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TORSIONAL VIBRATIONS AND TRANSMISSION NOISE

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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.

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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

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- 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.

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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

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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 over-critical 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.

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

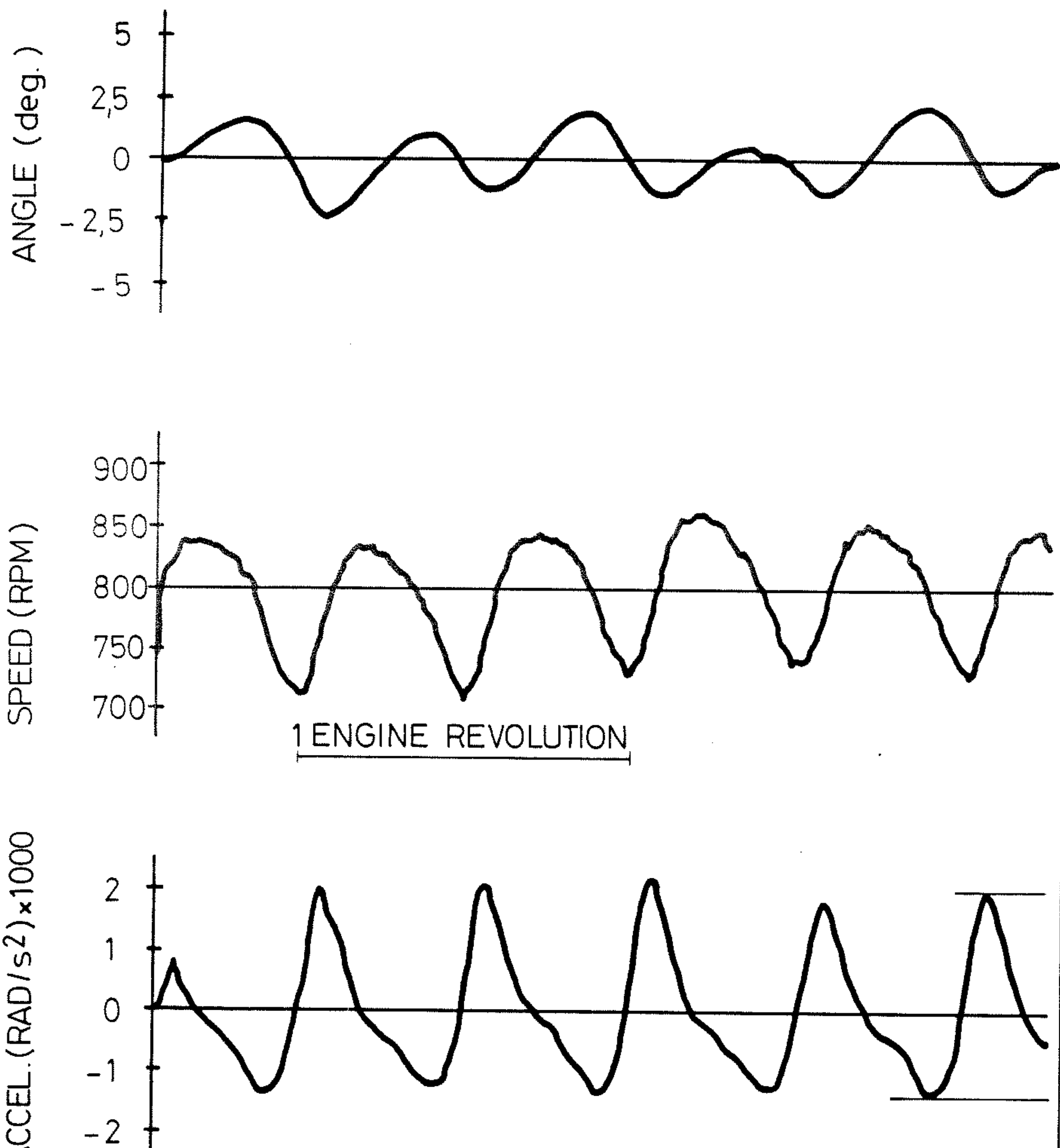
In neutral idle mode it is almost always possible to isolate the torsional vibrations from the transmission using an appropriate idle stage in the torsion damper. Because engine excitation, transmission design, moment of inertia, drag torque and noise insulation vary considerably, the idle characteristic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

