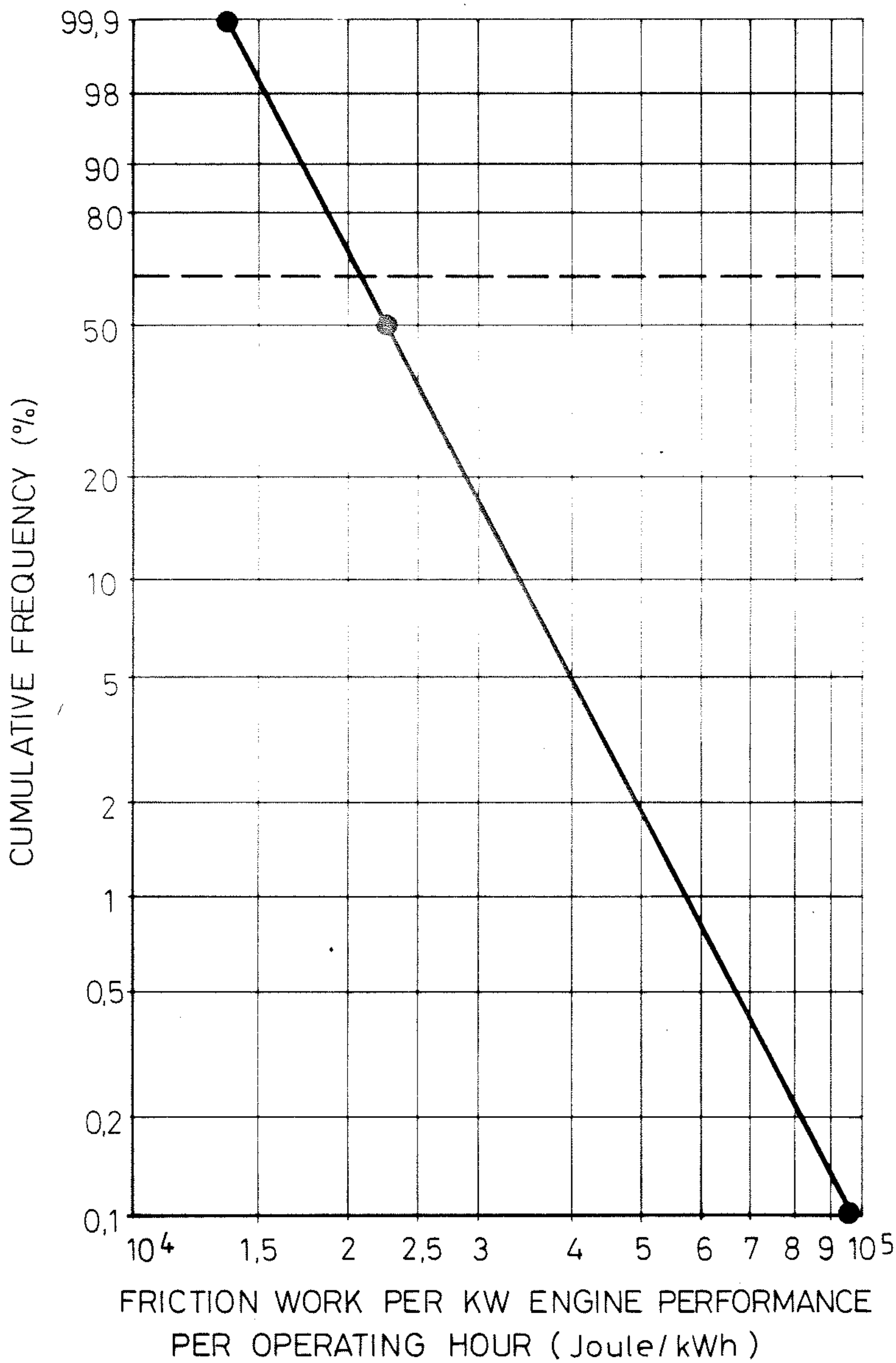
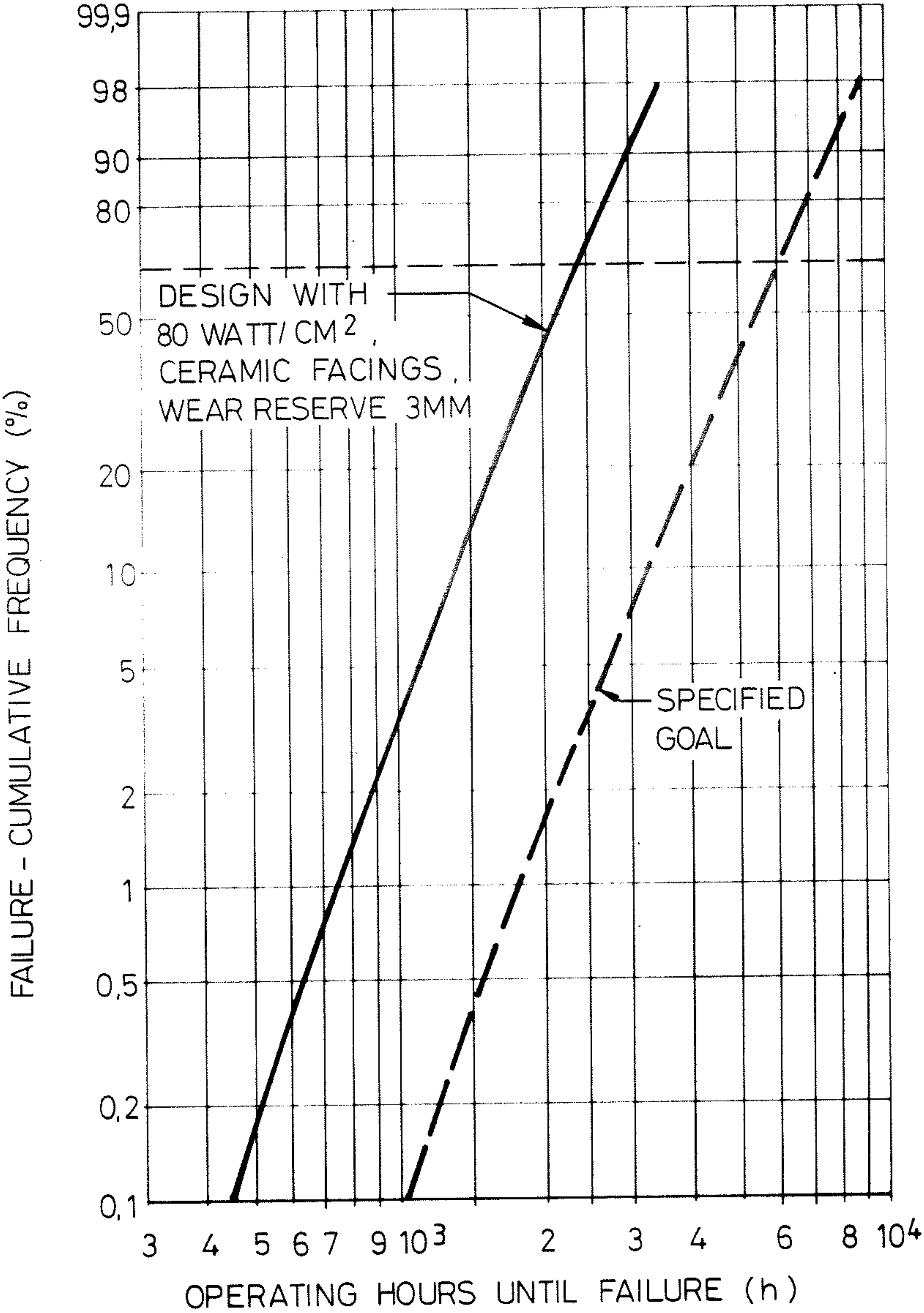
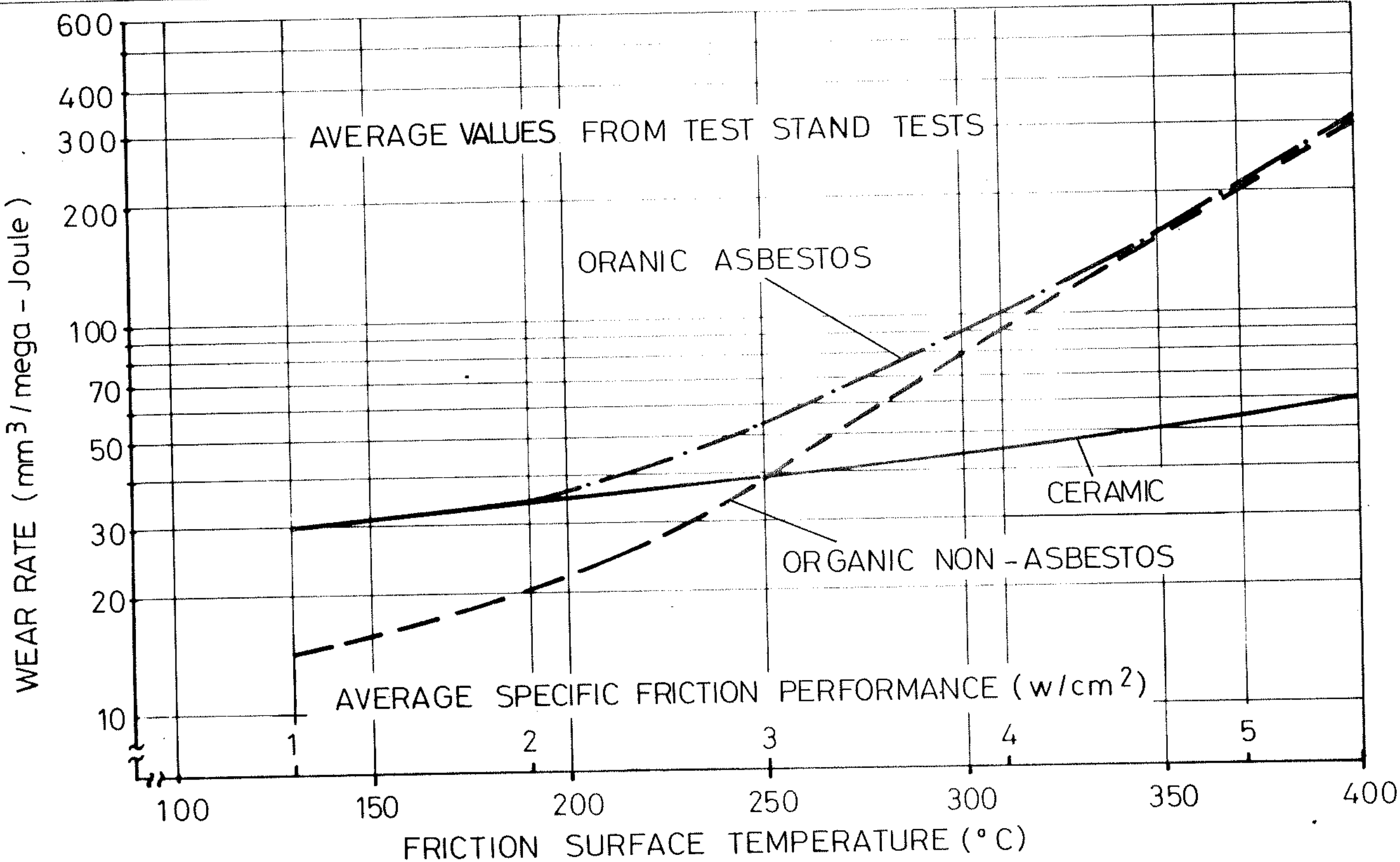


10 03 86	ASSUMED STRESS DISTRIBUTION FOR A TRACTOR POPULATION	LUK



11 03 86	RELIABILITY BASED ON FACING WEAR - SAMPLE CALCULATION



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WEAR RATE VS. FACING MATERIAL AND TEMPERATURE

FACING MATERIAL		ORGANIC, NON - ASBESTOS			CERAMIC			
SIZING (w/cm^2)		50	55	(60)	60	65	70	1
WEAR RESERVE (mm)		3,3	4,7	(6,4)	4,5	5	5,6	2
SERVICE LIFE (h) AT CUM. FAILURE FREQ.	0 %	1000	1000	(1000)	1000	1000	1000	3
	10 %	3 300	3 400	(3 500)	3 000	3000	3000	4
	20%	4 200	4 400	(4 600)	3 700	3 700	3 700	5
	62,5%	6 820	7 380	(7 900)	5 400	5 400	5 500	6

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REQUIRED CHARACTERISTIC SIZING VALUES FOR THE SPECIFIED RELIABILITY GOAL

