Torsional Vibrations in Tractor Drive Trains Damping Options

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Two thirds of LuK's current tractor clutch disc designs feature torsion dampers for master clutch operation, and in individual cases for PTO-drive as well. The general reason for using these torsion dampers is to eliminate annoying noises generated by torsional vibrations in the drive train.

The percentage of tractor clutch systems using torsion dampers has continued to increase over the last few years. As is the case with other types of vehicles, this is attributable to overall enhancement of total vehicle quality. In addition to improving mechanical efficiency by reducing internal friction, there can be further reasons in the specific case of tractor transmissions:

- closer ratios between the different gear stages, resulting in a greater number of components that are capable of vibrating and generating noise.

- greater overdrive transmission ratios, which proportionately magnify the irregularities of the internal combustion engine that cause vibrations.

- power-shift transmissions using the torsion damper as the connective element with the engine make, for various design reasons, special demands of the torsion damper.

LuK has kept pace with these customer demands and has over recent years invested increasing amounts of development work in torsion dampers for their tractor line.

The results of this work, which contributes as much to a general understanding of torsional vibrations in tractor drive trains as it does to the individual design problem, will be the topic of the following presentation. This discussion makes a number of comparisons to passenger car tuning because the basic problems involved and the general approach to problem solving are similar.

The Engine as a Source of Torsional Vibration

Today's tractors usually use four-stroke diesel engine drive systems. Discontinuous fuel combustion results in irregular crankshaft torque, which fluctuates around an average value.

Figure 1 shows the torques affecting the crankshaft.

In comparison to the first presentation, the graph has been expanded to include tractor values. The peak-peak amplitude of the torque transmitted from the crankshaft to the flywheel is plotted as a function of cylinder piston displacement, both for idle mode and for operation under full load. The graph yields the following specific average values that can be used for estimating magnitudes:

- Idle mode: specific peak-peak amplitude of the torque approx. 1000 Nm/dm^3
- Full load: specific peak-peak amplitude of the torque approx. 1800 Nm/dm^3

Both passenger car and tractor diesel engines adhere to this basic pattern. Passenger cars have cylinder piston displacements in the range of 0.3 – 0.6 dm^3; tractors, on the other hand, usually use cylinder piston displacements of about 1 dm^3.

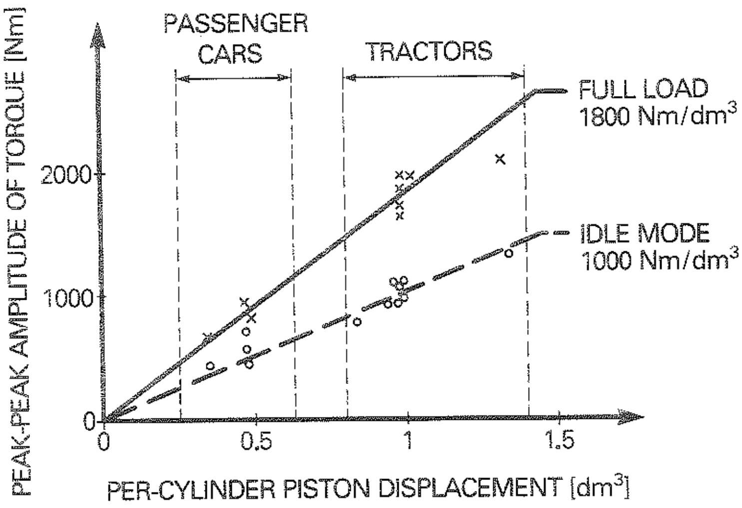

Figure 1: Torque amplitude for diesel engines

Depending on the size of the flywheel, the torque amplitude produces a specific angular acceleration amplitude and thus becomes the source of torsional vibrations. Based on this value, the number of cylinders in the engine, and the speed level, it is possible to calculate both the amplitude of the speed and the angle.

Figure 2 shows the speed amplitude for an idle speed of 700 rpm plotted as a function of the number of cylinders and the specific mass moment of inertia of the engine. In this case the specific inertia of the engine is to be understood as the total mass moment of inertia of the crankshaft drive + the flywheel + the clutch referenced to the piston displacement of the individual cylinder. The graph shows the standard ranges for tractors and passenger cars.