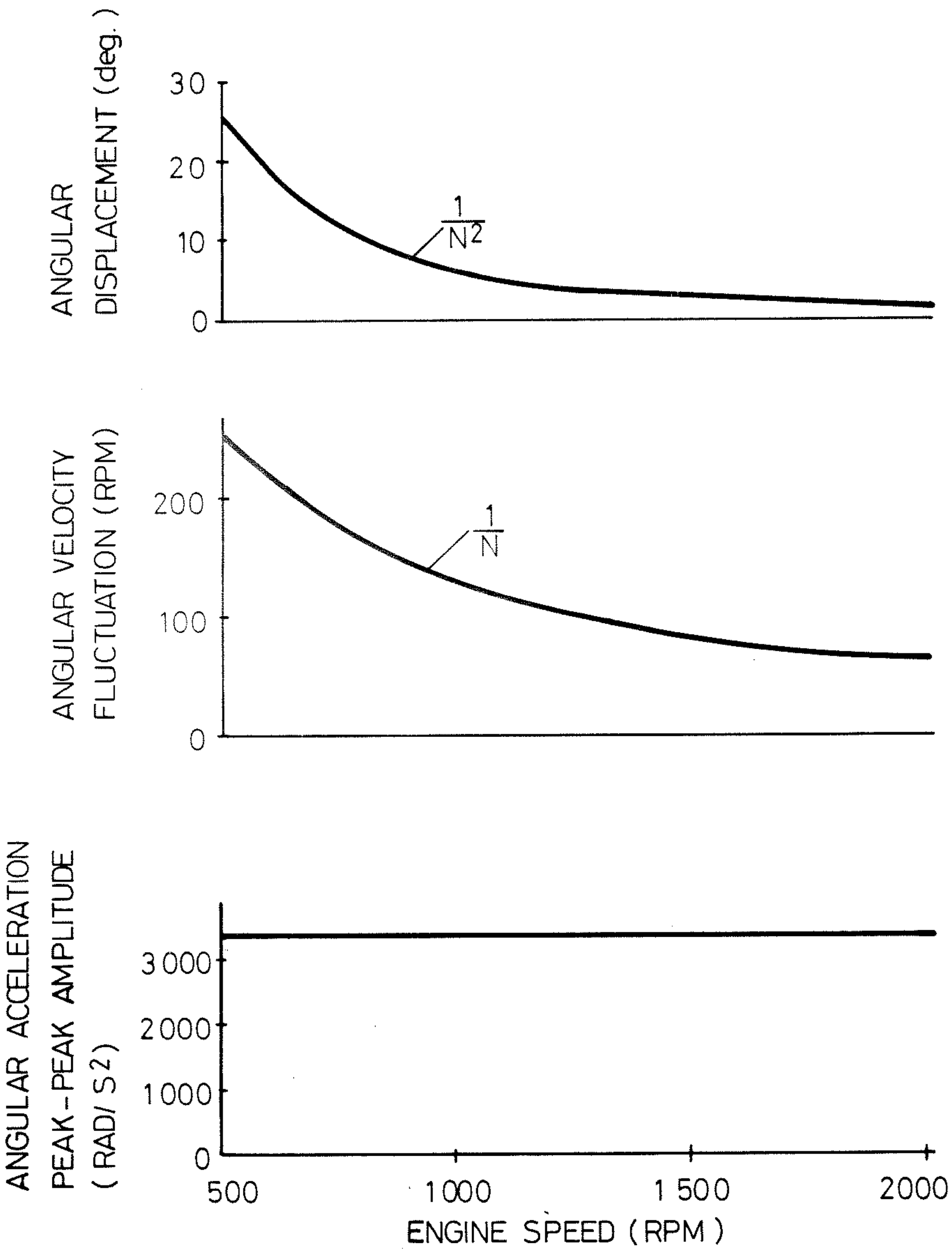


T = TORQUE

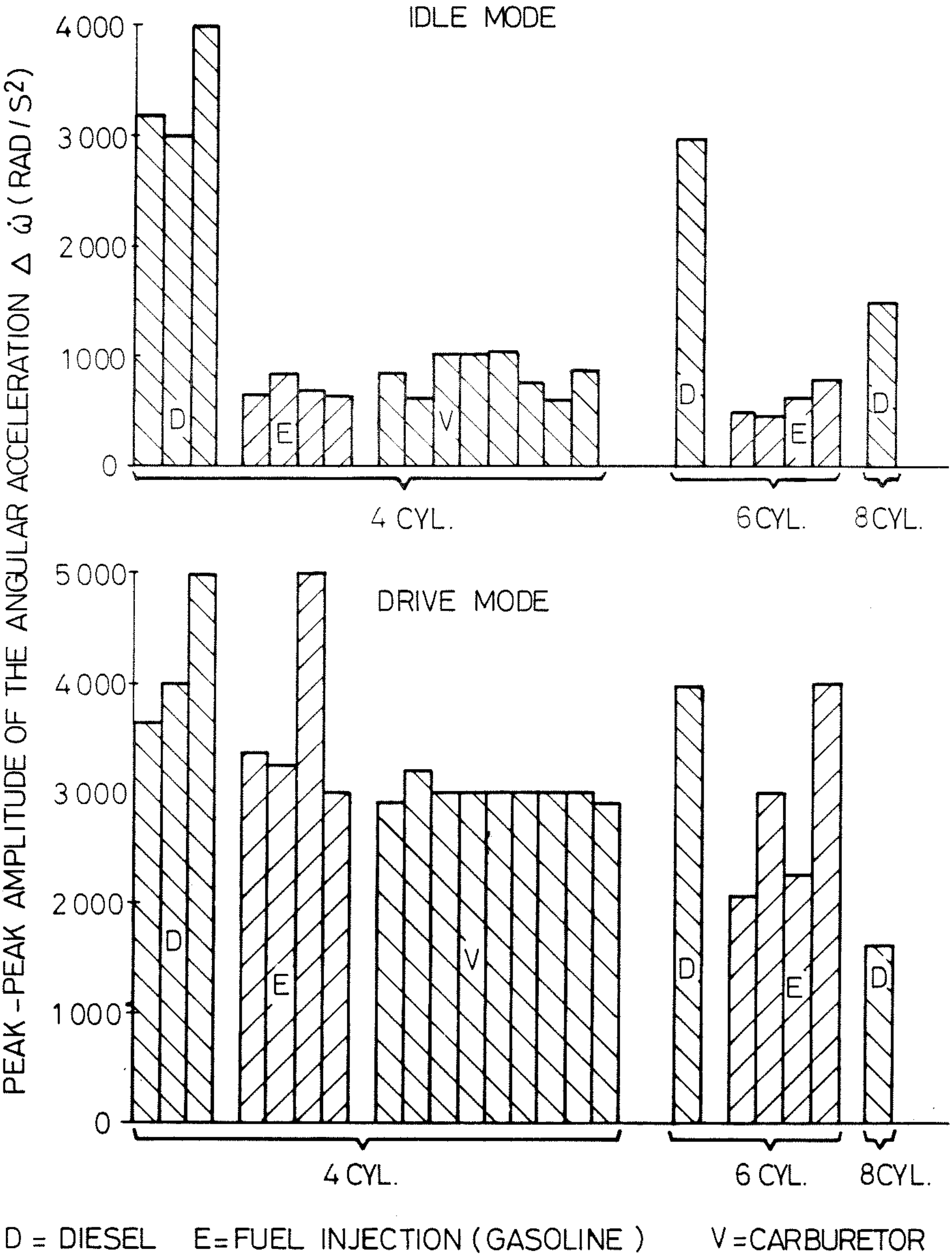
$T = J\dot{\omega}$ J = MOMENT OF INERTIA

$\dot{\omega}$ = ACCELERATION

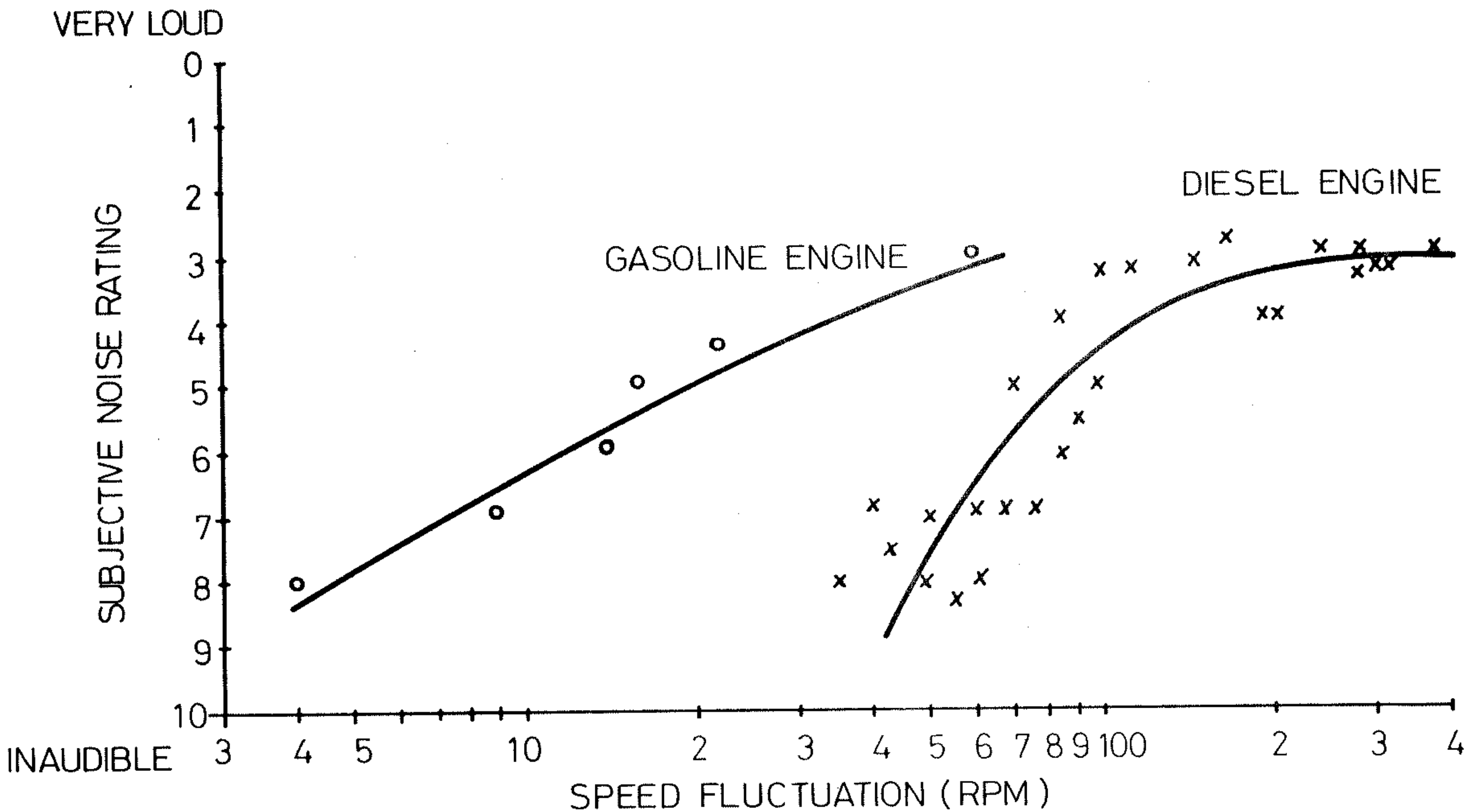
01 01 86	TORSIONAL VIBRATION OF THE CRANK SHAFT	LUK



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



03 01 86	TORSIONAL EXCITATION FOR VARIOUS ENGINES	LUK



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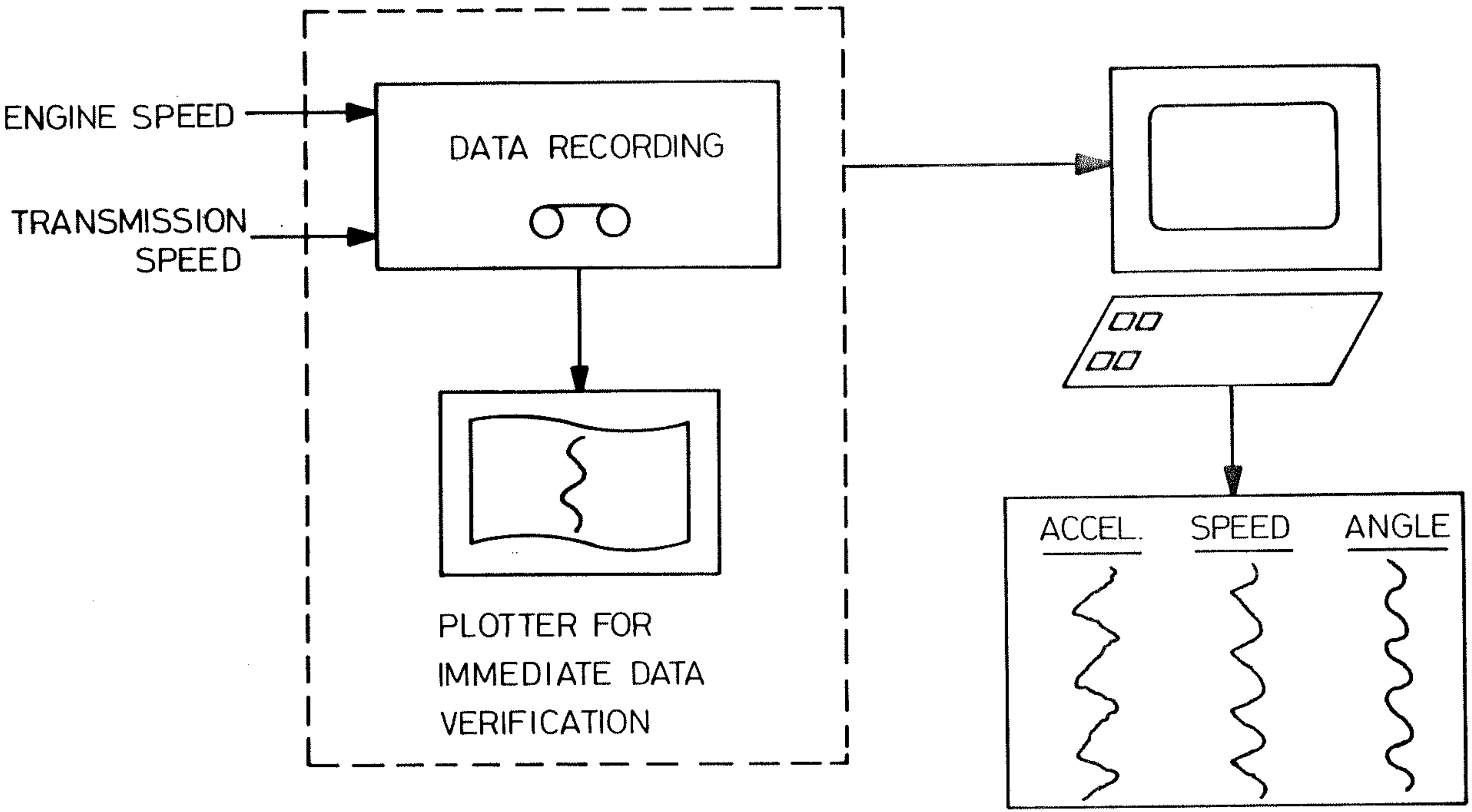
EFFECT OF SPEED FLUCTUATION ON THE NOISE RATING