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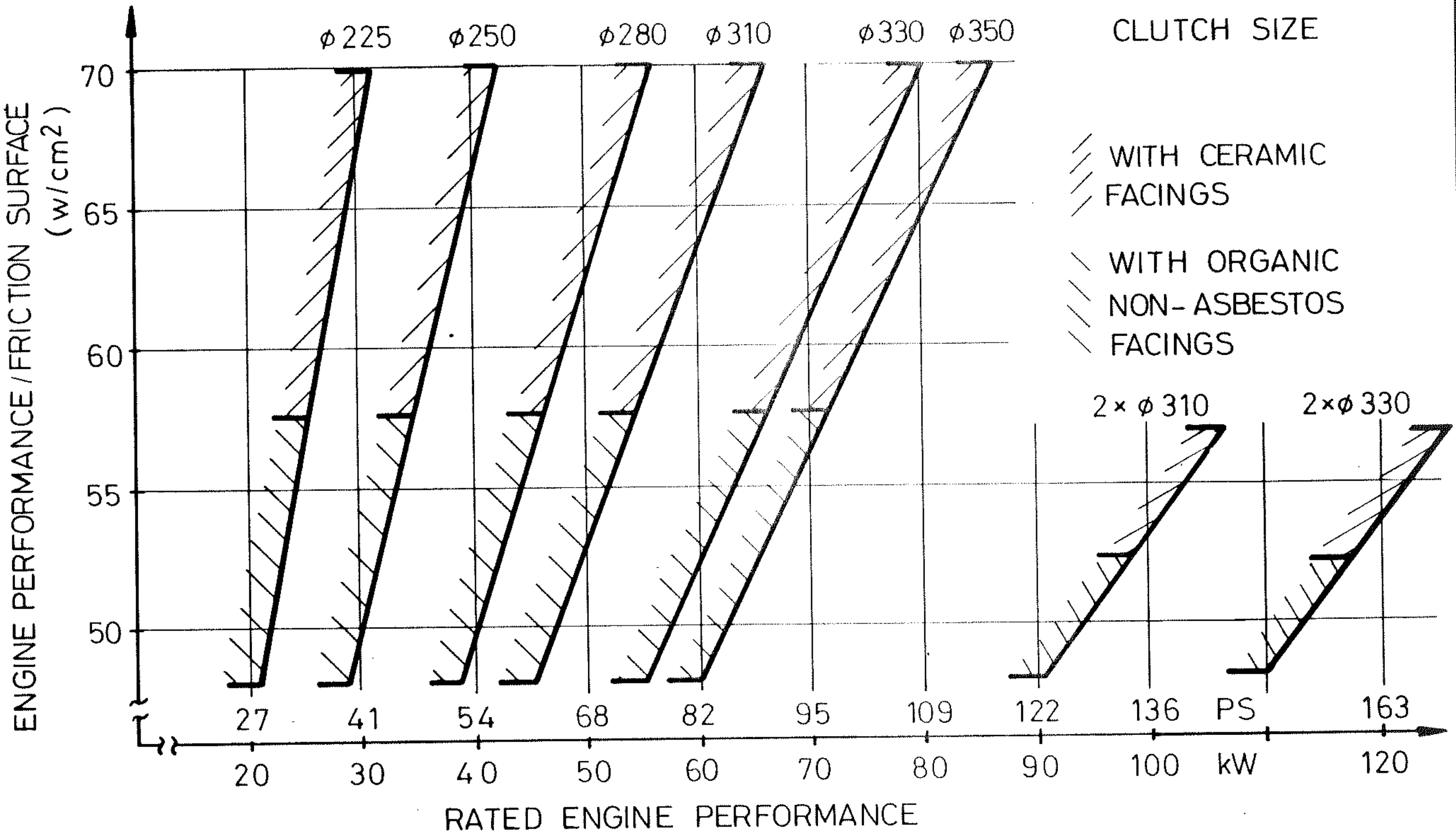
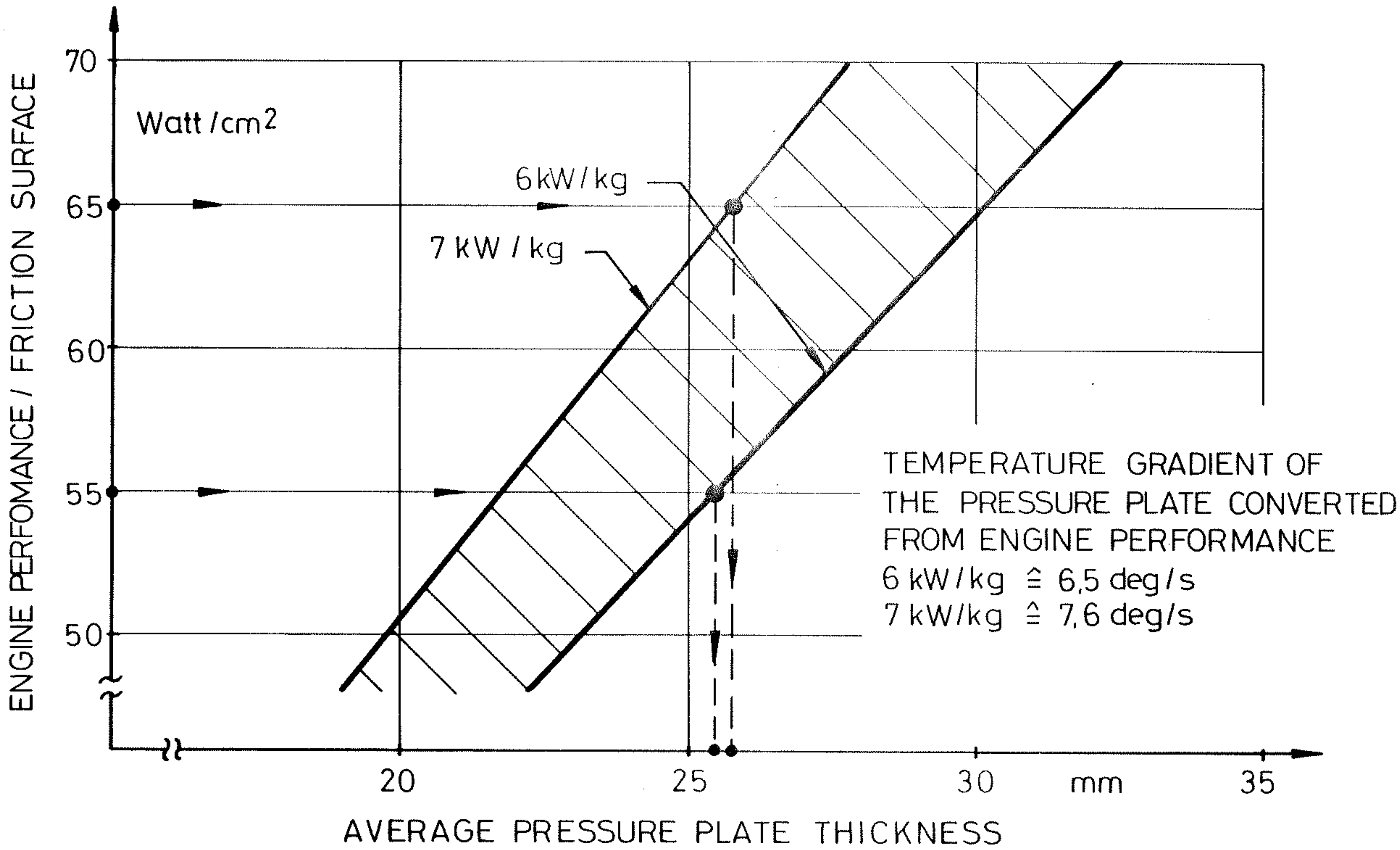


14 03 86	APPLICATION LIMITS FOR VARIOUS CLUTCH SIZES FOR THE SPECIFIED RELIABILITY GOALS	LUK



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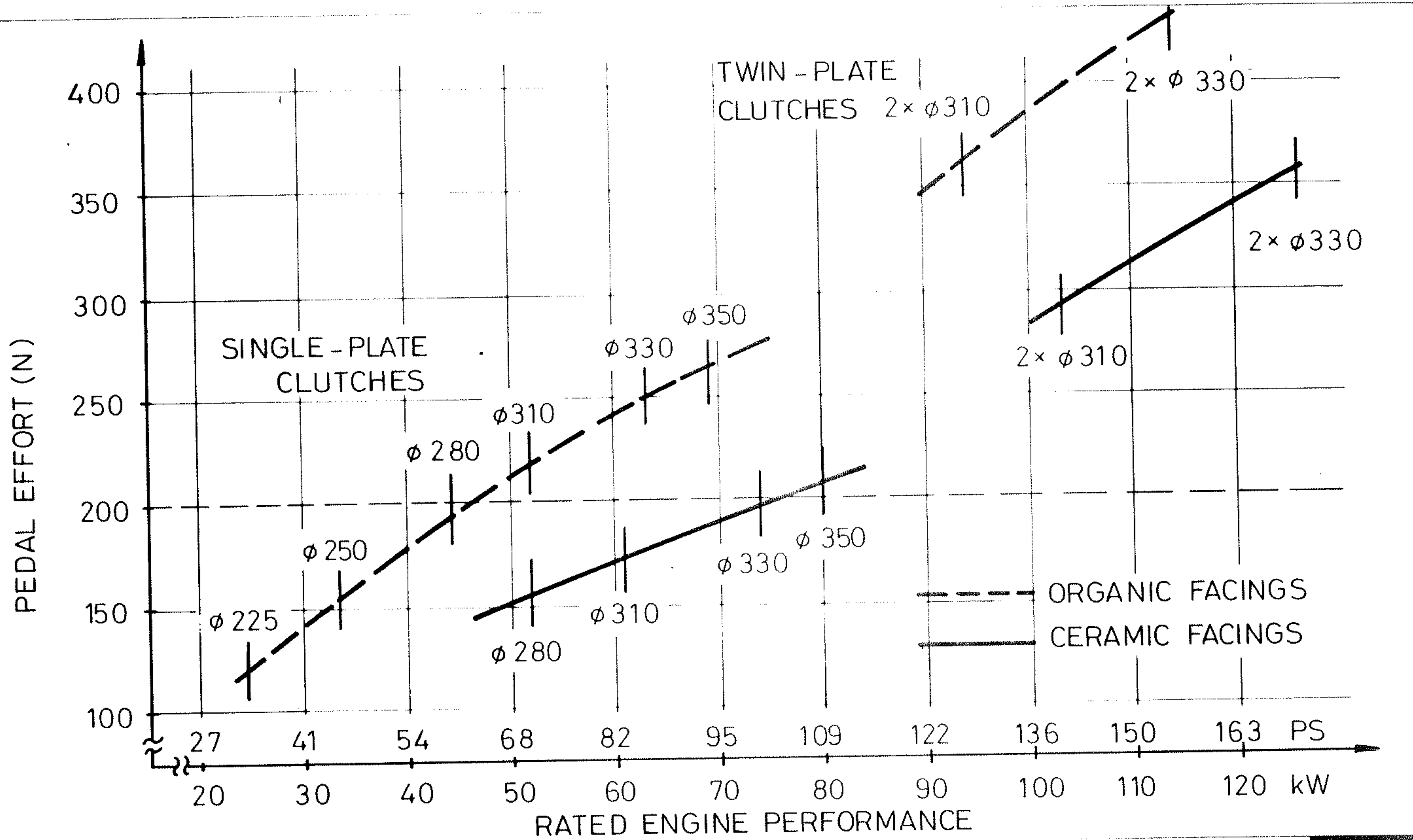
PRESSURE PLATE THICKNESS VS. CHARACTERISTIC SIZING VALUES

LUK



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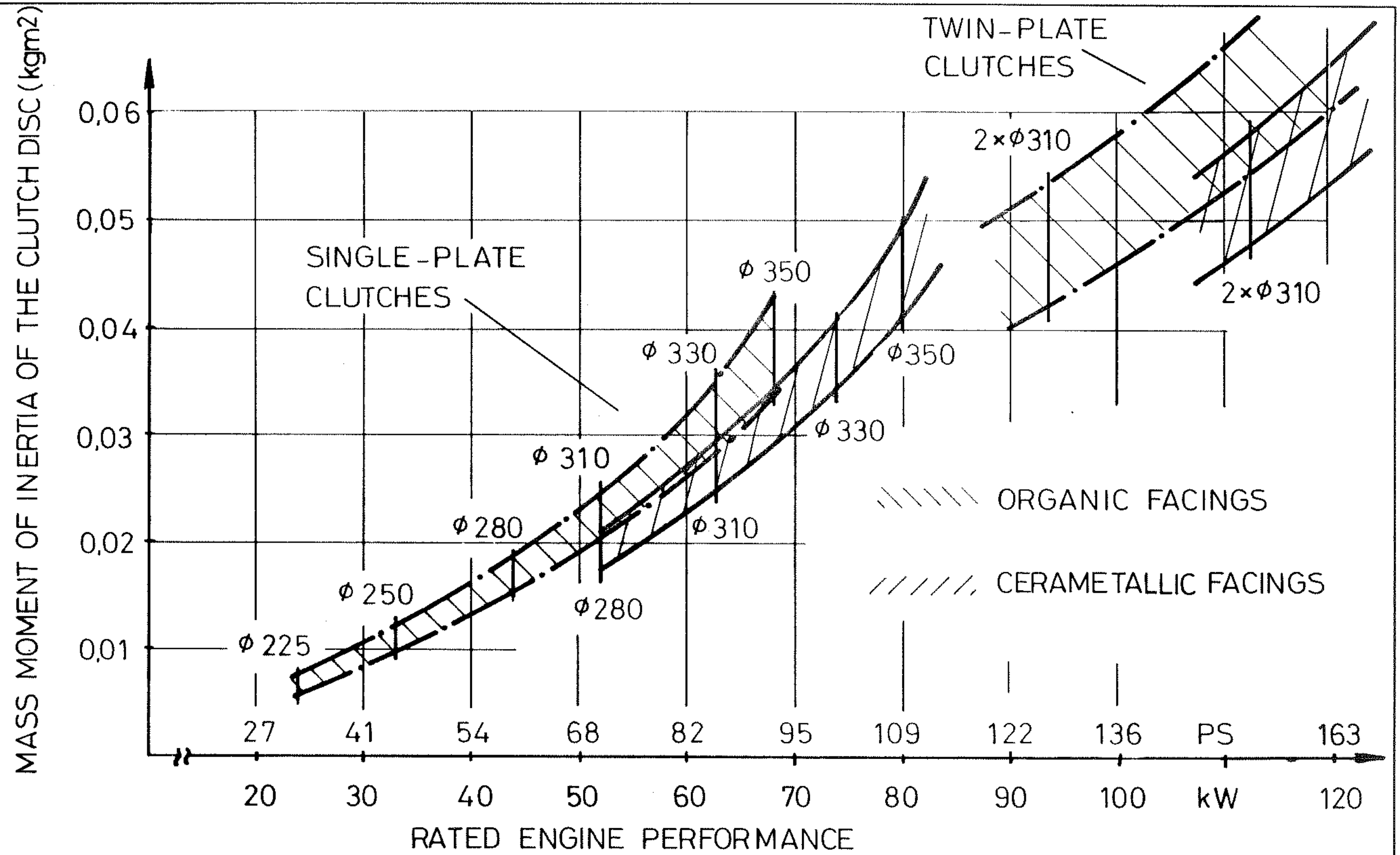
PEDAL EFFORT VS. ENGINE PERFORMANCE AND FACING MATERIAL

LUK



17 03 86	INERTIA OF THE CLUTCH DISC VS. ENGINE PERFORMANCE AND FACING MATERIAL

Kupplungen
Clutches
Embrayages
Embragues

OPTIMIZED CLUTCH DESIGN
RELEASE LOAD AND OPERATING COMFORT

DIPL.-ING. PAUL MAUCHER

APRIL 1986

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OPTIMIZED CLUTCH DESIGN

RELEASE LOAD AND OPERATING COMFORT

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OPTIMIZED CLUTCH DESIGN

RELEASE LOAD AND OPERATING COMFORT

Introduction

We have to take the following criteria into consideration when designing clutches (Figure 1):

- satisfactory service life
- reliable transmission of engine torque
- low release system losses
- low release bearing load
- reliable disengagement
- smooth engagement.

In addition to service life and transmission of engine torque, operating comfort is an essential consideration.

Early stages of motor vehicle clutch design used clutch discs without any cushion, which obviously resulted in little engagement comfort. As a result, the clutch lift-off could be lower than it is today.

It didn't take long for the introduction of clutch discs with cushion deflection to improve modulation. In order to ensure good clutch disengagement, it was necessary to increase the clutch lift-off.

Because of recurrent problems with disengagement and engagement, there has been a tendency to increase clutch lift-off and cushion deflection. As a result, today we often have very high cushion deflection and clutch lift-off, together with the disadvantage of high release bearing load.