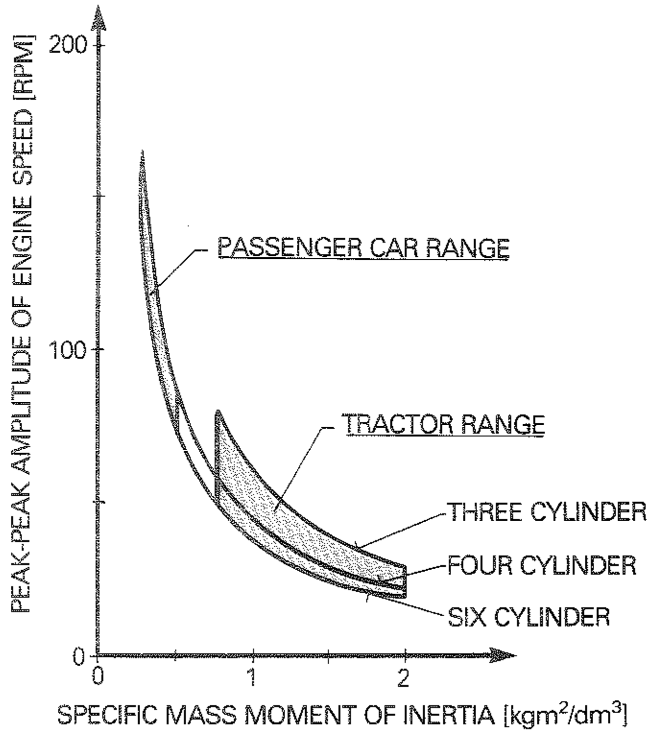

Figure 2:
Engine speed amplitude in idle at 700 rpm

In like manner, Figure 3 shows speed amplitudes for full engine load at 1000 rpm. It is important to note the following factors:

– The specific mass moment of inertia of the engine varies relatively drastically, as does the magnitude of the torsional vibrations generated in any individual case. The graph shows speed fluctuations for the tractor range and accounts for varying numbers of cylinders at full load. These values lie between 25 and 100 rpm, which represents a ratio of 1:4.

- Based on numerical values alone, tractor conditions are more advantageous than those for passenger cars. However, this advantage disappears immediately if a transmission features, for instance, a 2:1 overdrive transmission ratio, which is entirely possible. This means that the amplitude doubles at individual mating gears or synchronization elements.

- Under full load, the amplitude for passenger car engines averages about 3 times the magnitude of tractor engine amplitudes. However, the amplitude is lower than one would expect from a simple continuation of the curves for the tractor range. This could be attributable to the fact that passenger car diesel engines usually use indirect fuel injection.

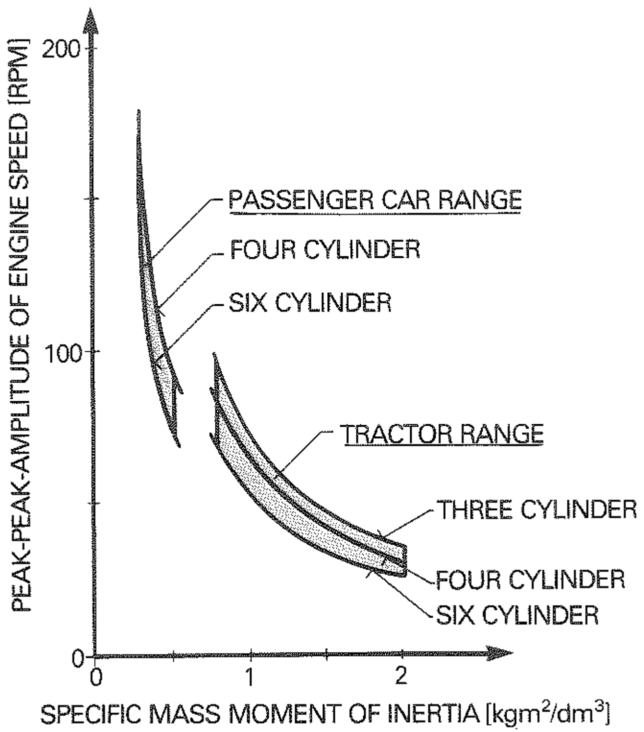

Figure 3:
Engine speed amplitude under full load at 1000 rpm

The Drive Train as a Torsional Oscillator

What do tractor transmissions look like? Depending on desired vehicle speeds ranging from approximately 1 to 25 mph, these transmissions must reduce nominal engine speeds of, for instance, 2,400 rpm at ratios of 1:20 to 1:400, working with drive wheels approximately 1.75 m in diameter. The highest transmission ratio of 1:400 requires a sequence of at least 4 reduction stages. To cover the ratio spread of 1:20 in steps of 1.2, approximately 16 gear sets are required.