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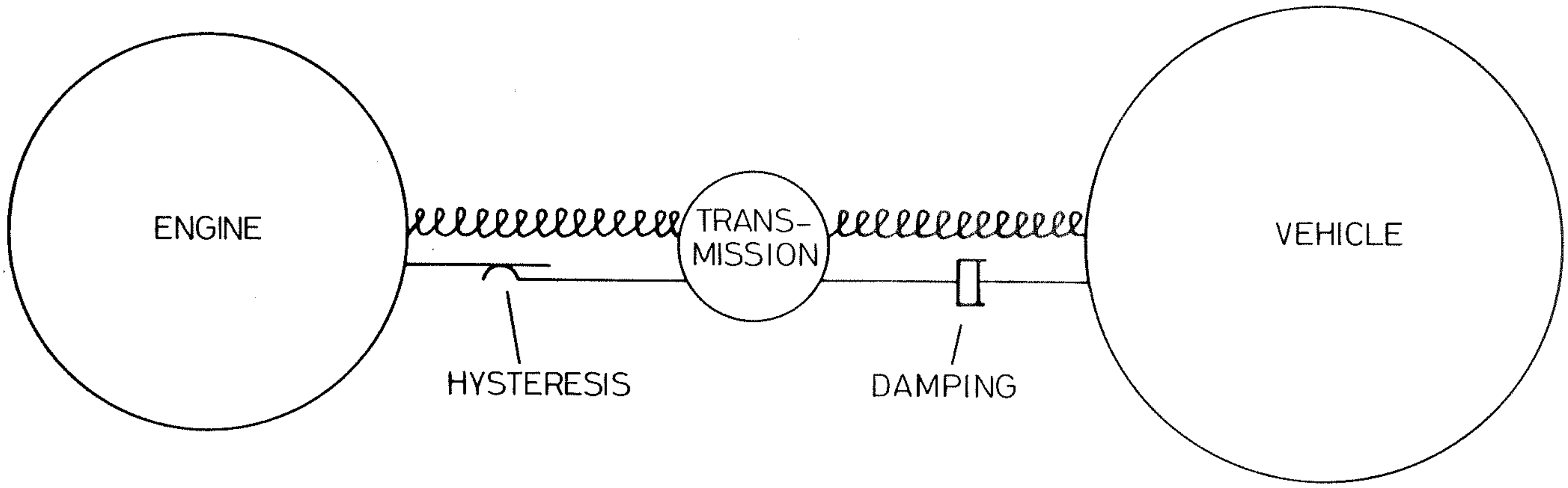
DATA RECORDING SYSTEM



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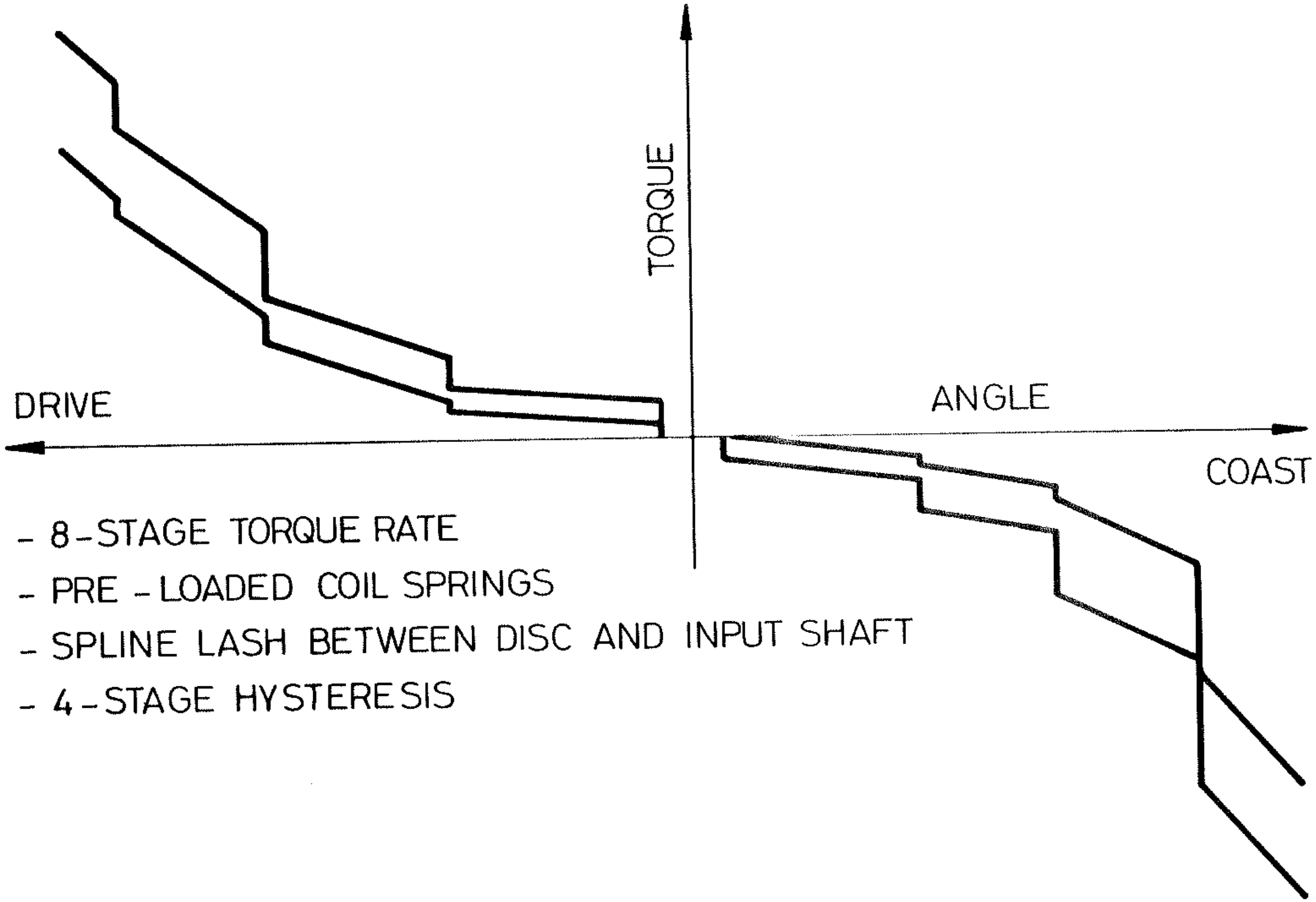
ANALYTICAL MODEL