High release bearing loads increase friction and elastic losses in the release system and consequently lead to even higher pedal effort and deterioration of engagement performance. This means that measures that were originally intended to improve clutch modulation can have exactly the opposite effect.

The following paper will deal with those criteria which lead to optimum clutch design. Statements regarding heat stress and the clutch safety factor apply specifically to passenger car clutches. The other observations apply in general.

Kupplungen
Clutches
Embrayages
Embragues

1. Satisfactory Service Life

The primary stress on the clutch occurs during vehicle start-up. Figure 2 shows a schematic representation of a start-up cycle. The friction work generated by the speed differential between the engine and the transmission is converted into heat. The clutch has to be designed to prevent overheating and excessive facing wear. The friction work per unit of facing friction surface--designated as specific heat stress 'a'--may not exceed certain limits.

Based on many years of experience, LuK GmbH uses the following engine speeds to calculate the heat stress:

Start-up on level ground:

$$n_{eng} = \frac{1}{6} \text{ (engine rpm at peak torque)} + 1000 \text{ rpm}$$

and on a 26% grade:

$$n_{eng} = \frac{1}{6} \text{ (engine rpm at peak torque)} + 2000 \text{ rpm}$$

Based on our experience, LuK uses the following limit values for the heat stress:

- Start-up on level ground:
  26 Nm/cm^2 to max. 32 Nm/cm^2

- Start-up on a 26% grade:
  max. 300 Nm/cm^2

Figure 3 shows the heat stress for a representative group of passenger car clutches for vehicles with between 1 liter and 3 liter engine displacement. The lower part of the graph shows the values for start-up on level ground. The limit values 26 and 32 Nm/cm^2 cited above are shown as broken lines.

It is obvious that in practice the heat stress is not the same for all vehicles. It increases with increasing engine displacement. The average for vehicles with 1 to 1.2 liter engines is about 22.5 Nm/cm^2; for vehicles with 2.6 to 3 liter, it is about 32 Nm/cm^2. The solid line shows the average trend.

The reason for this is that drivers of vehicles with small engines often start up at higher rpms than do drivers in vehicles with larger engines.

They also shift more frequently, exerting correspondingly higher friction work on the clutch. In addition, the wear reserve of small clutches is often reduced.

Therefore clutch design should strive to adhere to the specific heat stress values for start-up on level ground, represented by the solid line. The values represented by the dash-dot line should not be exceeded.

The top graph shows heat stress for start-up on a 26% grade. Most of the values lie below the 300 Nm/cm^2 limit. In this case the average heat stress is nearly constant over the entire range. Only the 3 vehicle groups 1.8 to 2 liter, 2.2 liter and 2.4 liter significantly exceed the limit value. These cases involve diesel vehicles with unfavorable drive line ratios. Grade-ability and the allowable trailer load of these vehicles is limited in comparison to standard designs.

2. Reliable Transmission of Engine Torque

Reliable transmission of engine torque is essential, even under extreme conditions. The criterion for this function is the clutch safety factor, that is, the ratio between the clutch torque capacity and the engine torque. The clutch torque capacity itself is calculated based on the mean friction radius, the clutch clamp load and the coefficient of friction of the clutch facings.

The clutch safety factor for asbestos facings should be min. 1.2, calculated with a coefficient of friction of 0.27. Until now, our goal for non-asbestos facings has been min. 1.3. Based on current test results, we will again be able to calculate with a clutch safety factor of 1.2 for some non-asbestos materials.

In Figure 4 the clutch safety factor is plotted versus the engine displacement. The average value for each engine displacement class is marked with a plus sign. As you can see, the average clutch safety factor is over 1.4. In many cases we could visualize reducing the clutch safety factor, and with it the clamp load, by up to 25%.

3. Low Release System Losses

The release system consists of a series of load transmitting components, such as the lever, bearing, control cables and return springs, all of which contribute more or less significant losses due to friction or elasticity. Some of these losses are very high and have considerable effect on pedal effort and clutch modulation.

At LuK we have measured various mechanical and hydraulic release systems. Figure 5 shows the results. The table in the middle of the slide shows the total loss V_{tot}, divided up into load loss V_F and travel loss V_S. The resulting total loss values for hydraulic release systems amount to up to 40%, and up to 55% for mechanical systems. The graph at the top of the illustration shows the effect on the pedal effort. The broken line shows the theoretical pedal effort curve vs. the calculated pedal travel based on the clutch release bearing load and the lever ratio of the release system without any losses. The solid curve represents a measured pedal effort curve. This example clarifies the effect of losses. The measured pedal effort is about 50% higher and the measured pedal travel about 40% greater than these values would be if there were no losses in the system.

In many cases it is possible to reduce loss significantly with an acceptable cost. If, for instance, we can reduce the total loss in the release system from 50% to 40%, we can cut the pedal effort by 20%. It is often easier to achieve pedal effort reduction in this area than by making changes in the clutch.

4. Cushion Characteristic, Diaphragm Spring Characteristic and Clutch Lift-off

4.1 Relationships within the Clutch

Clutch torque build-up during clutch engagement and release bearing load are dependent on the cushion deflection curve, on the diaphragm spring characteristic, and on the clutch lift-off. Some essential aspects of clutch engagement performance were discussed during the last clutch symposium under the topic "Operating and Engagement Performance of Motor Vehicle Dry Friction Clutches." That presentation studied and illustrated primarily the effect of friction and elasticity in the release system on engagement performance.

Our purpose in this presentation is to show the effects of the cushion and diaphragm spring characteristics as well as the amount of clutch lift-off on the release bearing load and clutch modulation.

Figure 6 illustrates the function of the clutch and the relationships which determine the loads. The clutch is mounted on the flywheel. The disc is clamped between the clutch pressure plate and the flywheel friction surface. The clutch disc facings are cushioned axially by the spring segments. A diaphragm spring supported in the clutch cover presses the pressure plate against the clutch disc. The diaphragm spring fingers enable the clutch to disengage.

There are three primary loads on the clutch: the cushion deflection load, the diaphragm spring load and the release bearing load. The three loads are maintained in constant equilibrium, that is, when the clutch is disengaged, during clutch engagement and when engaged. In each case the (Diaphragm Spring Load F_T minus

Cushion Deflection Load F_B) times Lever Arm a equals Release Bearing Load F_A times Lever Arm b (Equation # 1). Based on this equation, we can derive that the diaphragm spring load minus the cushion deflection load is equal to the release bearing load multiplied by the lever arm ratio, that is the lever ratio of the diaphragm spring in the clutch (Equation # 2).

Consequently the diaphragm spring load minus the cushion deflection load together with the lever ratio of the clutch determines the clutch release bearing load (Equation # 3).

This also means that the cushion deflection load F_B is equal to Diaphragm Spring Load F_T minus Release Bearing Load F_A times Lever Arm b divided by Lever Arm a (Equation # 4). This formula can be used for determining the effective cushion load and thus the clutch torque capacity during the engagement cycle.

Figure 7 shows these relationships based on the clutch characteristic. The broken line represents the diaphragm spring load curve. The cushion deflection characteristic is drawn in as a dash-dot line. The cushion deflection curve must intersect with the diaphragm spring curve at the operating point. In this condition, without release bearing load, the diaphragm spring load has to be equal to the cushion deflection load. The diaphragm spring load minus the cushion deflection load yields the release bearing load multiplied by the clutch lever ratio, shown in the graph as a solid curve.

This is true if we assume that all the other elements in the clutch are rigid. Only then are the relationships between the loads and the respective travel values easy to calculate. In reality the relationships become more complex because of the elastic deformation of various clutch components in conjunction with the non-linear diaphragm spring and cushion deflection characteristics.

4.2 Influence of the Cushion Deflection and Diaphragm Spring Characteristics on Engagement Performance

We conducted clutch measurements with the goal of determining the effects of the diaphragm spring and cushion deflection characteristics on engagement performance or on clutch torque build-up. For this purpose we used clutches with extreme diaphragm spring and cushion deflection characteristics.

The left-hand graph in Figure 8 shows curves for the two clutches we used. Clutch A has a so-called steep characteristic, whereas clutch B exhibits a flat characteristic. Consequently, as can be seen from the graph, the wear capacity of clutch B is lower than that of clutch A.

The graph on the right shows differing cushion deflection curves. Clutch disc 1 demonstrates a strongly progressive curve with a very flat initial slope, while, in contrast, clutch disc 2 is not at all progressive.

The following Figure 9 compares the engagement characteristic and the clutch torque build-up for clutch discs 1 and 2 combined with both clutches A and B. As the graph shows, clutches A and B have absolutely the same engagement characteristic for the same cushion deflection curve, although they represent extremely different diaphragm spring characteristics. In each case the deciding influence is the cushion deflection characteristic.

We have to conclude therefore that the diaphragm spring characteristic--specifically whether it is steep or flat--has practically no influence on clutch torque build-up, that is on clutch engagement performance.

4.3 Release Bearing load Comparison

Figure 10 illustrates the release bearing load curve for the clutch disc with the strongly progressive characteristic curve. It is shown for the clutches A and B used in the previous discussion--one with a very steep and one with a flat diaphragm spring characteristic.

The top graph shows the release bearing load for the new clutch, and the bottom one, the release bearing load after 1.5 mm facing wear. The solid line represents the release bearing load of clutch A with a steep diaphragm spring characteristic. The broken line represents that for clutch B with a flat characteristic.

One can see that the steep diaphragm spring characteristic results in significant advantages for the release bearing load of a new clutch. For the new condition, clutch A with 1100 N release bearing load is significantly better than clutch B with about 1500 N. Once the facing has worn, however, the release bearing load for clutch A, as a result of its steeper diaphragm spring characteristic, increases faster than it does for clutch B. After 1.5 mm facing wear, clutch A still has a small advantage (see bottom graph).

This all speaks well for a relatively steep diaphragm spring characteristic.

4.4 The Influence of the Clutch Lift-off, or the Clutch Lever Ratio, and the Cushion Deflection Characteristic on Operating Comfort

In the preceding section clutches with identical lift-off values were used for better representation of the influence of the cushion deflection and the diaphragm spring characteristic curves on engagement performance and release bearing load.

Based on the torque equation for determining the loads (Figure 6, Equation 3), it was shown that the clutch release bearing load is directly dependent on the clutch lever ratio, hence release bearing load decreases as the lever ratio increases. At the same time the clutch lift-off also decreases.

The question remains as to how the reduction in clutch lift-off affects the rise in clamp load and torque build-up during the engagement cycle, thus influencing clutch operating comfort.

First we will compare two typical examples of push-type clutches with extremely different lift-off values of 1.75 and 1.35 mm at 7 mm release travel.

In order to obtain a meaningful comparison, we used the same diaphragm spring characteristic for both clutches, as shown in Figure 11.

Figure 12 shows the cushion deflection characteristics with a cushion of 1.0 and 0.7 mm resp. for the clutch discs used.

Figure 7 used characteristic curves to illustrate the relationship between the diaphragm spring load, the release bearing load and the cushion deflection load, which is equal to the clamp load. Only the area to the right of the vertical line passing through the operating point is critical for release bearing load and engagement performance. The following discussion will be limited to this range.

Up until now we have not taken the elasticity of the clutch components into consideration.

Figure 13 shows the characteristic curves without the effect of the cushion deflection, but taking into consideration the elasticity of the load bearing components in the clutch. The release bearing load is shown as a dash-dot line, the clamp load as a broken line, and the pressure plate lift-off as a solid line, all plotted vs. the release travel. The effective pressure plate clamp load--which I will refer to simply as the clamp load--does not drop abruptly to zero at the beginning of the release travel. It decreases to zero only after a certain release travel--a little more than 2 mm in the example shown. Elastic deformations in the clutch are responsible for this. These are: the diaphragm spring finger deflection, the resiliency of the clutch cover and the elasticity of the diaphragm spring supports. All of these act like a linear cushion. Therefore, during the engagement cycle, the clamp load does not increase abruptly, even without any cushion deflection, but rather builds up almost linearly over part of the engagement travel. This portion of the travel varies in direct proportion to the elasticity of the clutch. However, the elasticity of the clutch does not usually ensure smooth engagement performance. Additionally, we need a clutch disc with progressive cushion deflection.

Figure 14 shows the characteristic curves for the clutch with the diaphragm spring characteristic as shown in Figure 11. Here we take into consideration the elasticity of the clutch and the effect of the cushion deflection load. The solid lines represent the individual characteristics for the new clutch, and the broken lines for 1.5 mm facing wear.

While releasing the clutch, the pressure plate releases the load on the clutch facings. In this process, the clamp load, which is always equal to the cushion deflection load, decreases toward zero, beginning at the operating point. As the release travel continues, beginning after about 5 mm, the pressure plate lifts off the clutch facings to provide clearance equal to the pressure plate lift-off minus the cushion travel.

When we compare this example to the previous one without cushion deflection, it is obvious that the clamp load build-up begins much earlier due to the effect of the cushion deflection. At first the load rise is very flat and increases progressively with increasing engagement travel. The clamp load and clearance curves are essentially parallel for new condition and for 1.5 mm facing wear. They are simply displaced somewhat because of increased elastic deformation due to the higher diaphragm spring load and release bearing load as a result of wear. The release bearing load is greater after wear than in new condition because of the steep diaphragm spring characteristic we chose to use.

The relationship between new condition and after 1.5 mm facing wear is similar for both clutch examples. Therefore in the following comparison we have omitted the curves for clamp load and clearance after 1.5 mm facing wear.

Figure 15 compares two push-type clutch assemblies, K1 and K2, with different lift-off values. K1 has a high lift-off and cushion deflection, K2 a relatively low lift-off and a correspondingly low cushion deflection. The curve of the pressure plate clamp load exerted on the clutch disc with respect to the release travel and the clearance are almost identical, although the clutch lift-off for the two clutches is very different.

It is plain to see that given the proper adjustment of the cushion deflection characteristic to the pressure plate lift-off, we can achieve the same engagement characteristic. This is independent of the magnitude of the lift-off, therefore vehicle start-up performance is not dependent upon the amount of lift-off.

Figure 16 compares the release bearing loads for the clutches tested. As might be expected, the push-type clutch K2 with the low lift-off and reduced cushion deflection exhibits an essentially lower release bearing load than clutch K1 with the higher lift-off and cushion deflection.

Theoretically we can expect that the two clutches will exhibit the same engagement performance. To confirm this, we evaluated them in the same vehicle. The pedal effort for K1 was 130 N, and about 100 N for K2. Engagement performance was good in both cases. Clutch K2, with the low lift-off and cushion deflection, tended to be better.

This can be attributed to lower loads and consequently lower friction losses, enabling more sensitive clutch operation.