Transmissions of this type are approximately 1 – 2 m long. Transmission input torques for 30 to 200 hp engines average between 100 and 800 Nm.

The number of reduction stages and gears allow numerous design options, which we have in production and which make it impossible to generalize tractor transmissions.

The following discussion will deal with torsional vibration problems and concrete numerical values for the simplest possible type of transmission. Figure 4 illustrates this kind of transmission in schematic form. The illustration shows the 4 reduction stages, specifically:

- speed section
- range section
- differential
- final drive ratio

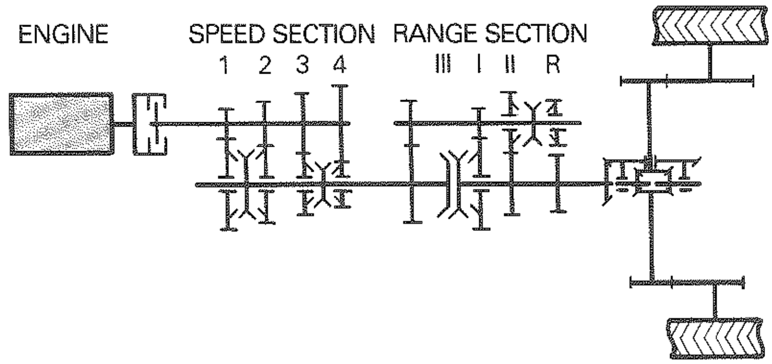

Figure 4: Block diagram for a simple tractor transmission (12/4)

There are four transmission gear reduction ratios in the first stage and another four in the second stage, including one reverse gear. This yields twelve forward and four reverse gears. For clarity's sake, not all components are represented here, specifically the PTO, the front wheel drive train, the splitter and the creeper gear used to increase the gear number.

Engine vibration is transmitted to these transmission components in an as-yet unknown way. It is unchanged, damped or magnified. If it causes noises, usually reducing vibration will either diminish or eliminate the noises. It is possible to introduce a torsion damper between the engine and the transmission in order to influence vibration behavior. Let us examine this influence, based on the standard distinction between idle and drive mode.

Vibration Behavior in Idle Mode

The presence of several shiftable transmission sections in a tractor transmission means that – unlike in a passenger car – there are several neutral positions. The example we are dealing with here produces 5. When the first transmission section is in neutral position, only the front 4 gear pairs revolve. However, when the driver shifts into any one of gears 1 – 4 while the second transmission section is in neutral, a series of additional components will turn at significantly different speeds, depending on the chosen gear. Consequently, the different neutral positions yield significantly different values for the inertia, which is decisive for torsional vibration. Figure 5 on the right represents this situation. The sample tractor design used here yields inertias from 0.016 to 0.061 kgm^2, which gives us a ratio of about 1:4.

The left side of the illustration shows the model for a two-mass vibration system. Measurements taken on about 10 different tractors provide empirical values indicating that this model is adequate for accurately describing idle mode vibrations. The variable inertia of the transmission is symbolized by different sized rectangles drawn with broken lines.

Figure 6 illustrates the effect that these considerations have on torsion damper design. The illustration is based on the assumption that the engine ignition frequency excites the resonance speed at 300 rpm. With a design like that represented in the sectional drawing on the right, the engine clearly operates above natural frequency during normal idle mode speeds of 600 to 700 rpm. Vibration is reduced by about half. The illustration shows the maximum permissible spring rate plotted as a function of the transmission inertia.