LuK



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

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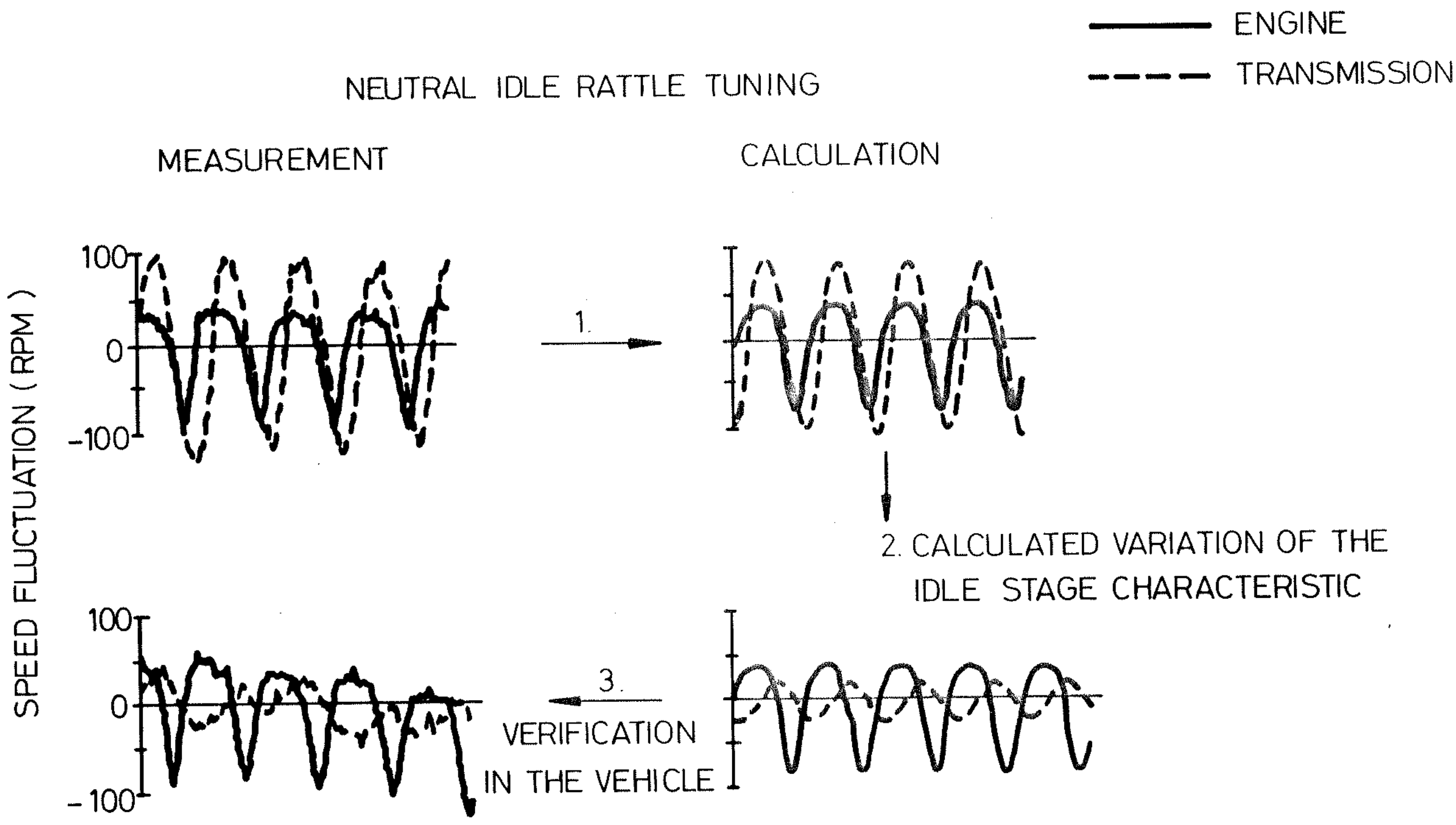
TORSION DAMPER CHARACTERISTIC CURVE



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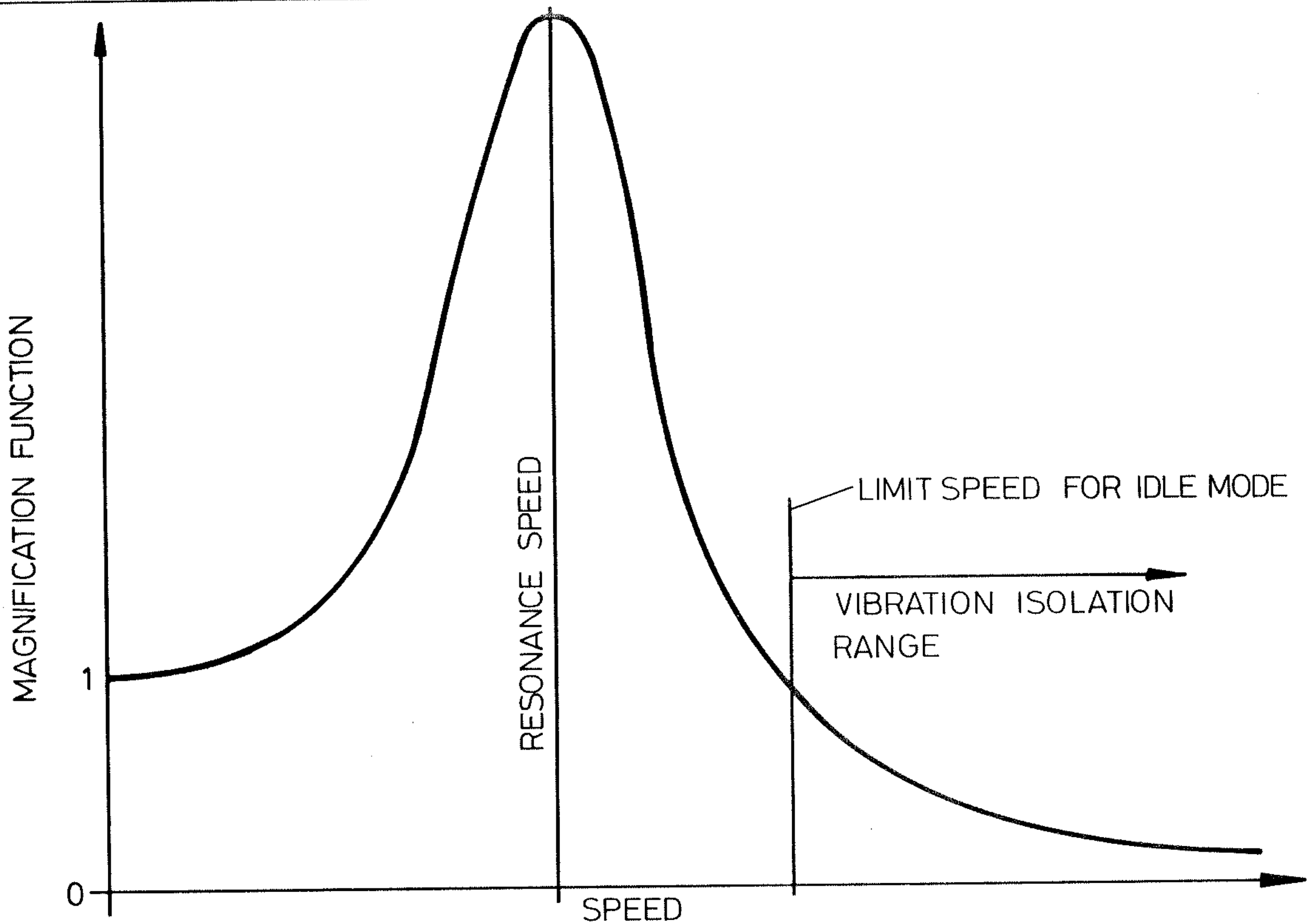
INTERACTION OF VIBRATION MEASUREMENT AND CALCULATION



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RESONANCE CURVE

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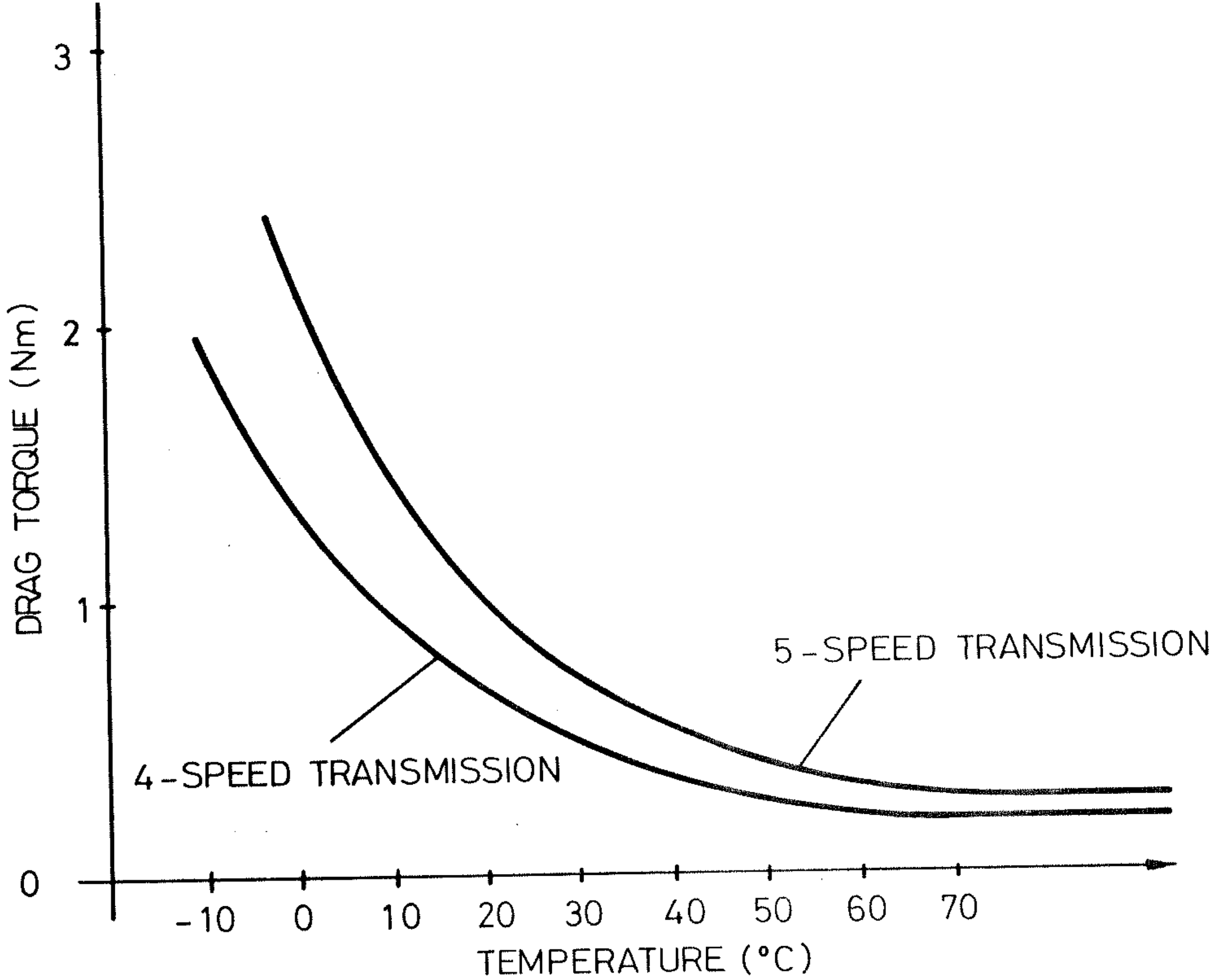