4.5 Comparing Push-type and Pull-type Clutches

It is generally assumed that a pull-type clutch has a significantly better release bearing load than a push-type clutch. Therefore in Figure 17 we compare a pull-type clutch assembly K3 and the push-type assembly K2 from the previous example. Both clutches have the same lift-off, the same diaphragm spring characteristic, and the same cushion deflection characteristic.

The clamp load build-up for both clutches runs parallel over the first part of the engagement travel, that is, both clutches exhibit equally good engagement performance in this range. Toward the end of the engagement travel, the clamp load of the pull-type clutch rises more steeply. This can have a slight adverse effect on engagement characteristics when the clutch is almost locked up.

Figure 18 compares the release bearing loads of the pull-type clutch K3 and the push-type clutches K1 and K2. The push-type clutch K1 with the high lift-off and cushion deflection has--as expected--the highest release bearing load. As already illustrated in Figure

16, the push-type clutch K2, with its low lift-off and reduced cushion deflection, exhibits a 20% reduction in release bearing load. In comparison to K2, the release bearing load for the pull-type clutch K3 is only about 12% lower.

This proves that the magnitude of the clutch lift-off has no influence on engagement performance, but it does affect the release bearing load, as well as engagement comfort. It also demonstrates that, if a push-type clutch is properly designed, it can achieve almost the same release load level as a pull-type clutch.

5. Summary and Conclusions

Within the framework of this presentation, we have discussed the essential factors involved in clutch design, in particular low release bearing load and engagement comfort.

The important variables for determining clutch dimensions, such as heat stress during start-up and the clutch safety factor, have been analyzed for a large number of passenger cars currently on the market. We have noted the high release system losses in some cases.

Based on calculations and measurements we have analyzed the influence of the diaphragm spring and cushion deflection characteristics as well as the clutch lift-off. We have examined the mutual interaction of these factors, and compared the results using typical examples.

It has been shown that there are still many possibilities for optimizing clutch design.

The following values must be taken into consideration when designing a clutch:

5.1 Satisfactory Service Life

The specific heat stress should not exceed the following limit values (Figure 19):

a) Start-up on level ground:
   Engine displacement limit values:
   22 Nm/cm^2 to 40 Nm/cm^2

b) Start-up on a 26% grade: 300 Nm/cm^2

5.2 Reliable Transmission of Engine Torque

A clutch safety factor of 1.2 is required in order to ensure reliable engine torque transmission. Any significantly higher values are to be avoided in order to keep clutch loads low.

5.3 Low Losses in the Release System

Release systems are frequently subject to high losses. In general LuK has no influence on release system design. The automobile manufacturer can exploit reserves in this area to reduce release bearing load.

5.4 Cushion Deflection Characteristic, Diaphragm Spring Characteristic and Clutch Lift-Off

a) Cushion Deflection Characteristic

Engagement performance is determined primarily by the cushion deflection characteristic. Characteristic curves with strong progressive slopes have proven to be advantageous. The beginning of the cushion deflection curve should be as flat as possible.

b) Diaphragm Spring Characteristic

The diaphragm spring characteristic curve doesn't have any significant effect on engagement performance, but it does influence clutch release bearing load and wear reserve. The diaphragm spring should exhibit a load ratio of about 1 to 0.6 between the peak and valley spring curve loads. This will ensure optimum operating comfort in new condition, limited release bearing load build-up after wear and sufficient wear reserve.

c) Clutch Lift-off

If we reduce the clutch lift-off, we can usually reduce the release bearing load considerably without decreasing engagement comfort. We recommend a lift-off of 1.2 mm. The cushion deflection must then be adapted to the reduced lift-off.

d) Pull-type clutches

With pull-type clutches we can reduce the release bearing load by about 12% in comparison to push-type clutches while using comparable lift-off. This advantage sometimes gets lost due to a more complicated release system.

The advantages of pull-type clutches include:

- simpler design and better cooling of the diaphragm spring
- longer diaphragm spring characteristic curve and as a result, higher wear reserve.

It is necessary to note the disadvantages of the pull-type clutch:

- more complicated release system
- difficult installation and removal from the engine and the transmission
- for disassembly, the transmission must be pulled farther from the engine due to the fact that the release bearing is attached to the clutch.

Consequently, a pull-type clutch only makes sense for engines with extremely high performance and sufficient package space, which means its use is limited to large clutches with high heat stress and the need for high wear reserve.

SATISFACTORY SERVICE LIFE

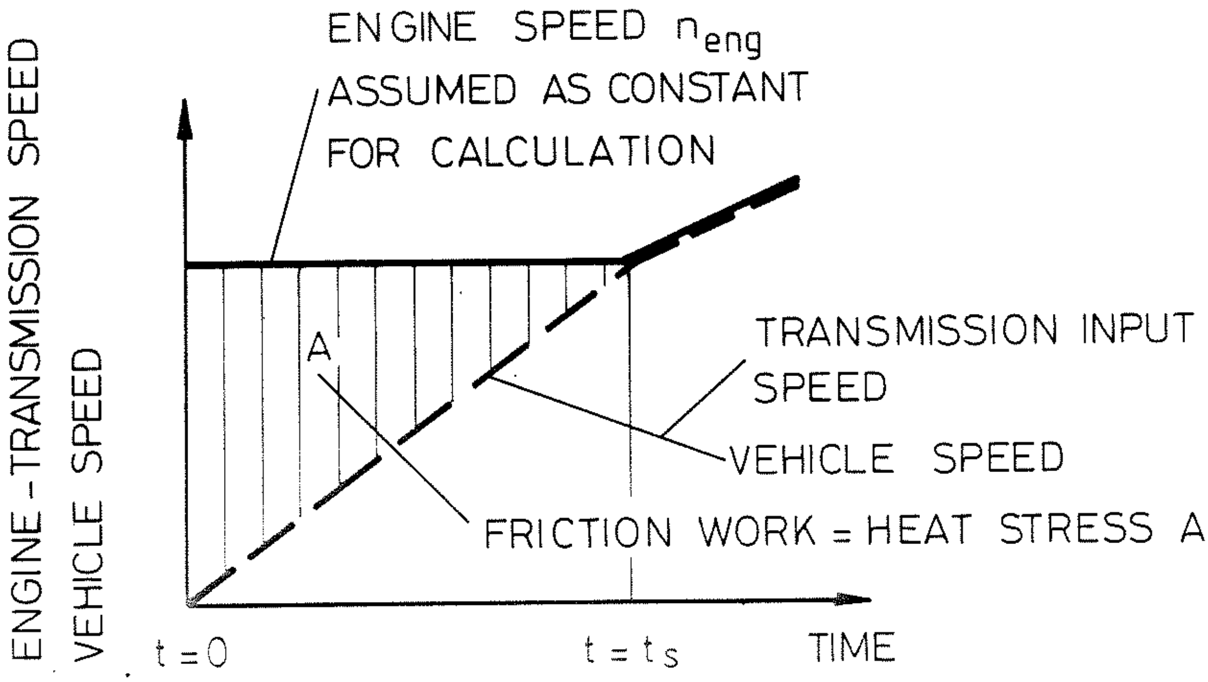
RELIABLE TRANSMISSION OF ENGINE TORQUE

LOW RELEASE SYSTEM LOSSES

LOW RELEASE BEARING LOAD

RELIABLE DISENGAGEMENT

SMOOTH ENGAGEMENT



HEAT STRESS $$A = \int_{t=0}^{t=t_s} M \times \omega_{rel.} \times dt$$

SPECIFIC HEAT STRESS $$a = \frac{A}{F} = \frac{1}{F} \int_{t=0}^{t=t_s} M \times \omega_{rel.} \times dt$$

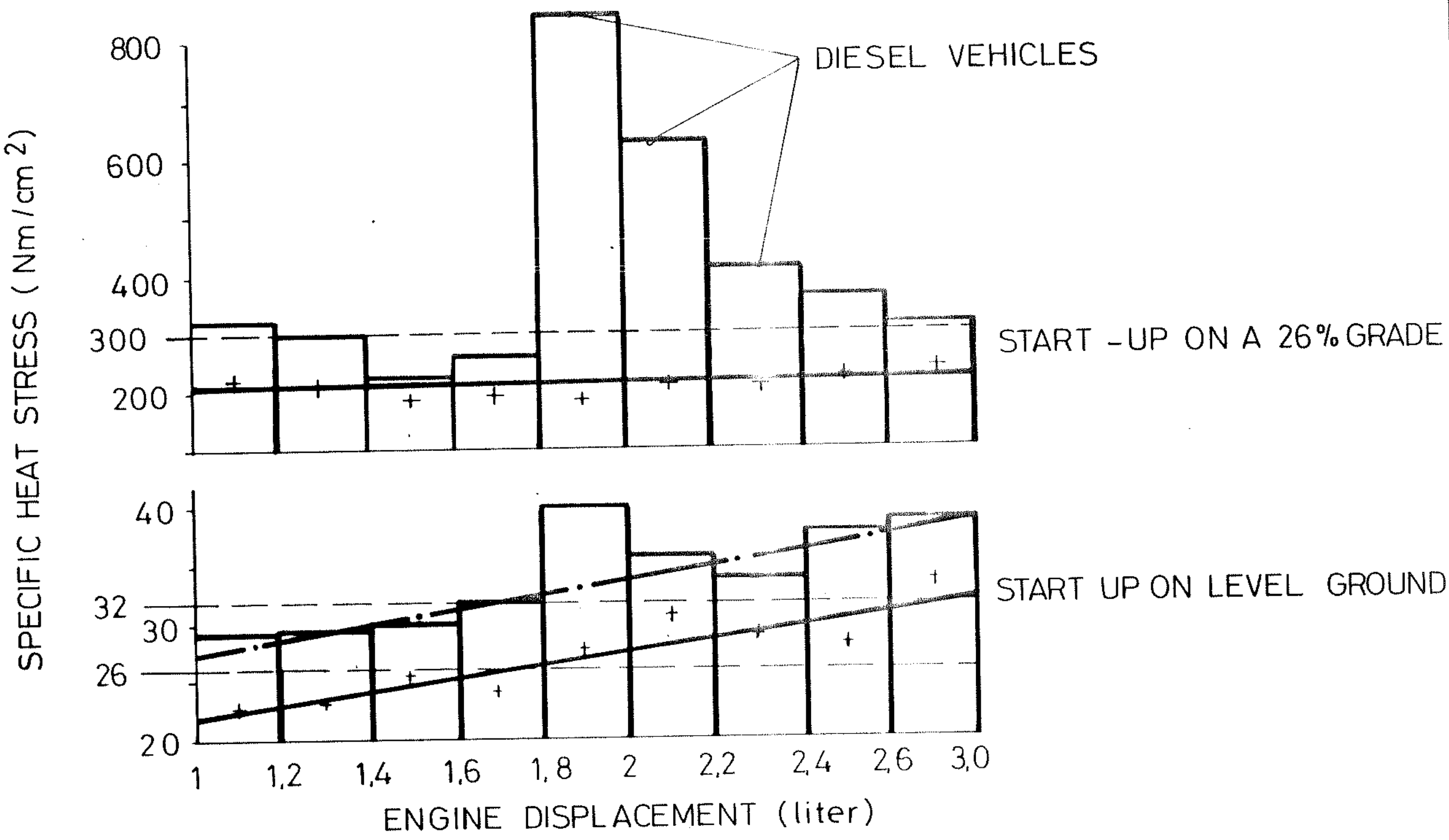
ENGINE SPEED FOR START-UP ON LEVEL GROUND

$n_{eng.} = \frac{1}{6}$ ENGINE RPM AT PEAK TORQUE + 1000 RPM

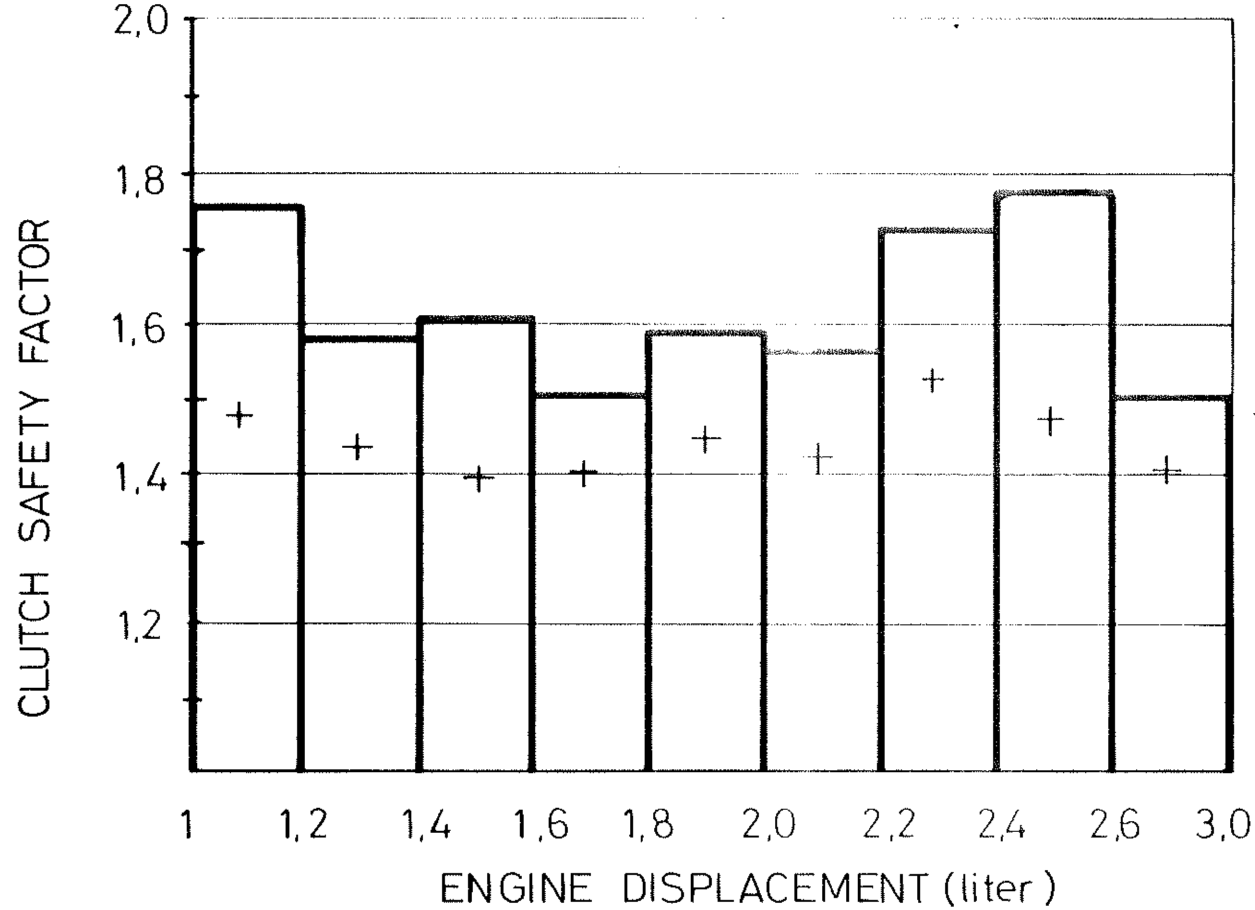
ENGINE SPEED FOR START-UP ON A 26% GRADE