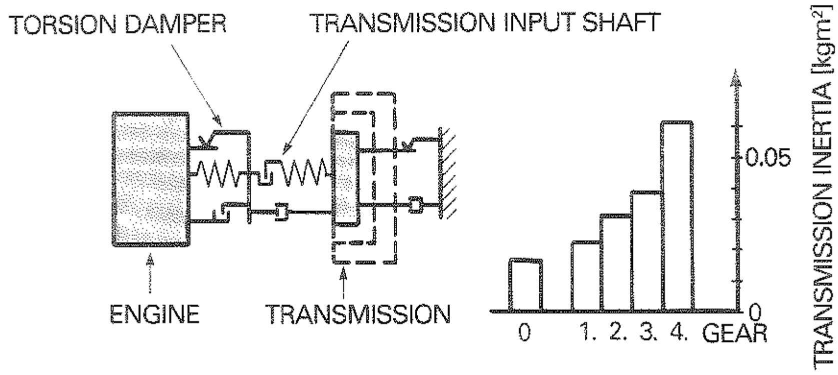

Figure 5: Model for vibration simulation in idle mode

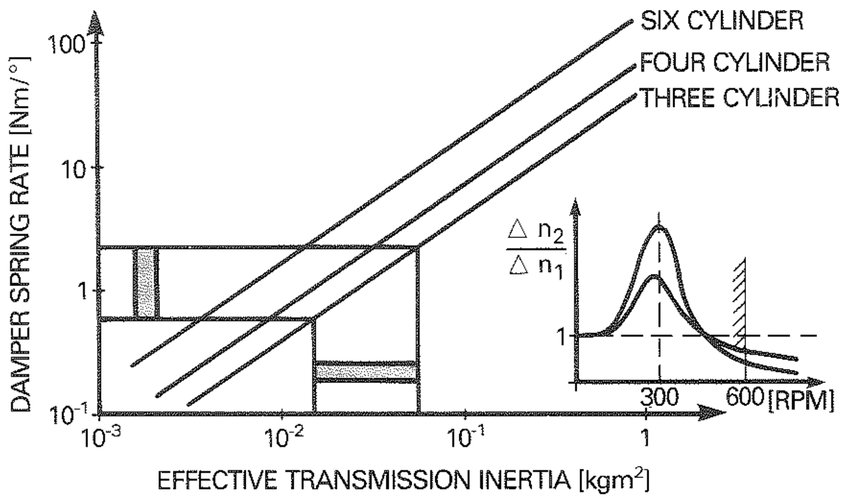

Figure 6: Spring rates for 300 rpm resonance speed

Depending on the shifting configuration, for a three cylinder engine the inertias used in our example would require spring rates of about 0.6 to 2 Nm/°.

Should vibration isolation be required in all cases, we would of course only be able to use the lowest value, 0.6 Nm/°.

Inertias for tractor transmissions in idle mode generally lie in a range from just under 0.01 to about 0.1 kgm^2. Hence, this criterion would require spring rates of about 1 to 2 Nm/° for smaller tractors with three and four cylinder engines. For larger tractors with six cylinders, 2 – 5 Nm/° are possible.

The idle stage characteristics for the production torsion dampers shown in Figure 7 reveal that this requirement is not always met.

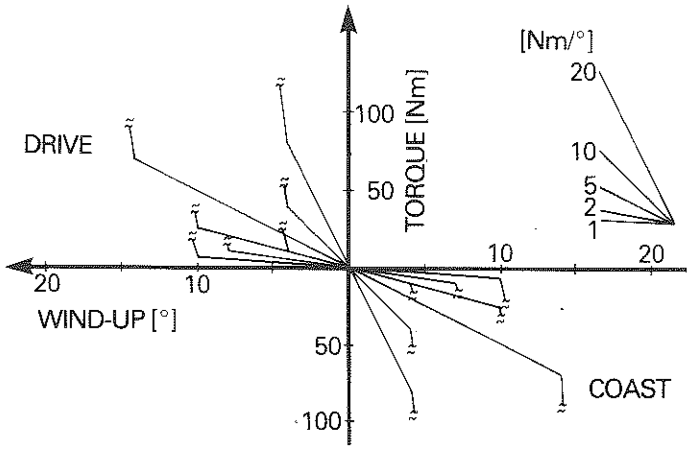

Figure 7: Idle mode characteristics for typical tractor torsion dampers

Some of the spring rates are much higher, although they are successfully used to eliminate idle mode noises. This somewhat surprising situation can be explained as follows:

– As a rule, in absolute neutral position, which also represents the lowest inertia, tractor transmissions exhibit acceptable noise performance. Noise problems increase in higher gears, obviously parallel to the increasing rotational speed of the revolving gears. Consequently, it is often sufficient to design the torsion damper for the two highest gears, which at the same time represent the greatest inertia. In smaller transmissions, spring rates of about 1 – 2 Nm/° are appropriate for this purpose. Some of the torsion dampers shown in the illustration actually have these slopes. Of course, this design poses the risk that the transmission will operate in the critical range in lower gears and that the lack of damping in the transmission or in the torsion damper will lead to resonance and consequently to unacceptable noises.