10 01 86	DRAG TORQUE - TEMPERATURE RELATIONSHIP

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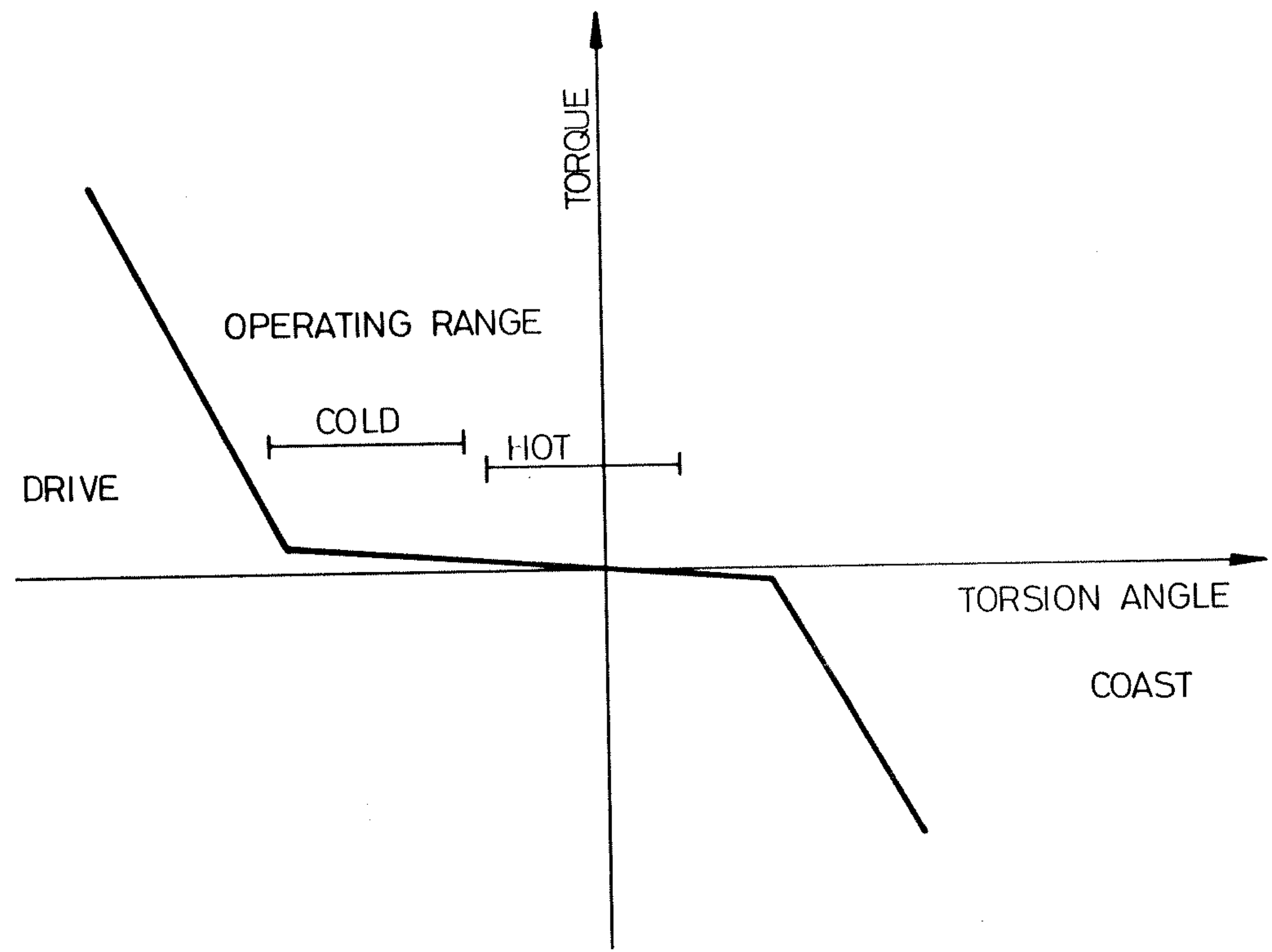
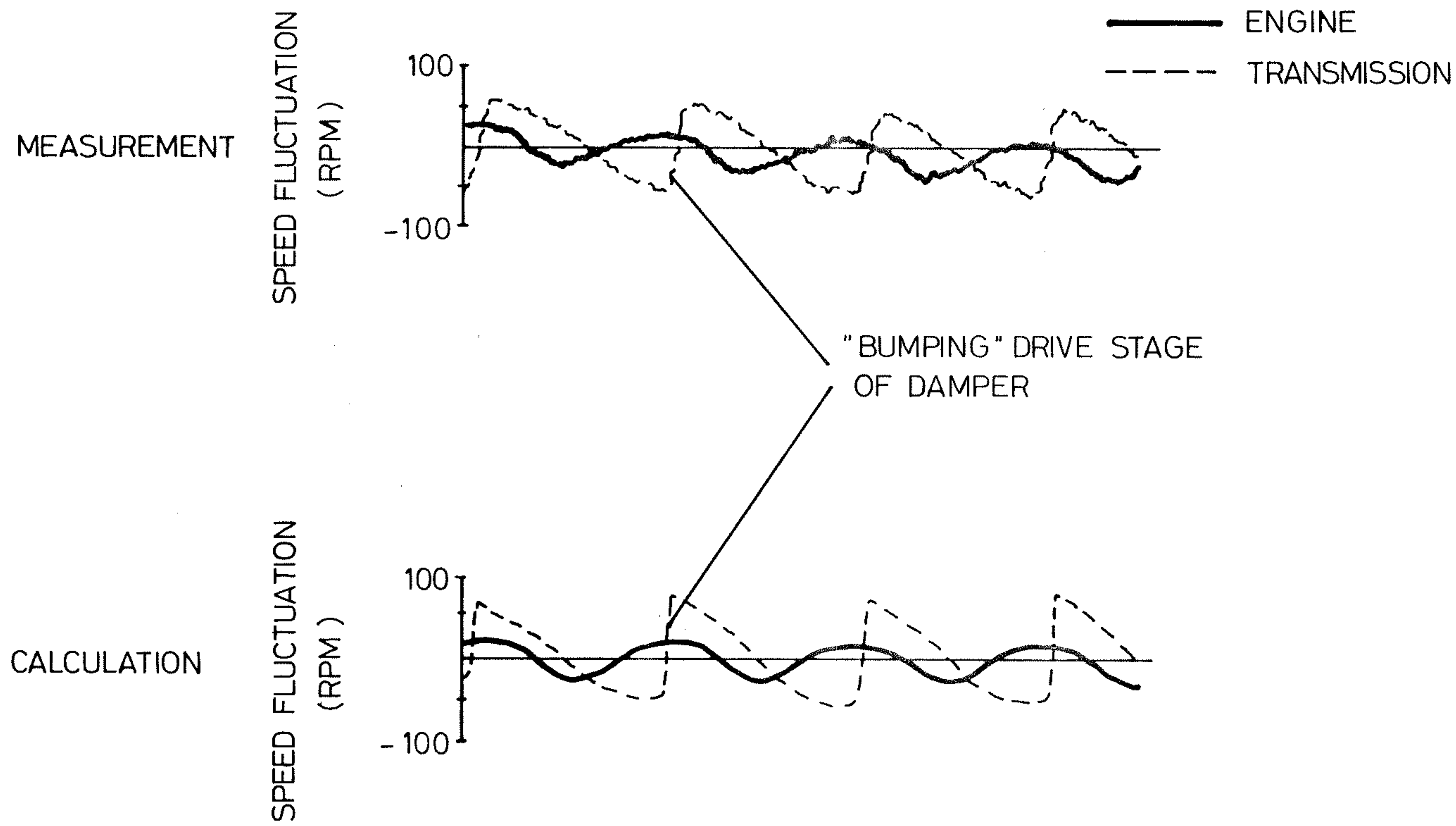