$n_{eng.} = \frac{1}{6}$ ENGINE RPM AT PEAK TORQUE + 2000 RPM

02 04 86	SPECIFIC HEAT STRESS DURING START-UP	LUK



03 04 86	SPECIFIC HEAT STRESS ON PASSENGER CAR CLUTCHES	LUK



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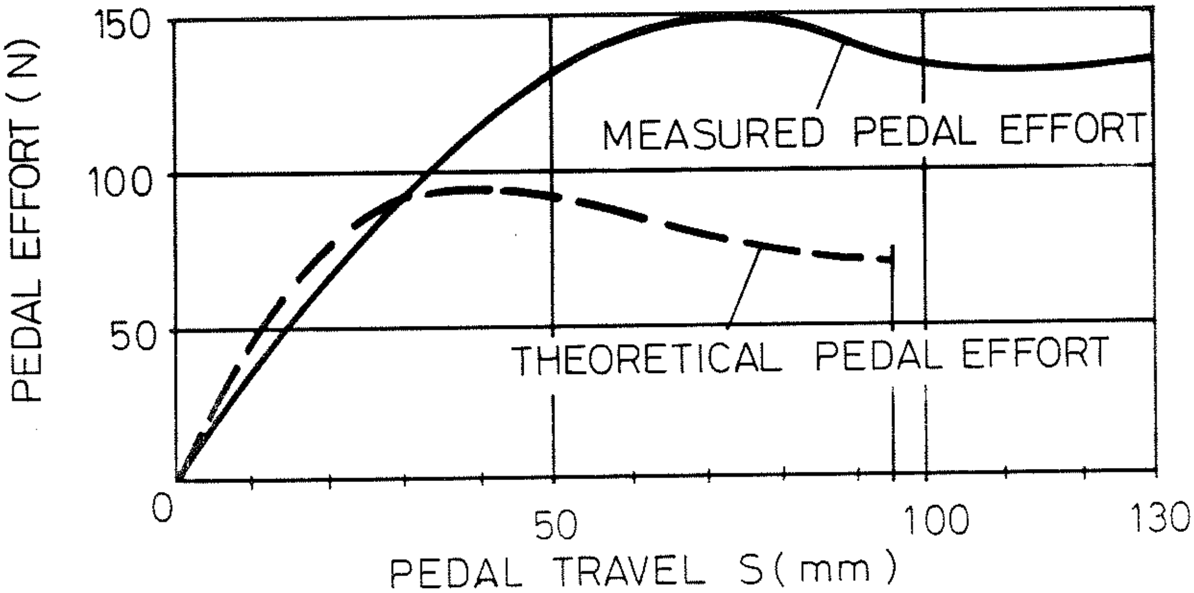
CLUTCH SAFETY FACTOR FOR PASSENGER CAR CLUTCHES

LUK



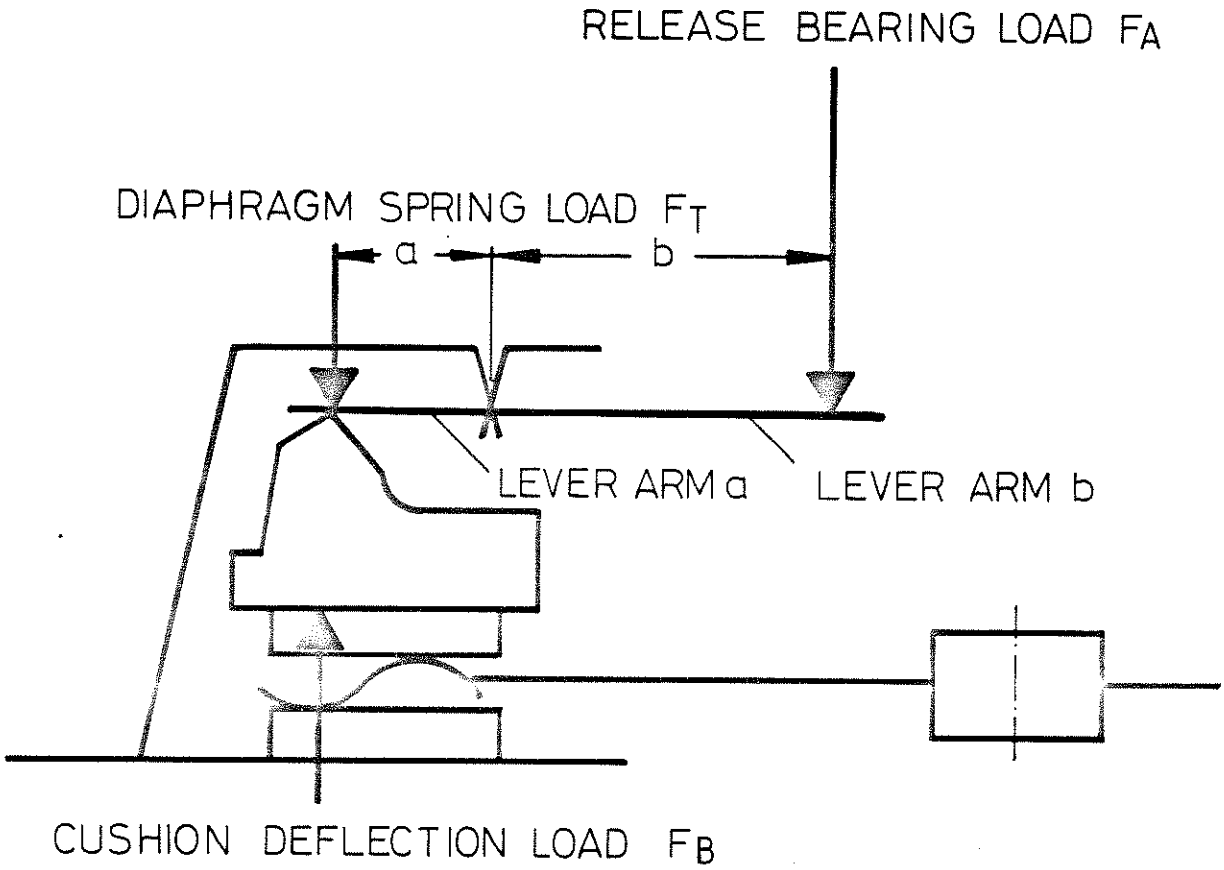
RELEASE SYSTEM	LOSSES		
	$V_{tot.}$ %	V_F %	V_S %
MECHANICAL	32 ÷ 55	20 ÷ 35	15 ÷ 30
HYDRAULIC	28 ÷ 40	15 ÷ 20	15 ÷ 25

$$V_F = \frac{\text{MEASURED PEDAL EFFORT} - \text{THEOR. PEDAL EFFORT}}{\text{MEASURED PEDAL EFFORT}} \times 100\ \%$$

$$V_S = \frac{\text{MEASURED PEDAL TRAVEL} - \text{THEOR. PEDAL TRAVEL}}{\text{MEASURED PEDAL TRAVEL}} \times 100\%$$

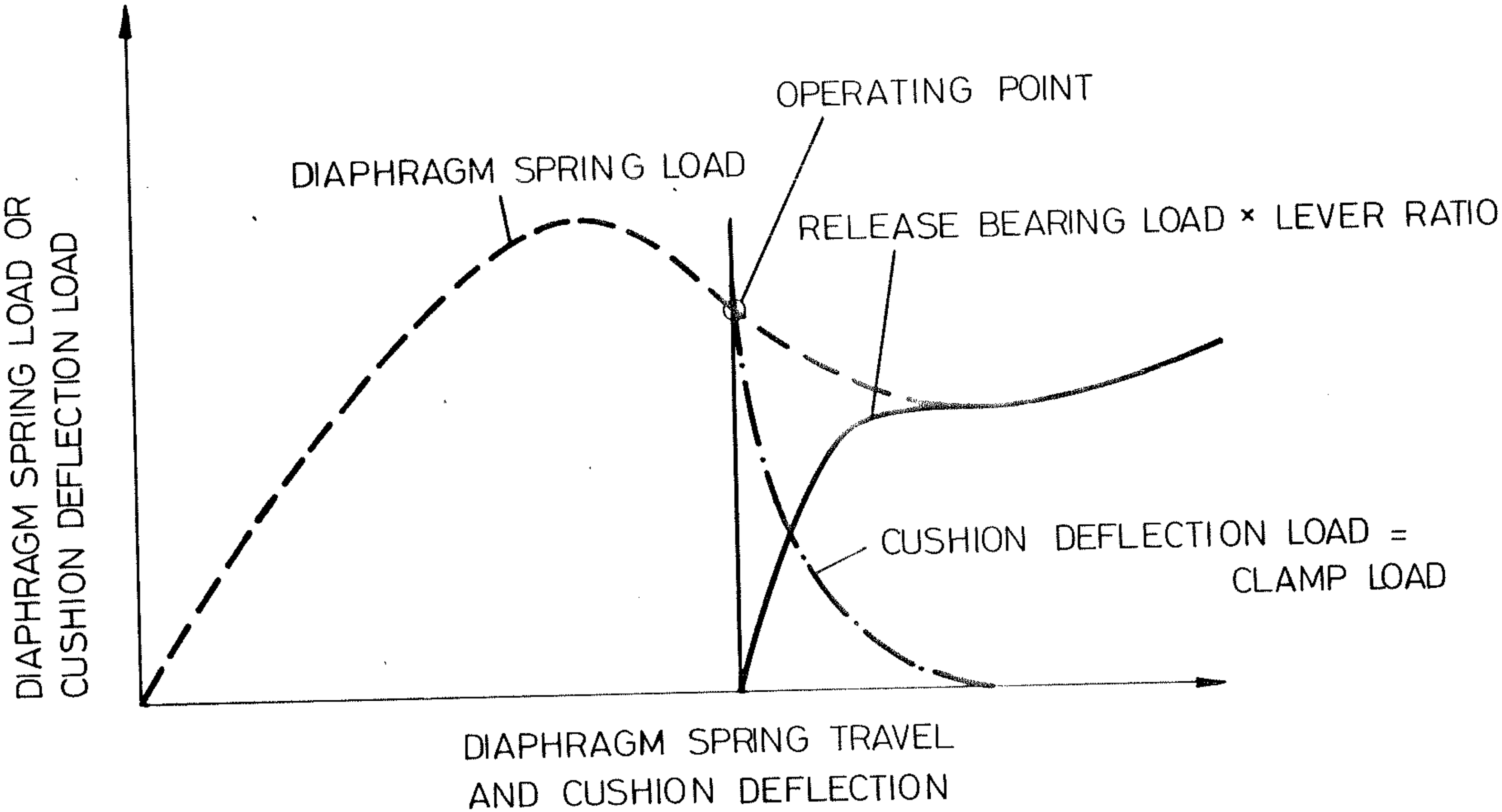
$$V_{tot.} = 100 - \frac{(100 - V_F)(100 - V_S)}{100}\ \%$$