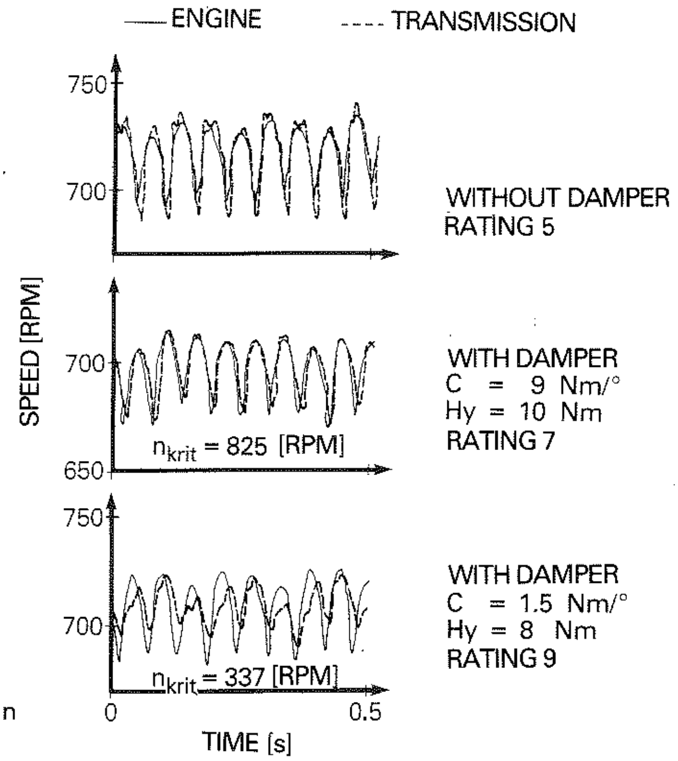

Figure 8:
Vibration measurements in idle mode, 2nd gear

– Some of the torsion dampers shown in the illustration have spring rates of 5, 10 and even 20 Nm/°, which does not lead us to anticipate any vibration isolation above natural frequency. Figure 8 illustrates an attempt to arrive at a physical explanation for this condition based on vibration measurements. We conducted measurements using a three cylinder engine in idle mode at 700 rpm, with the transmission in 2nd gear with an inertia of 0.03 kgm^2. Engine speed is shown as a solid line and transmission speed as a broken line. The top graph shows a measurement using a rigid clutch disk. Because of the unavoidable gear lash between the shaft and the hub, transmission vibration is clearly amplified over the engine vibration, resulting in a subjective rating of 5, which is considered objectionable. The graph in the center shows a measurement using a damper with a slope of 9 Nm/°, which was cited above as unreasonably high; this is evident from the resonance speed of 825 rpm, which lies close to the idle speed. The damper does little to reduce engine vibration, but the situation is clearly better than operating without any damper at all. The subjective noise rating of 7 is classified as acceptable. Finally, the bottom graph shows a measurement using a spring rate of 1.5 Nm/° designed to operate above natural frequency. Transmission vibration has been reduced by about half. The subjective noise rating is 9, which indicates that no noise is audible.

This also explains how an idle stage with an apparently excessive spring rate produces an acceptable noise rating. In such cases, a high spring rate can offer certain advantages. It shortens the required wind-up angle to control drag torque and has a favorable effect on driving operation for low engine loads as well.