11 01 86	OPERATING RANGE OF TORSION DAMPER WITH HOT AND COLD TRANSMISSION FLUID	LUK



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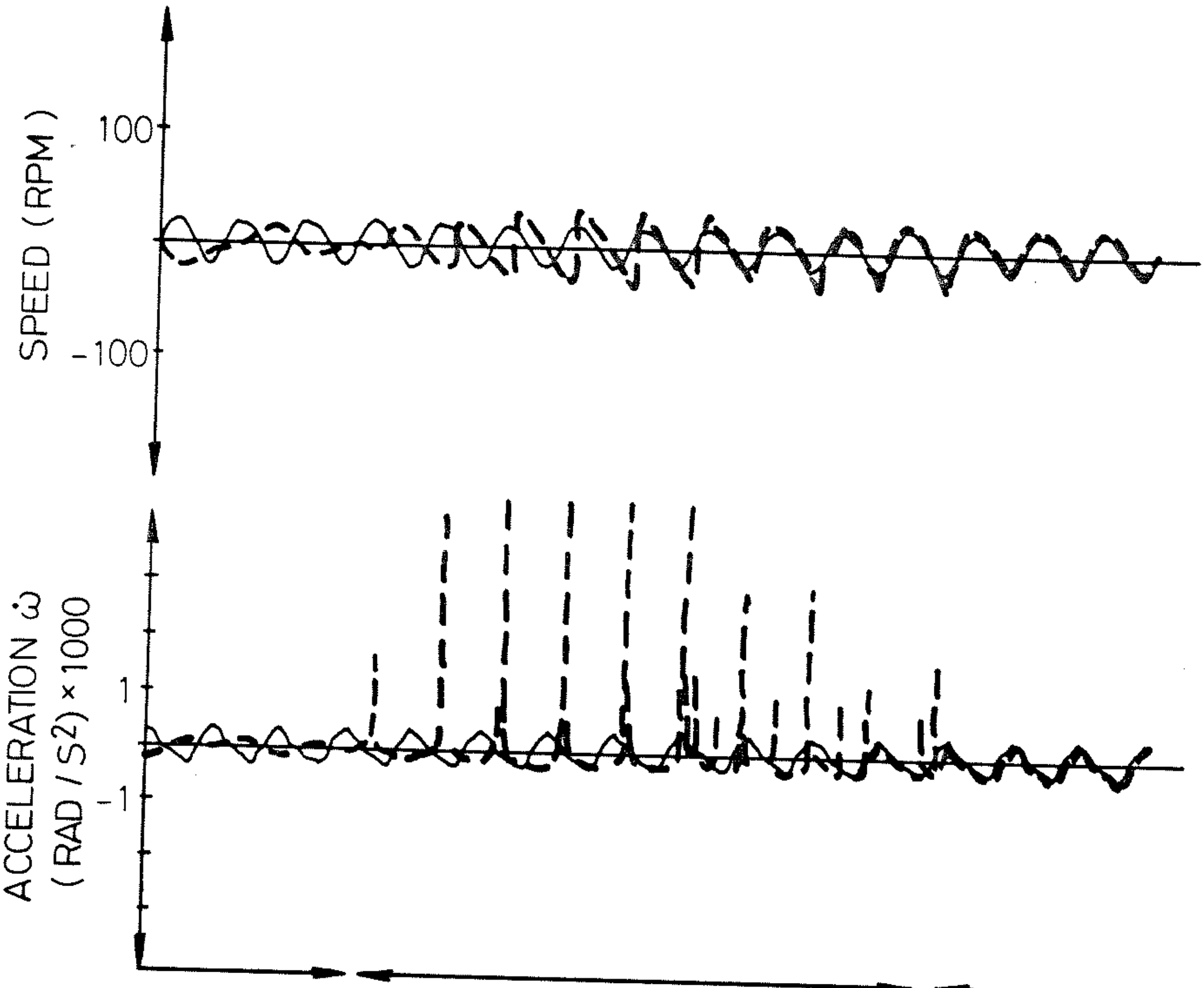
"BUMPING" DRIVE STAGE OF DAMPER