05 04 86	RELEASE SYSTEM LOSSES	LUK

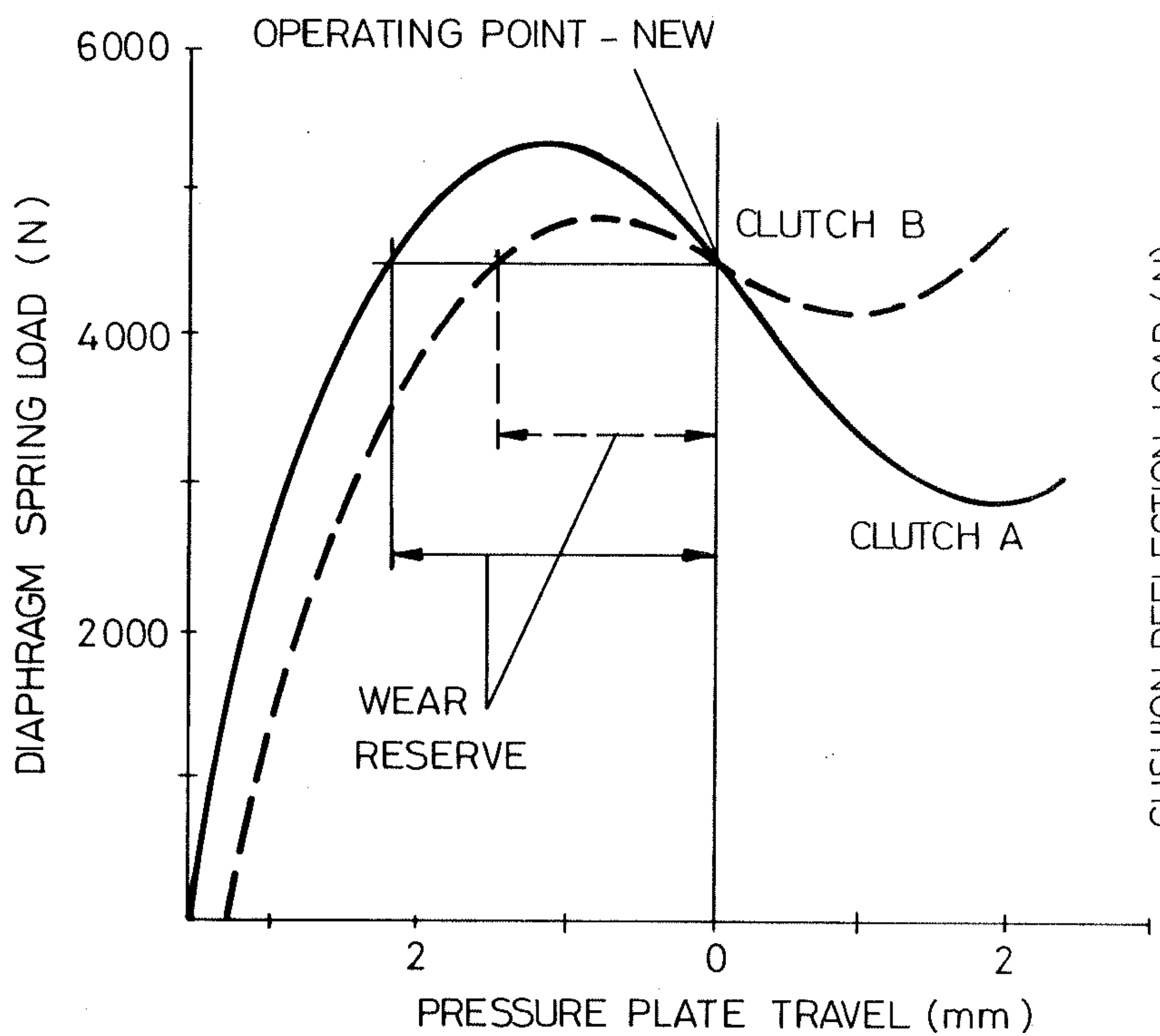


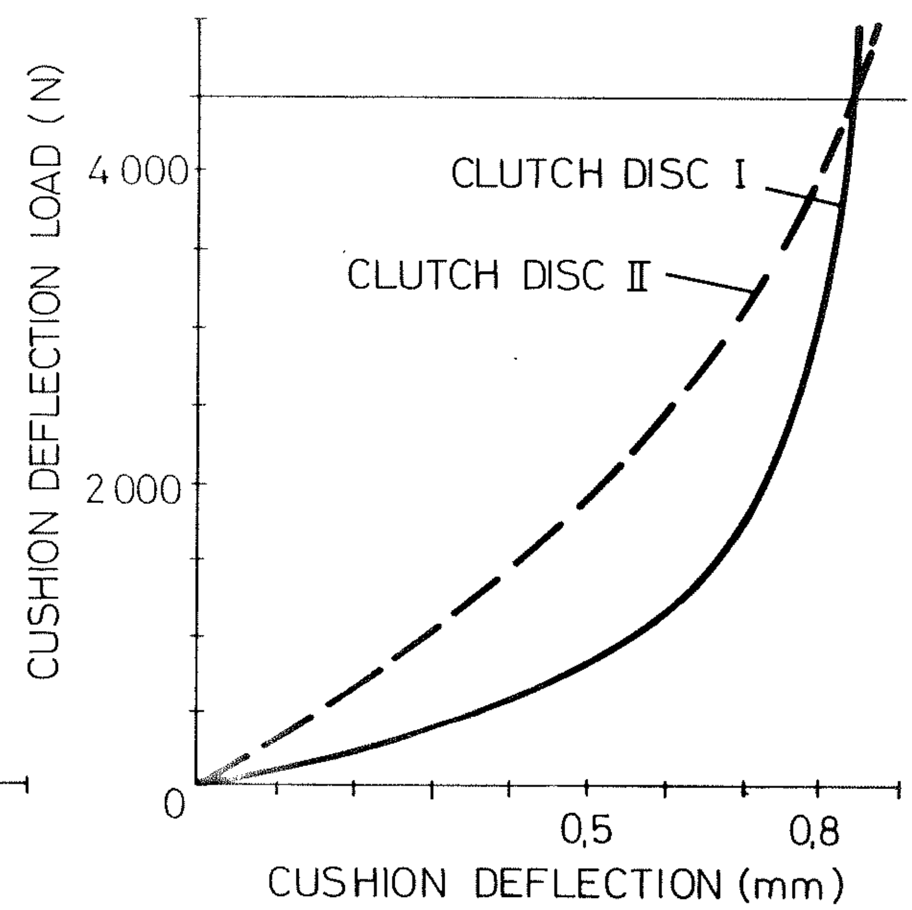
MOMENT EQUILIBRIUM

$(F_T - F_B)\,a = F_A \times b$	EQUATION 1
$F_T - F_B = F_A \times \frac{b}{a}$	EQUATION 2
$F_A = (F_T - F_B)\frac{a}{b}$	EQUATION 3
$F_B = F_T - F_A \times \frac{b}{a}$	EQUATION 4



07 04 86	DETERMINING THE RELEASE BEARING LOAD BASED ON THE DIAPHRAGM SPRING AND CUSHION DEFLECTION CHARACTERISTIC CURVES





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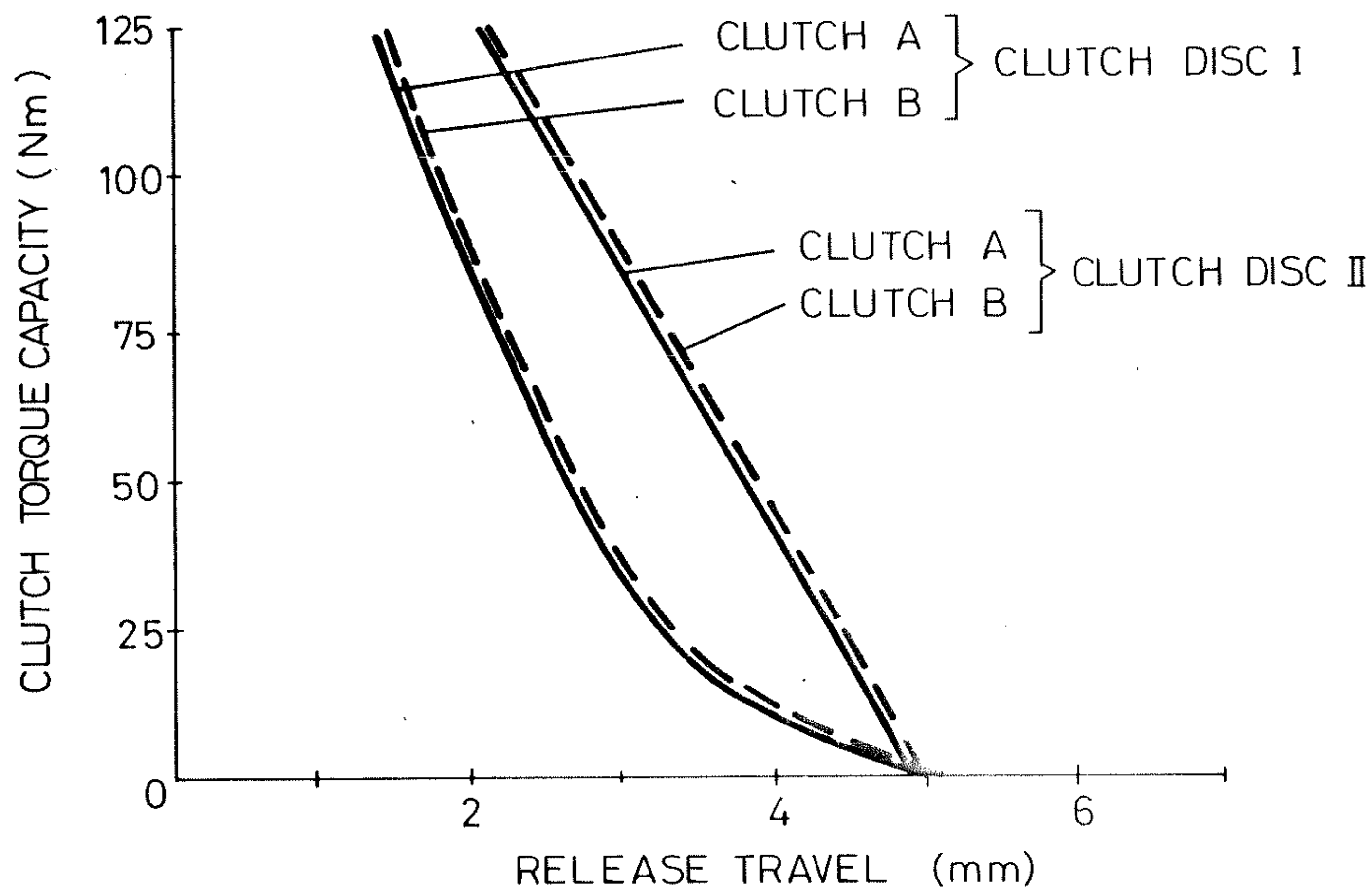
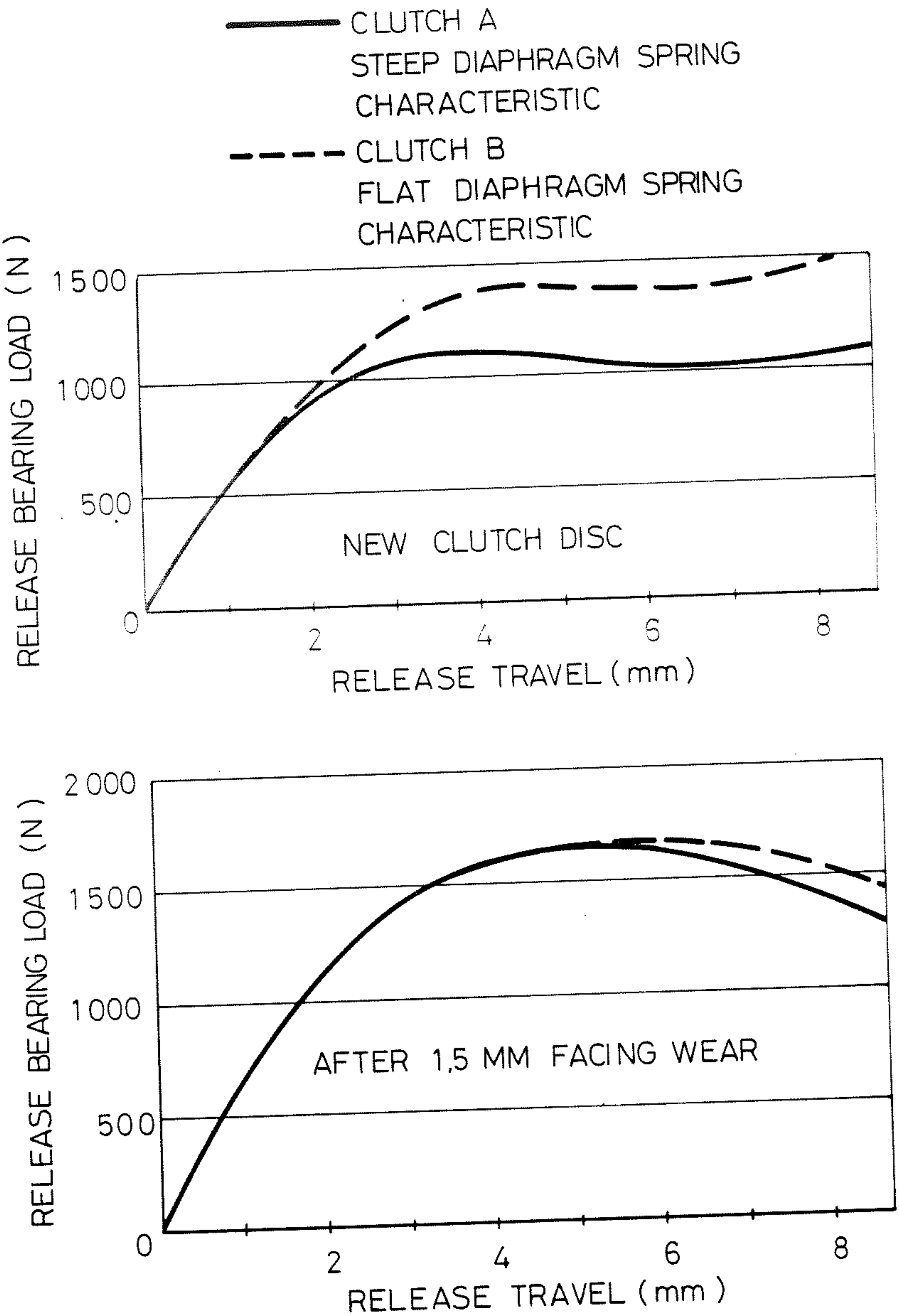
DIAPHRAGM SPRING CURVE

CUSHION DEFLECTION CURVE

LUK



09 04 86	CLUTCH TORQUE COMPARISON	LUK



10 04 86	RELEASE BEARING LOAD COMPARISON BETWEEN CLUTCH A AND CLUTCH B

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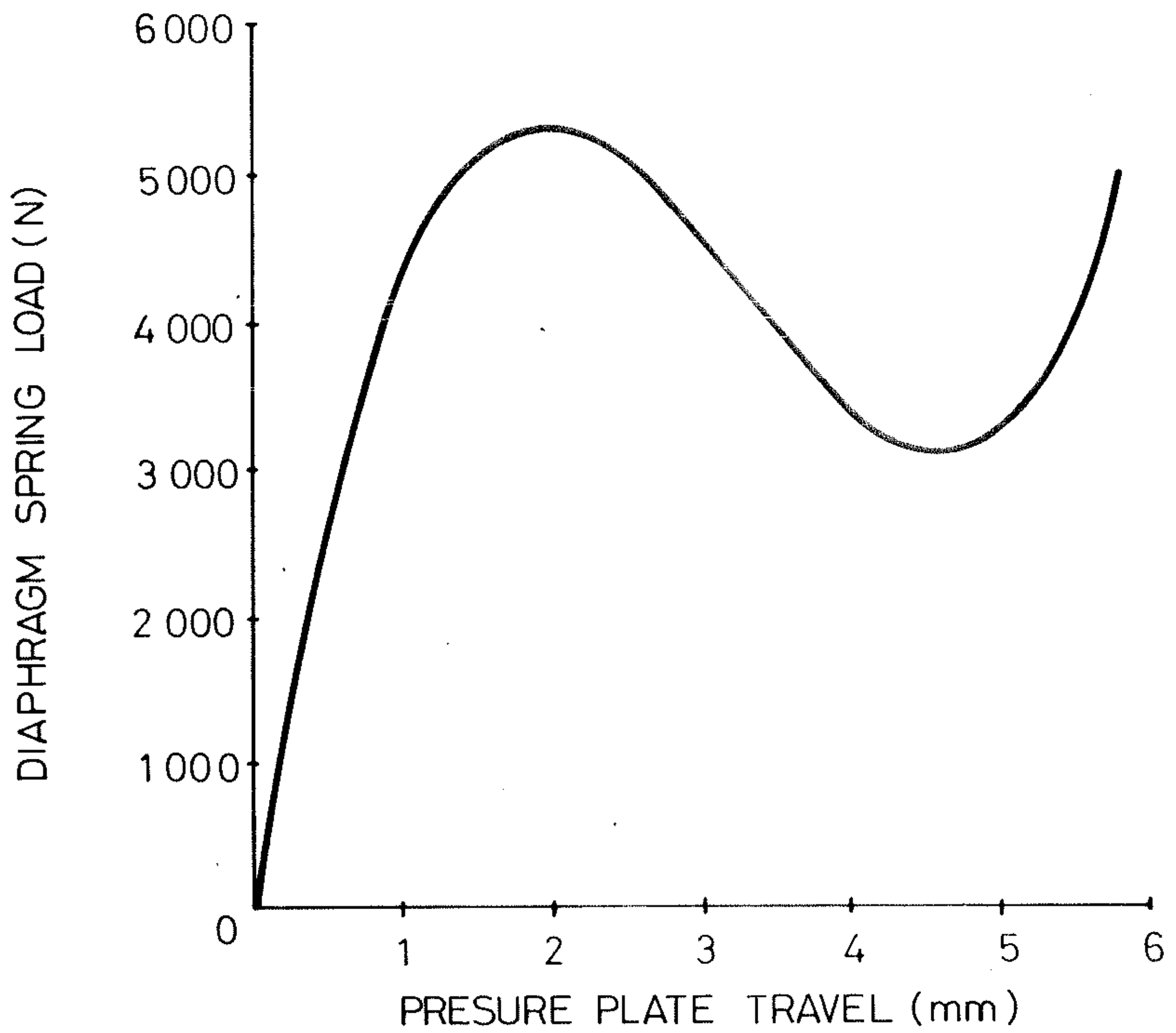