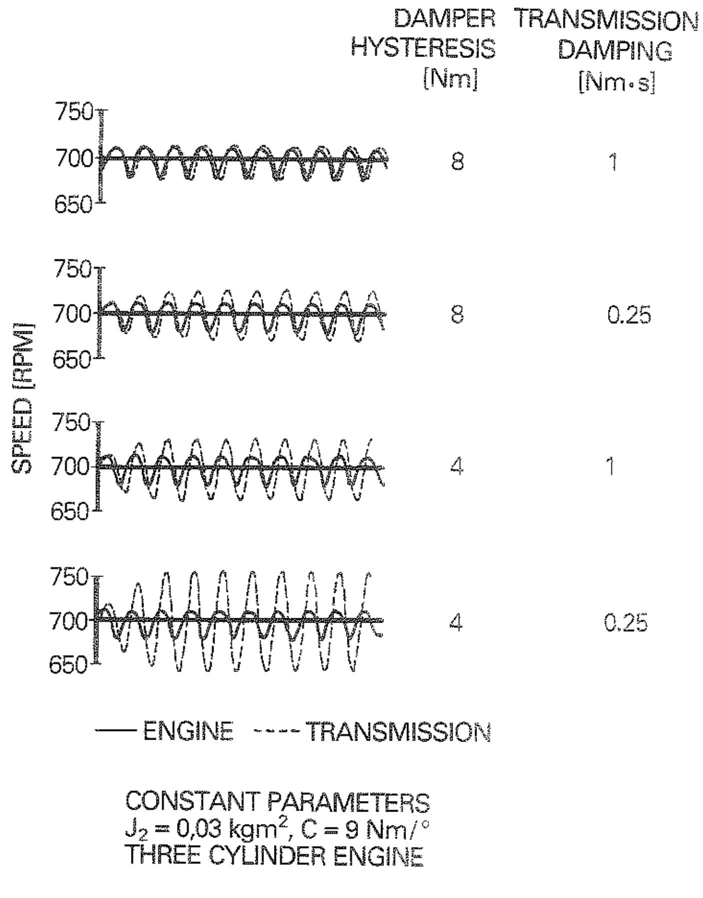

Figure 9:
Computer vibration simulation in idle mode

However, since operation close to resonance speed is impossible without adequate damping, this design is always associated with a certain risk. Computer vibration simulation allows us to estimate this risk by varying damper parameters as shown in Figure 9.

The top graph simulates the apparently excessive spring rate of 9 Nm/°, which is initially satisfactory with respect to noise. The second graph from the top does not change the torsion damper hysteresis, but does reduce transmission damping to a value that is representative for the passenger car range. The vibration curve is already less favorable than it was previously in Figure 8 for measurement without a torsion damper. Finally, the two bottom graphs show that any reduction in the damper hysteresis produces further deterioration.

Before closing this chapter on idle mode tuning conditions, let us take one more look at engine excitation, since the angular amplitudes for engine excitation are critical for determining the length of the damper torsion curve. Figure 10 shows angular amplitude in idle mode at 700 rpms plotted as a function of the specific inertia.

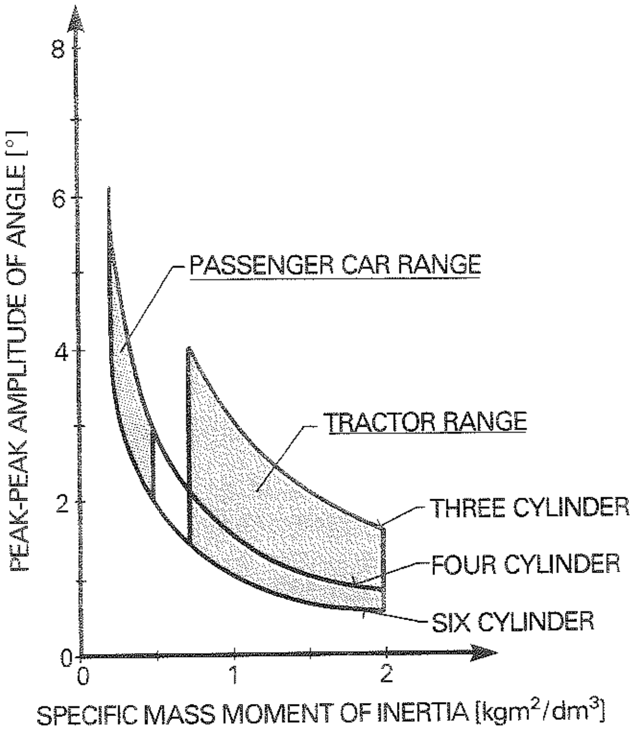

Figure 10:
Angular engine amplitude in idle mode at 700 rpm

With three-cylinder engines, we achieve vibration angles up to 4°. Low spring rates of 1 – 2 Nm/° and transmission drag torques of up to 5 Nm can lead to idle stage torsion angles of 10°. In contrast, the vibration angle for four and six cylinder engines shown on the right side of the range is only about 1°. If it is possible to have a spring rate of 5 – 10 Nm/° at the same time, then an idle stage torsion angle of 2° will be sufficient.