LuK

LOW DRAG TORQUE

HIGH DRAG TORQUE



TORSION DAMPER OPERATES IN THE IDLE STAGE

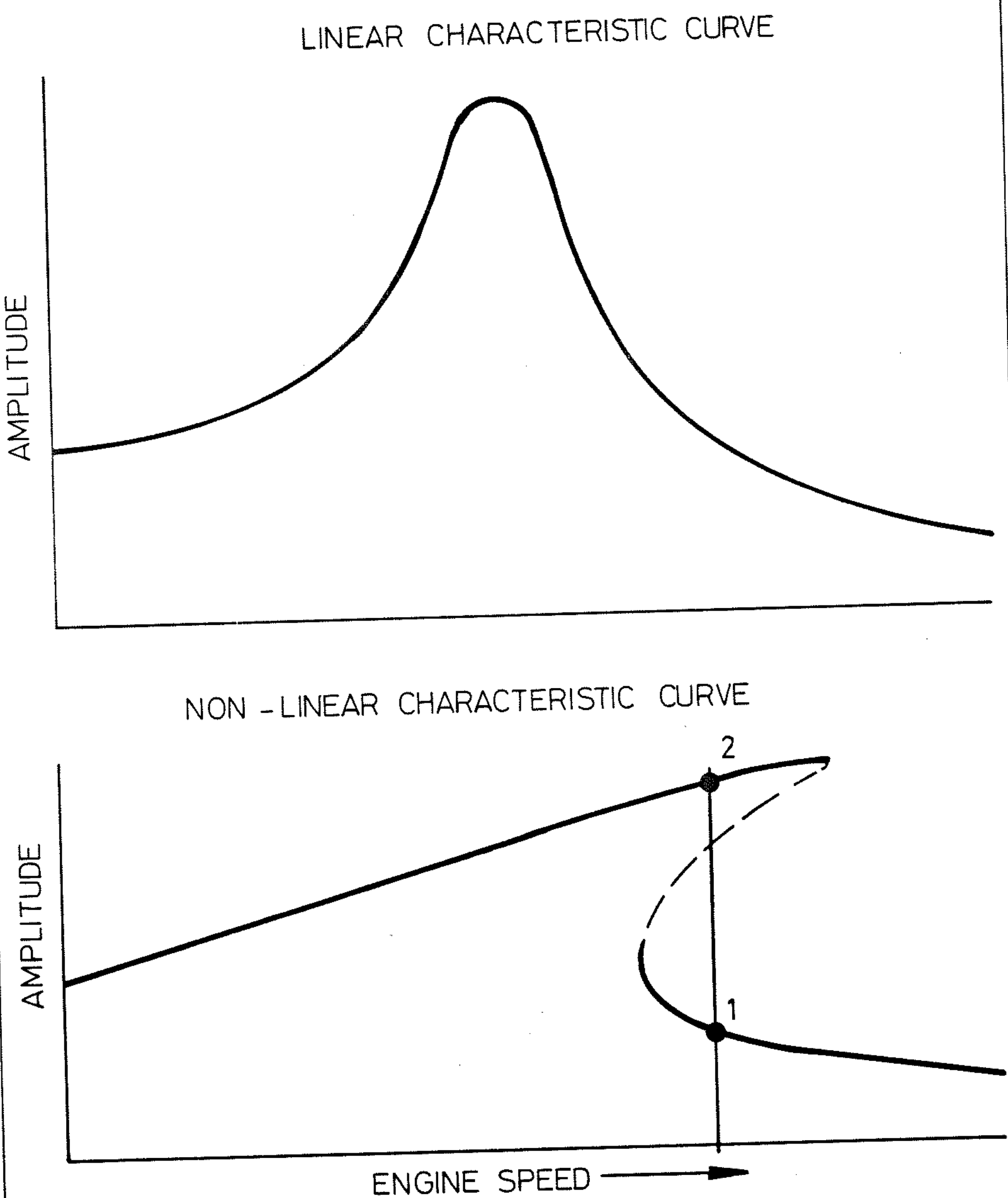
"BUMPING" ON THE DRIVE STAGE

TORSION DAMPER OPERATES IN DRIVE STAGE

ENGINE

TRANSMISSION

13 01 86	"BUMPING" ON THE DRIVE STAGE	LUK



14 01 86	RESONANCE CURVES FOR LINEAR AND NON-LINEAR CHARACTERISTICS

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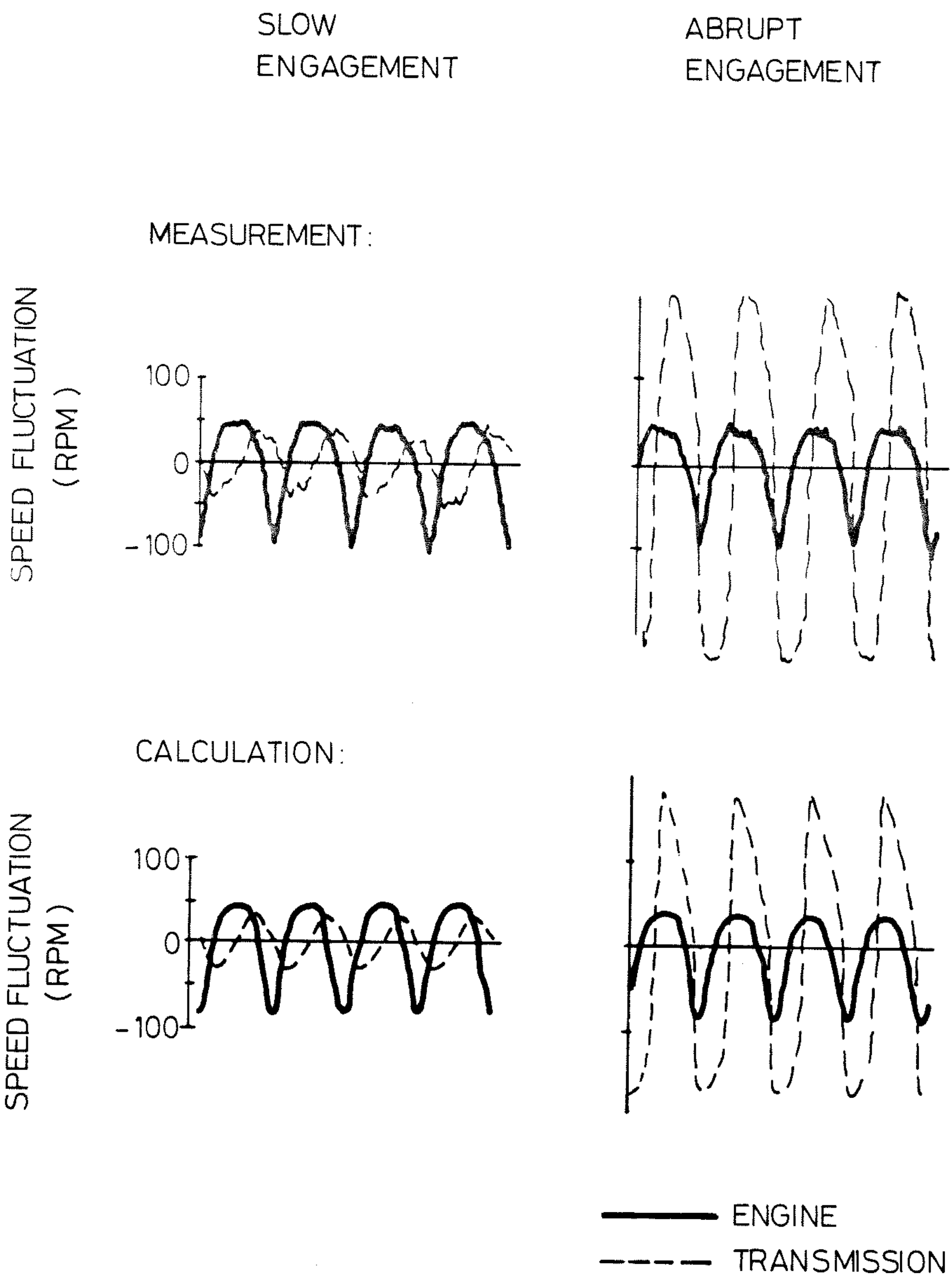

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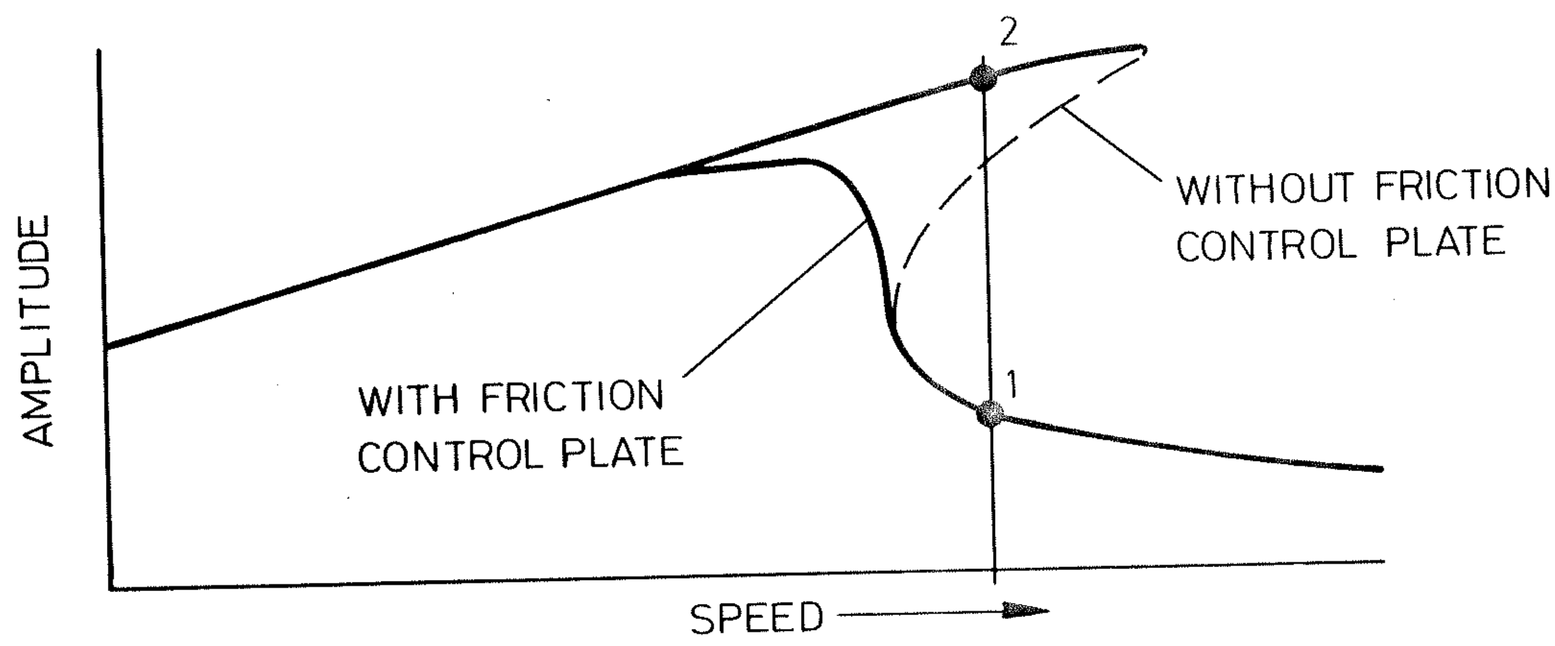
TORSIONAL VIBRATIONS FOR NON - LINEAR CHARACTERISTICS