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CLUTCH DIAPHRAGM SPRING CHARACTERISTIC

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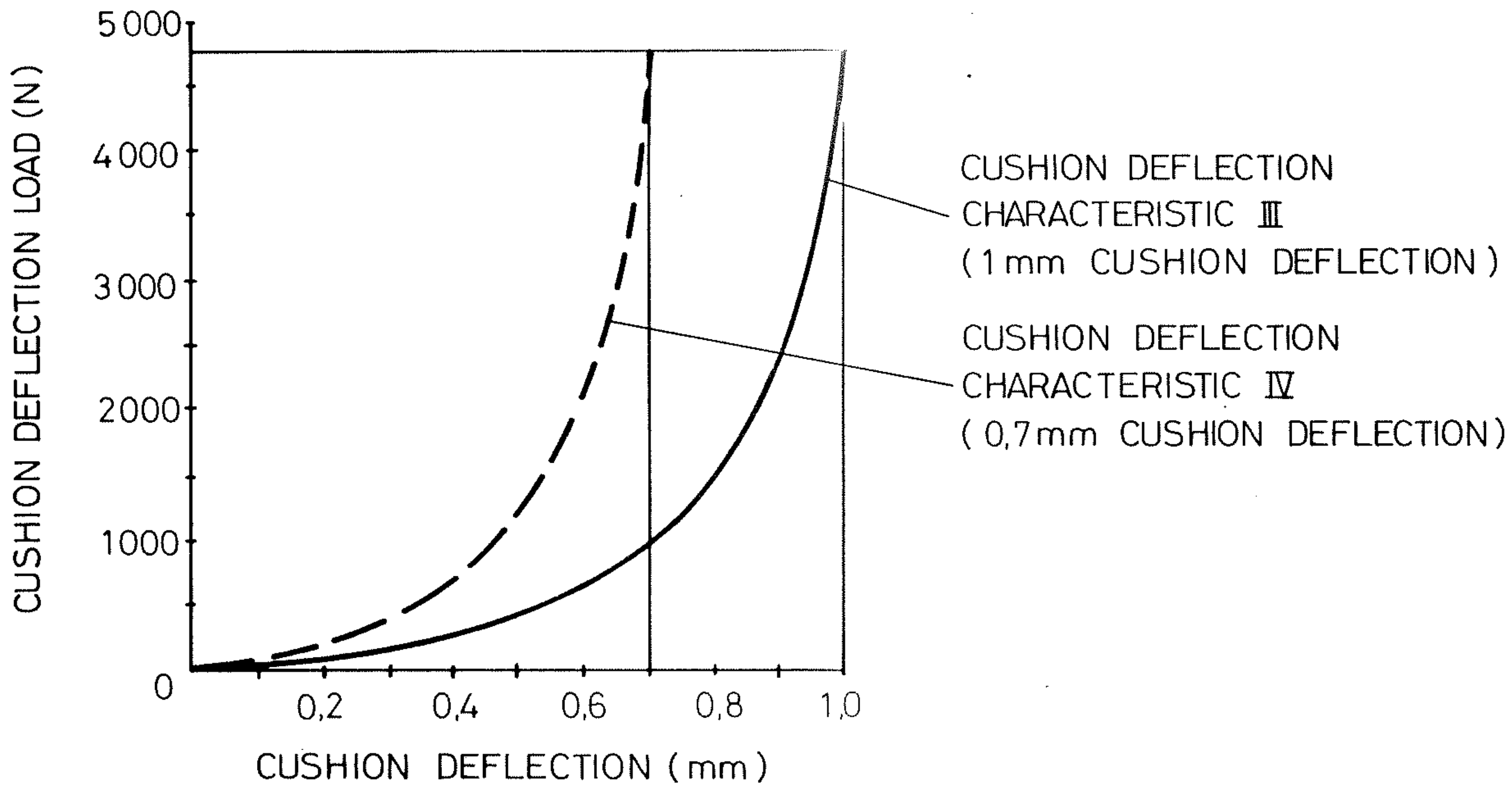

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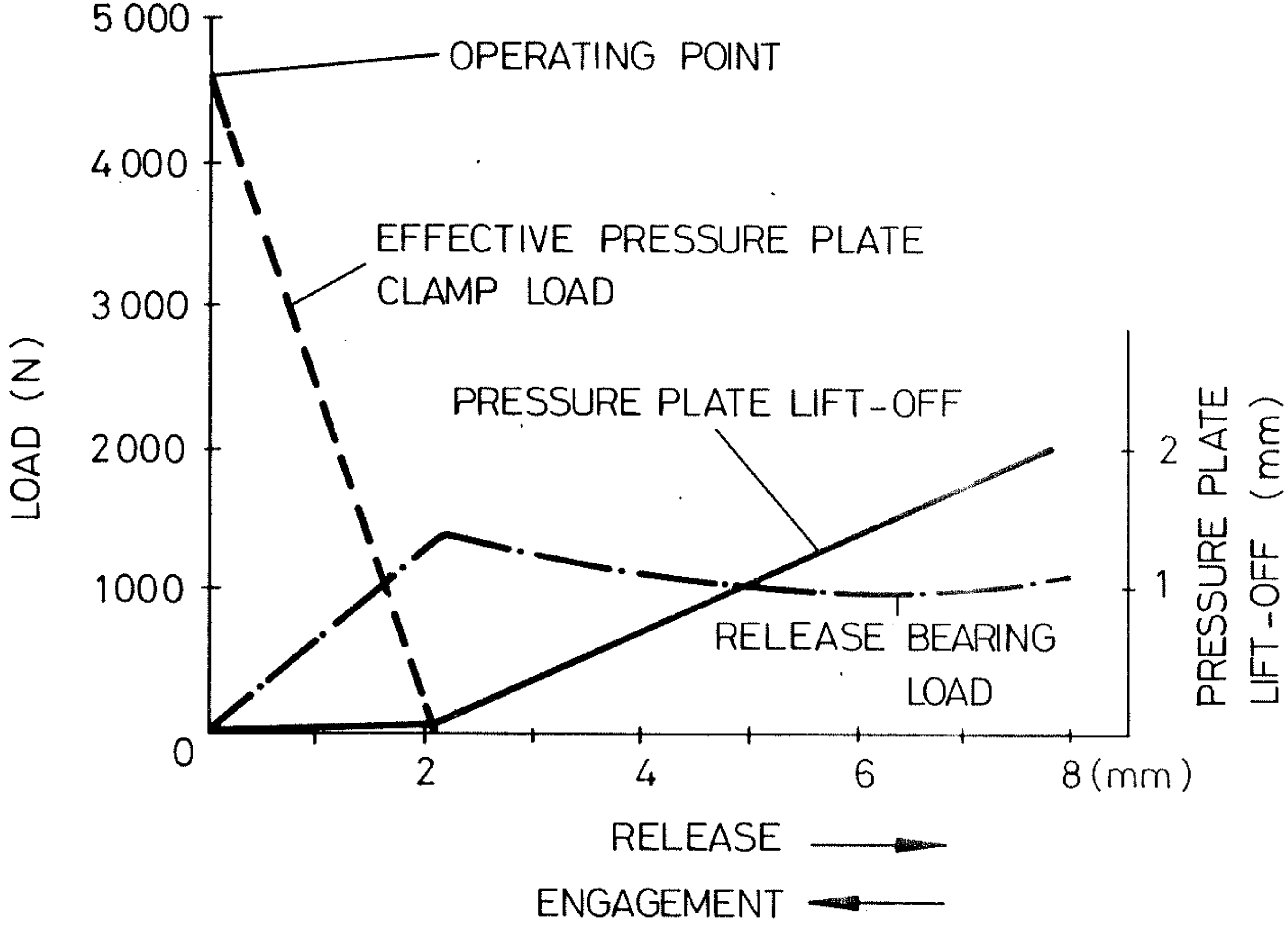
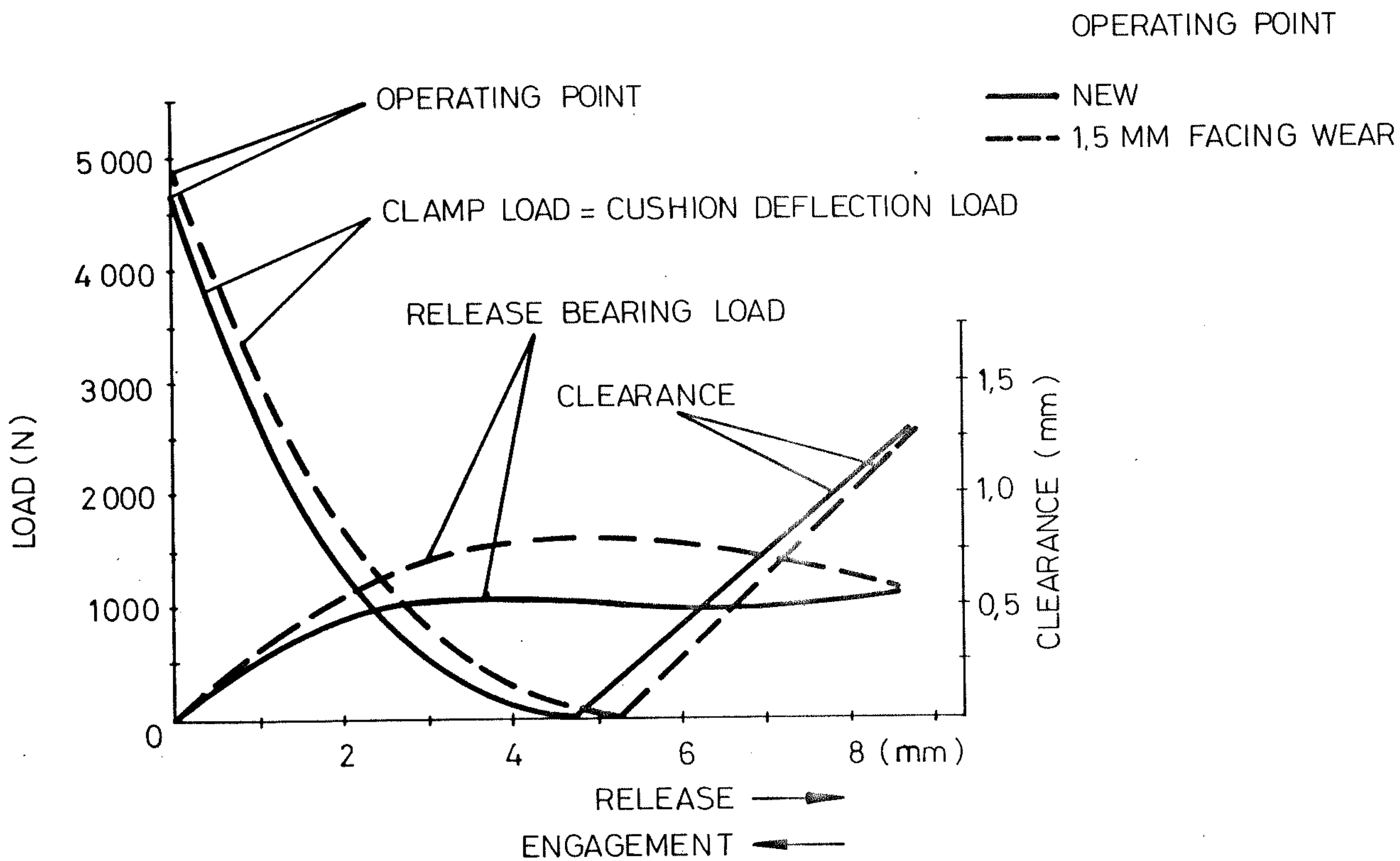
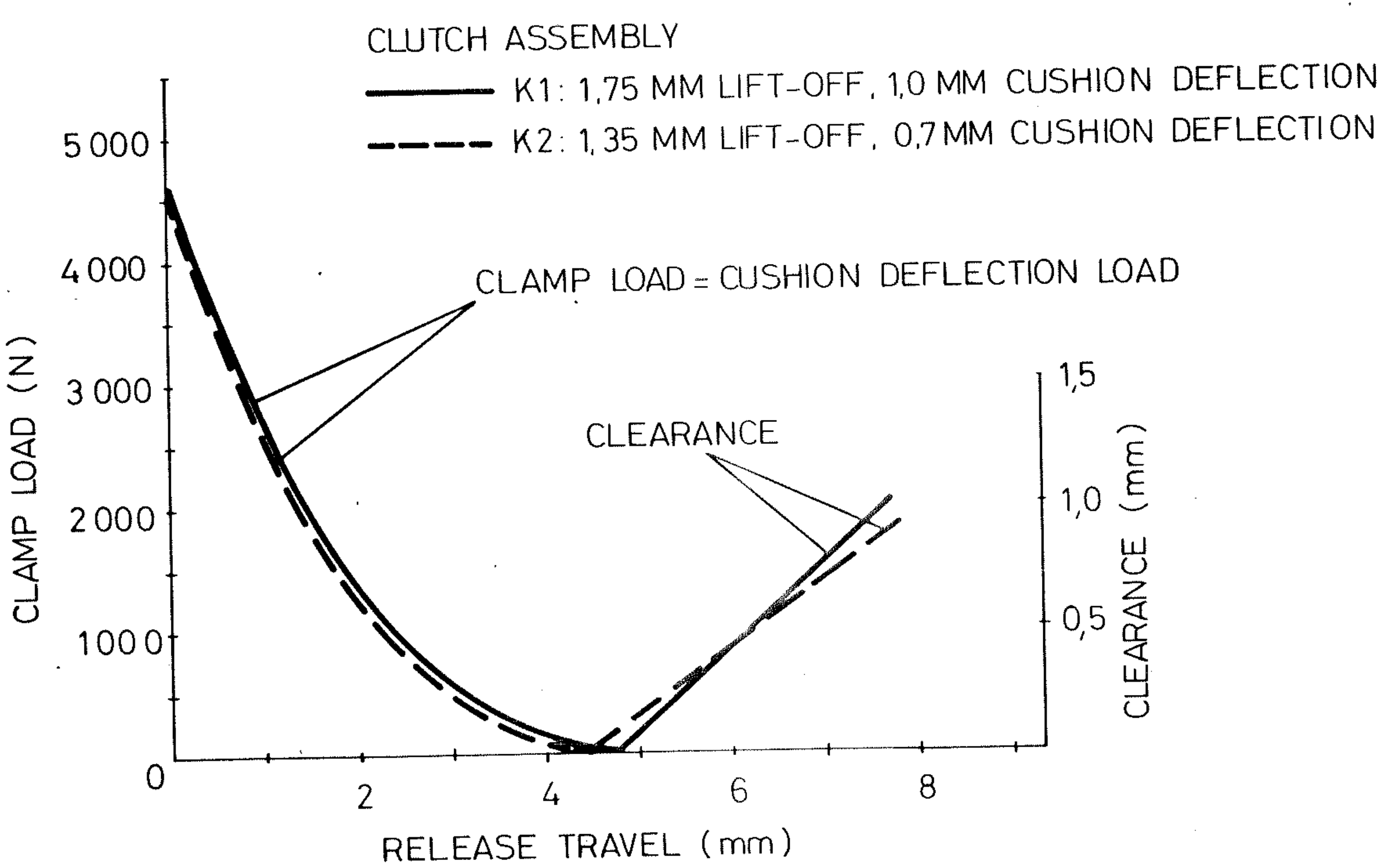
CUSHION DEFLECTION CHARACTERISTICS