Vibration Performance in Drive Mode

In order to interpret vibration performance for tractor drive mode, it seemed obvious to start out using the 3-inertia model familiar from passenger car tuning. Because of the relatively low torsional spring rates in the half-shafts, for the most part, this model allows the transmission to vibrate on its own with the differential, as if it were disconnected from the wheels and the vehicle mass.

However, vibration measurements taken in tractors during driving operation in the highest gears exhibit inexplicable behavior, as described in the following two examples:

- Figure 11 shows measurements taken on a relatively small three-cylinder tractor, with the same transmission used at the beginning of our discussion while driving under full load between about 750 and 800 rpm. According to the simple 3-inertia model, one would expect to find amplified transmission vibration in response to the engine vibrations because the system is operating below natural frequency. In actuality, we see a clear reduction. This example shows how important it is to conduct vibration measurements.

- Figure 12 shows measurements taken on a relatively large tractor with a completely different transmission and a six cylinder engine driving below 600 rpm. The simple model would have anticipated a slight vibration reduction based on the favorable initial values, but it would not have predicted the nearly perfect isolation.

A vibration model with 4 torsional inertias provides an explanation for this phenomenon, as demonstrated in Figure 13.

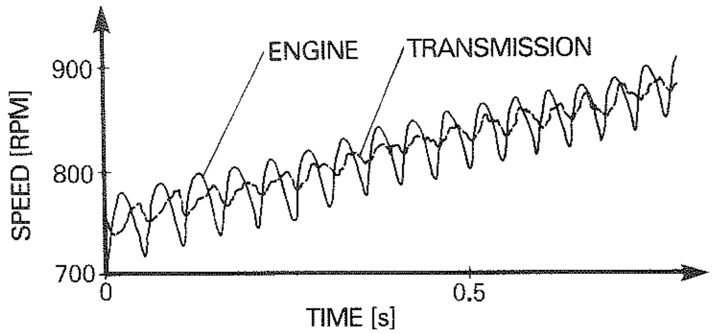

Figure 11: Vibration measurement while driving in the 30 km/h gear, full load; low-power tractor with three cylinder engine

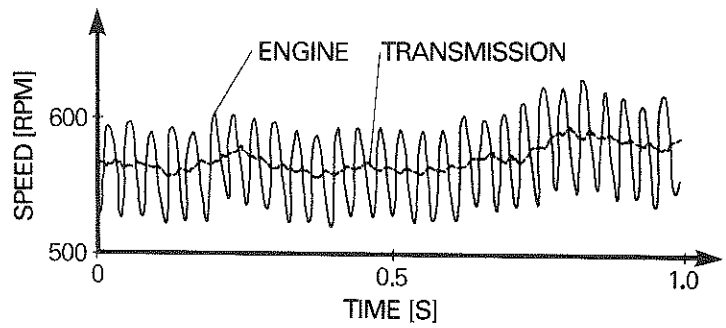

Figure 12: Vibration measurement while driving in the 30 km/h gear, full load; high-power tractor with six cylinder engine

As usual, the transmission is combined in one inertia. However, the drive wheels are treated as an additional mass. The concrete values for our sample transmission are entered as effective mass moments of inertia and spring rates. The numbers without parentheses apply for the 30 km/h gear, while the values in parentheses are valid for a 7.5 km/h gear. The decisive factor is that unlike the situation with passenger cars, the drive wheels can no longer just be assumed to be part of the vehicle mass in the high gears. This is attributable to the low spring rate of the tires in relationship to the drive axles. Consequently, the system will generate a vibration mode in which the transmission and the drive wheels vibrate together in opposition to the vehicle and the engine. Taken together, this produces a considerable mass that shifts the resonance into a favorable position and can lead to good vibration isolation at speeds above this critical condition. In contrast, the high reduction ratios associated with the low gears reduce the effective inertia of the wheels considerably so that it no longer significantly influences vibration performance.