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RESONANCE CURVES WITH AND WITHOUT FRICTION CONTROL PLATE

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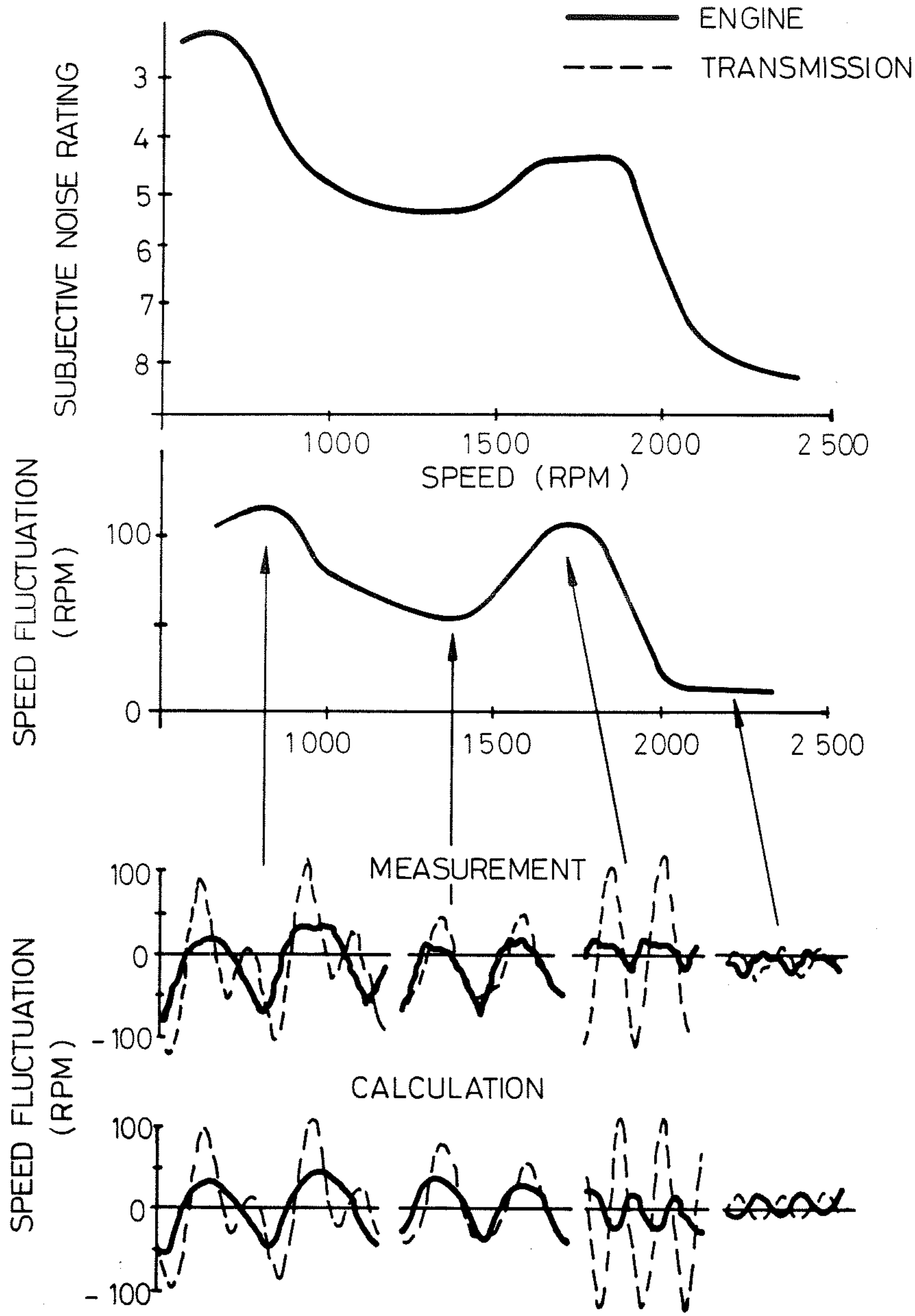
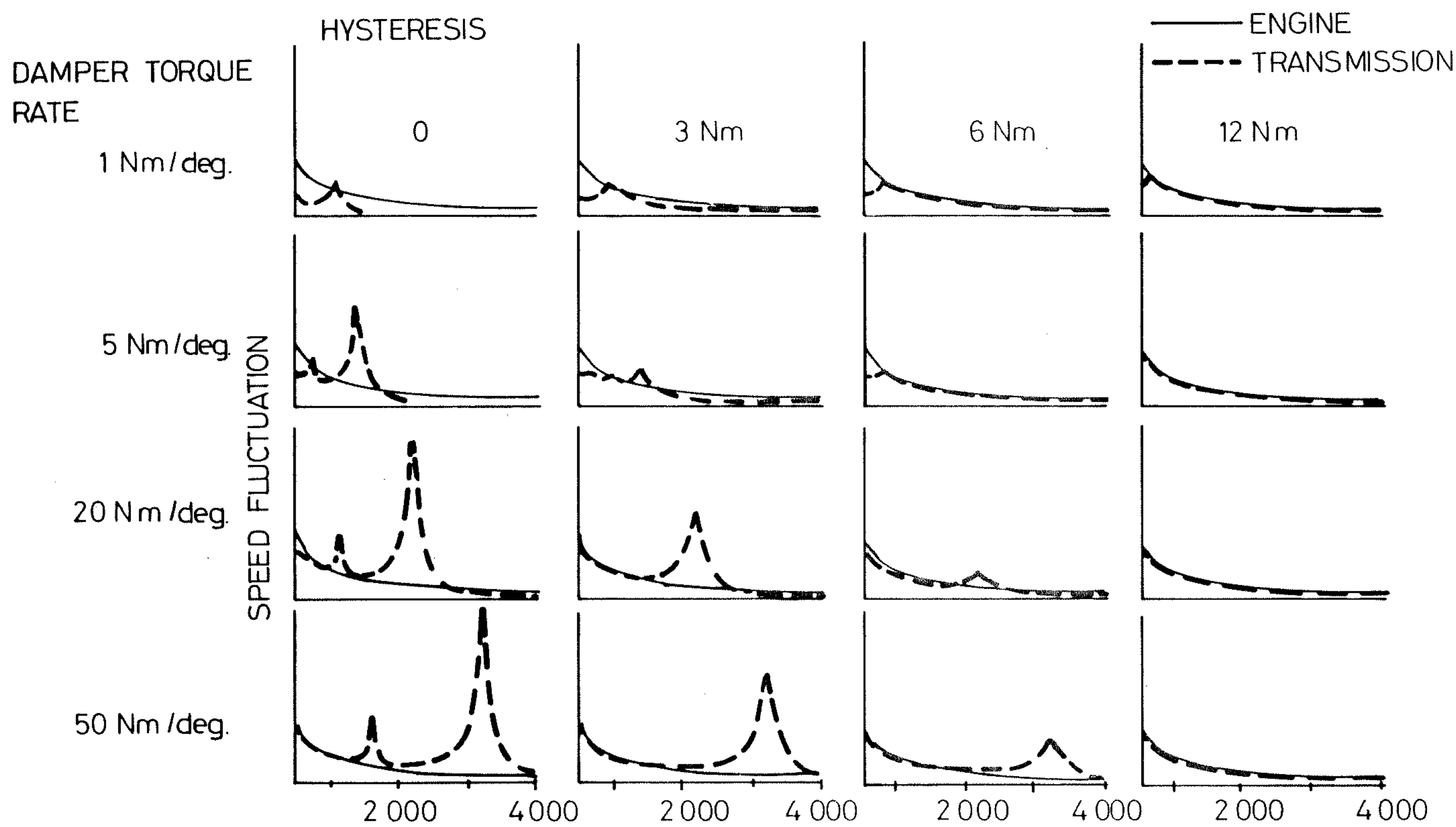


17 01 86	NOISE RATING AND VIBRATION AMPLITUDE IN DRIVE MODE	LUK



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SPEED FLUCTUATIONS IN RELATIONSHIP TO THE TORQUE RATE, THE HYSTERESIS, AND THE ENGINE SPEED

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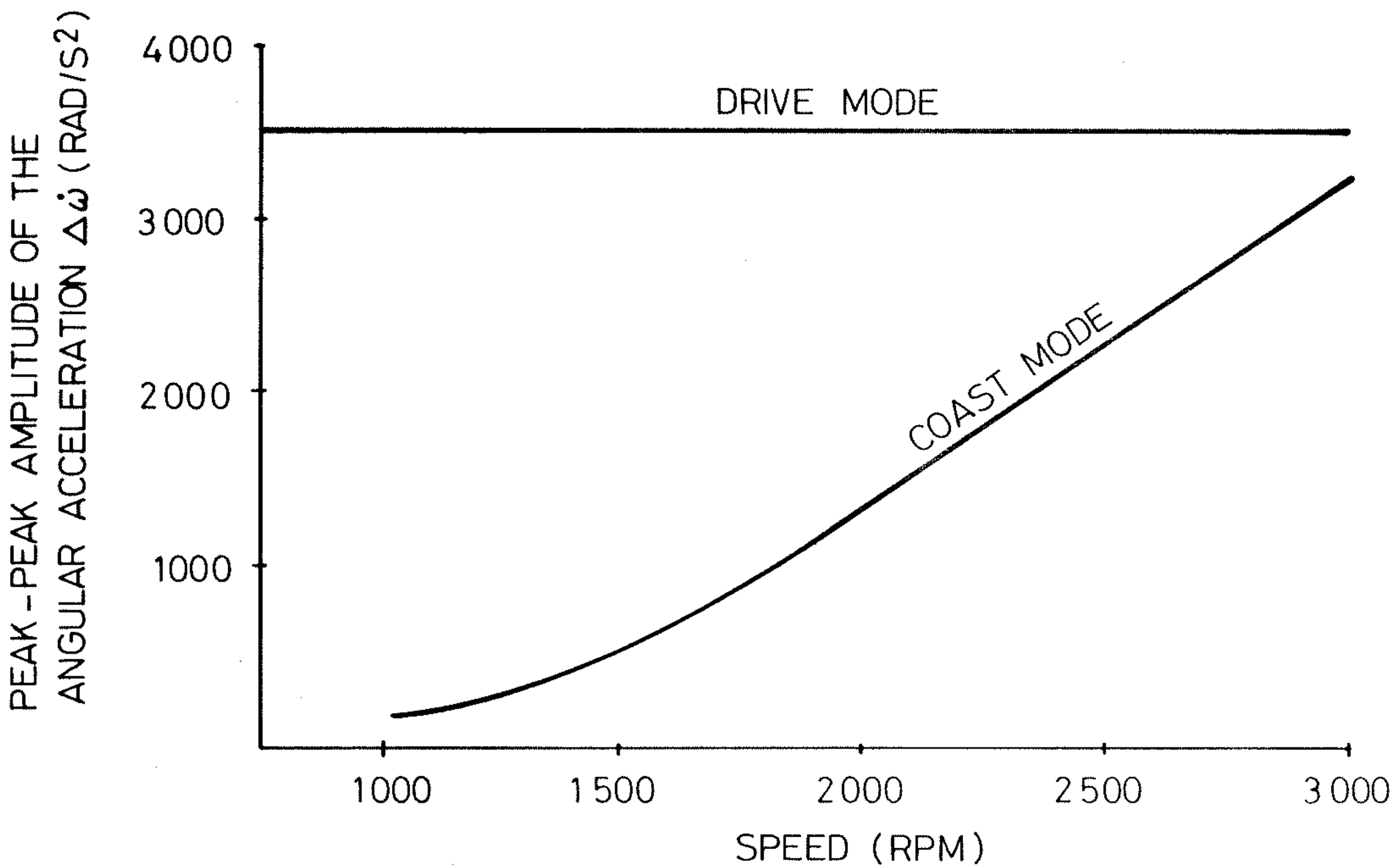

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COMPARISON OF ENGINE EXCITATION IN DRIVE AND COAST MODES

LUK