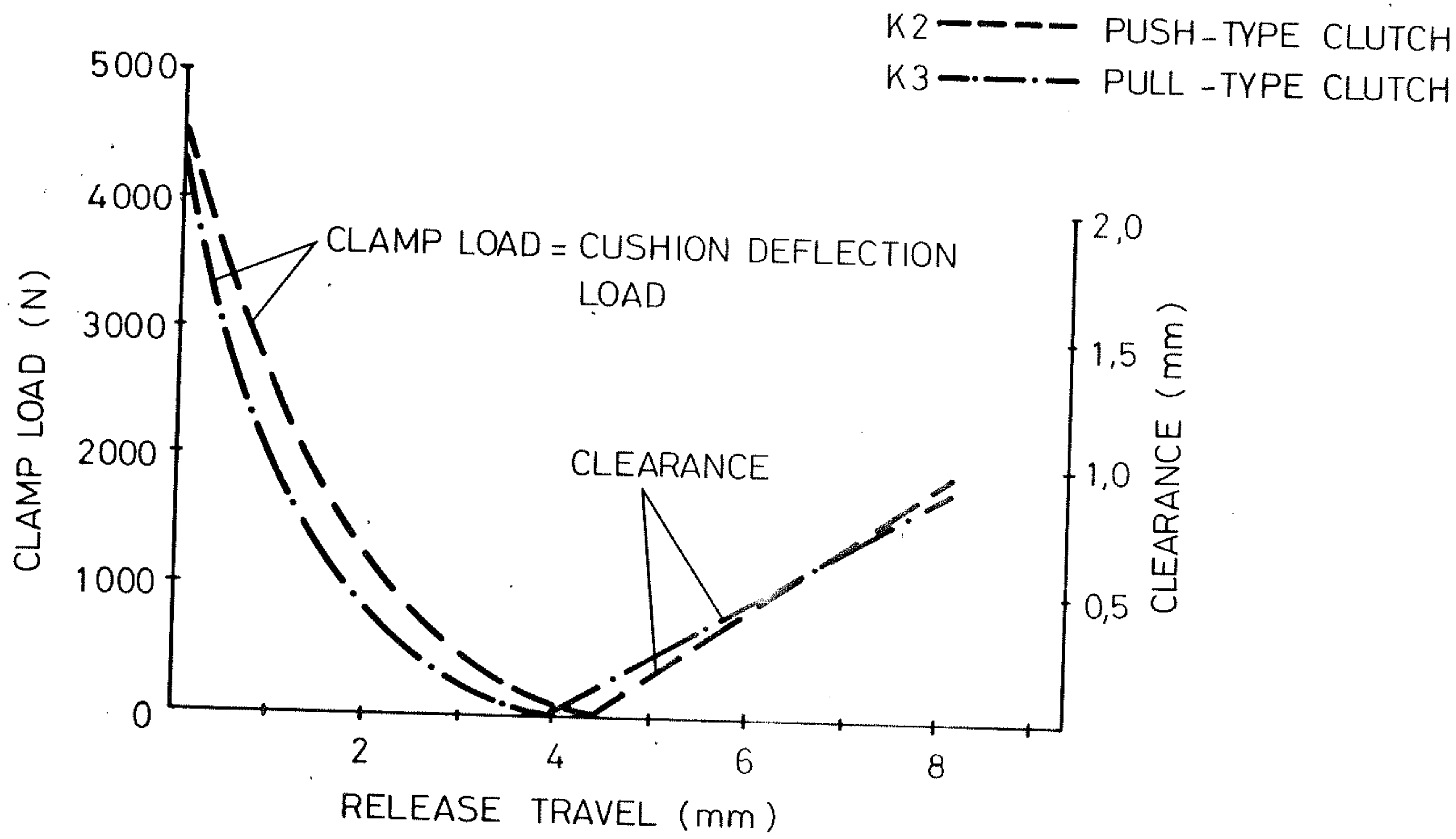
13 04 86	CLUTCH CHARACTERISTIC CURVES WITHOUT THE INFLUENCE OF THE CUSHION DEFLECTION



14 04 86	CLAMP LOAD, RELEASE BEARING LOAD AND CLEARANCE FOR CLUTCH K1 (WITH CUSHION DEFLECTION CHARACTERISTIC III)



15 04 86	CLAMP LOAD AND CLEARANCE FOR K1 AND K2	LUK



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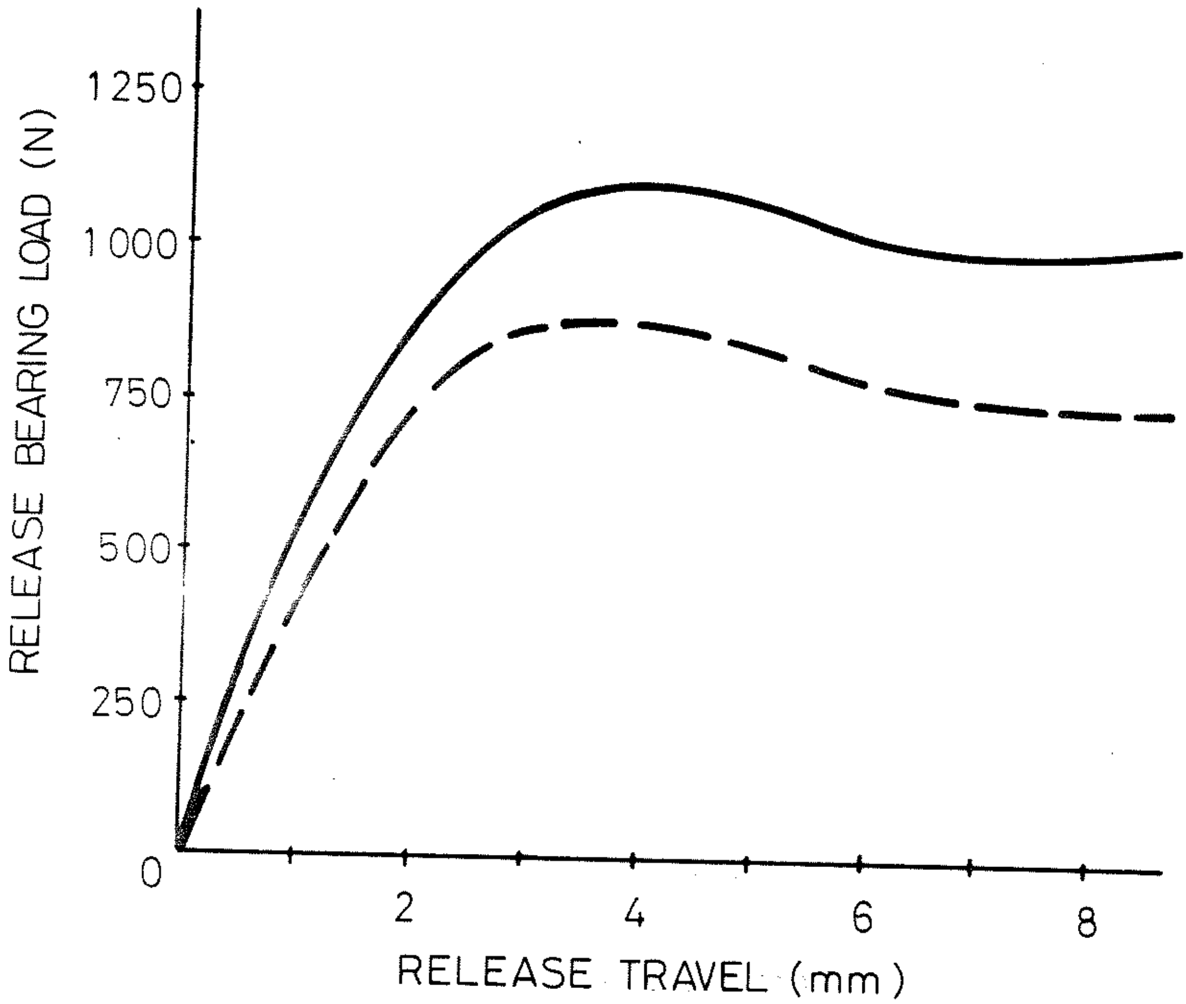
CLAMP LOAD AND CLEARANCE
FOR PUSH-TYPE CLUTCH K2 AND PULL-TYPE CLUTCH K3

LUK

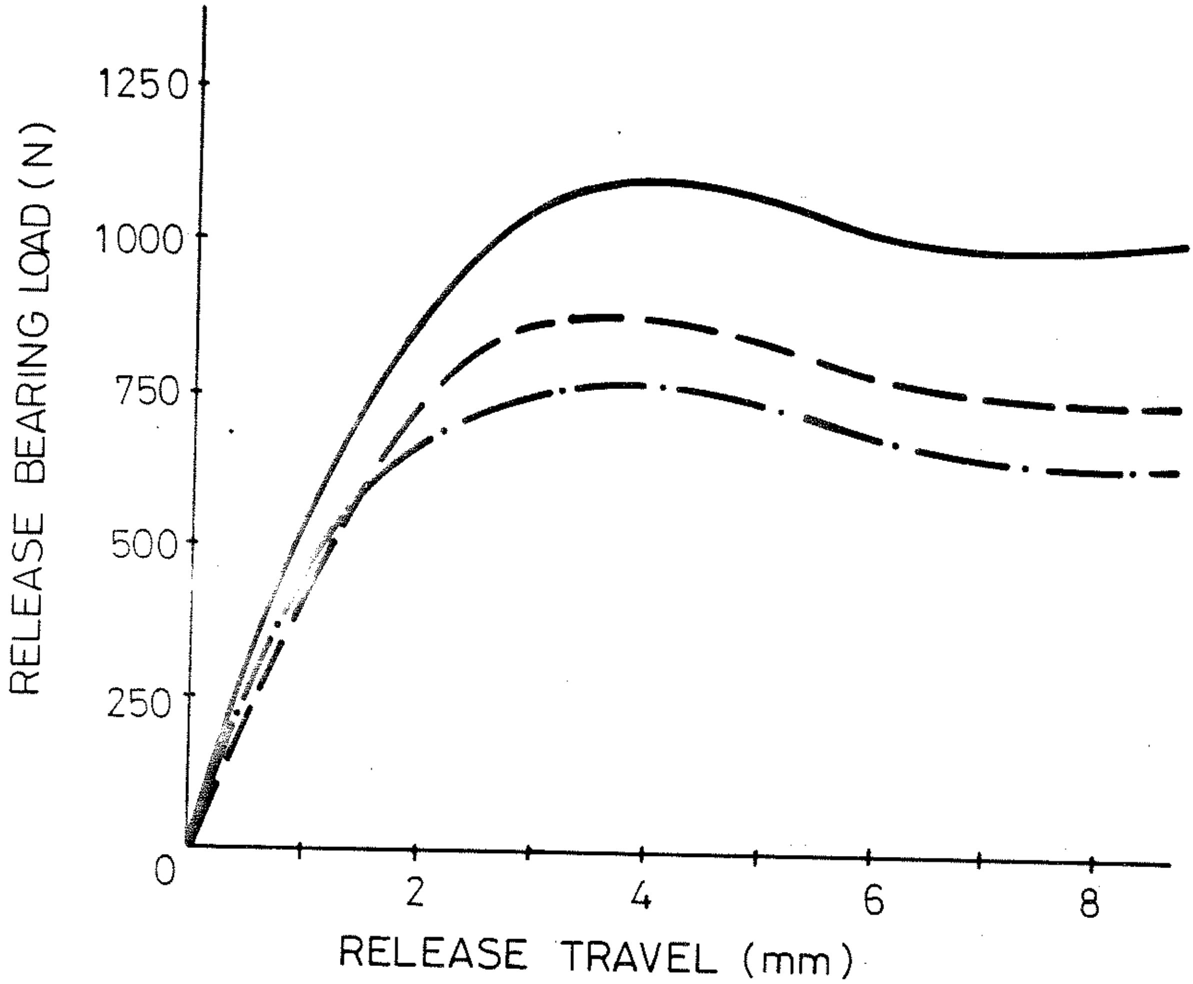


CLUTCH ASSEMBLY

K1 : PUSH - TYPE CLUTCH , 1,75 MM LIFT-OFF

K2 : PUSH - TYPE CLUTCH, 1,35 MM LIFT-OFF

16 04 86	COMPARISON OF RELEASE BEARING LOAD FOR CLUTCHES K1 AND K2	LUK



CLUTCH ASSEMBLY

K1 : PUSH-TYPE CLUTCH, 1,75 MM LIFT-OFF

K2 : PUSH-TYPE CLUTCH, 1,35 MM LIFT-OFF

K3 : PULL-TYPE CLUTCH, 1,35 MM LIFT-OFF

18 04 86	RELEASE BEARING LOAD COMPARISON FOR K1, K2, K3	LUK

- SPECIFIC HEAT STRESS:

  START-UP ON LEVEL GROUND:

  LIMIT DEPENDENT ON ENGINE DISPLACEMENT:
  22 Nm / CM^2 – 40 Nm / CM^2

  START-UP ON 26% GRADE:

  MAX. 300 Nm / CM^2

- CLUTCH SAFETY FACTOR: MIN. 1,2

- SLOPE OF DIAPHRAGM SPRING CURVE: $F_{MAX.} : F_{MIN.} = 1 : 0{,}6$

- CLUTCH LIFT-OFF: ABOUT 1,2 MM