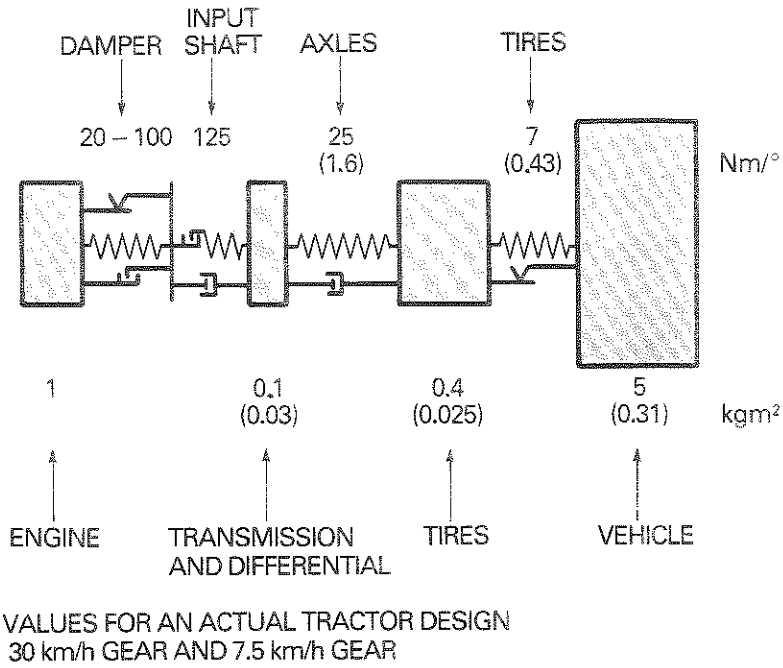

Figure 13: Vibration model for driving mode

Figure 14 shows the possible resonances and the natural vibration modes that occur for these 4 masses. These calculations used the numerical values indicated for the 30 km/h gear and a frequently used tractor damper spring rate of 100 Nm/°.

- The top section of the illustration shows a vibration mode during which the complete drive train including the engine vibrates in opposition to the vehicle mass. In passenger cars, this condition would usually result in unpleasant surging, but this is rarely evident in tractors.

- The center diagram shows the previously mentioned condition where the transmission and the drive wheels are vibrating together. Given the numerical values on which this drawing is based, the wheels vibrate considerably more strongly than the transmission. The transmission does not represent a concentrated, rigid body. Instead, it extends over a great length and represents a flexible component. Hence the representation shown here indicates that vibration amplitudes within the transmission can vary, which means that care must be taken when measuring them.

– The same applies in general for the third possible natural mode shown in the bottom graph, during which the transmission and the drive wheels vibrate in opposition to each other.

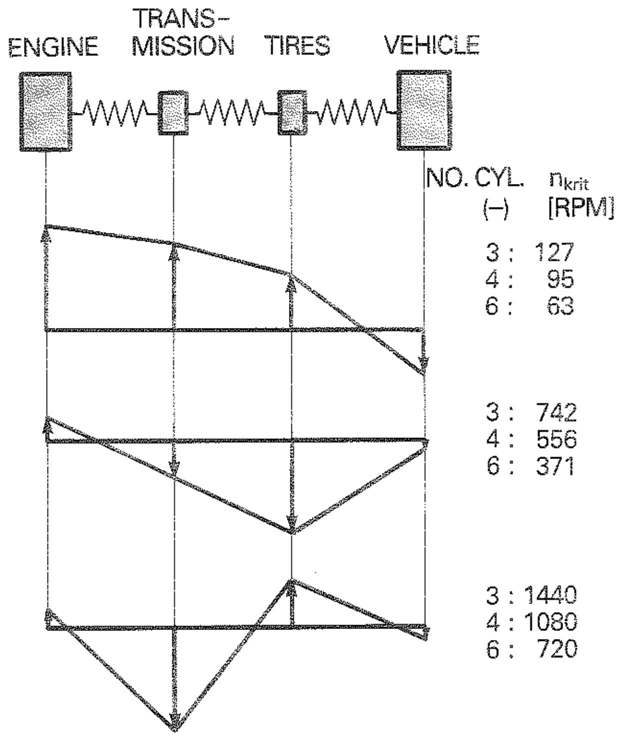

Figure 14:
Four-inertia model – natural modes and resonance speeds for spring rate of 100 Nm/°, 30 km/h gear

The graph sections also show the resonance speeds for three, four and six cylinder engines. For instance, these figures may explain the fact that the spring rate of 100 Nm/° cited for tractors with three cylinder engines leads to noise when driving in the lower speed range, but does not cause this problem with four and six cylinder engines.

Finally however, these drawings only represent relative vibration modes. Figure 15 shows the absolute values, which occur for each given spring rate in conjunction with different damping values. This illustration shows the analysis of measurements taken on our model tractor. On these four graph sections, engine speed fluctuations are plotted as a solid line and transmission speed fluctuations are plotted as broken lines and dot-dash lines, all as a function of the engine speed on the abscissa.

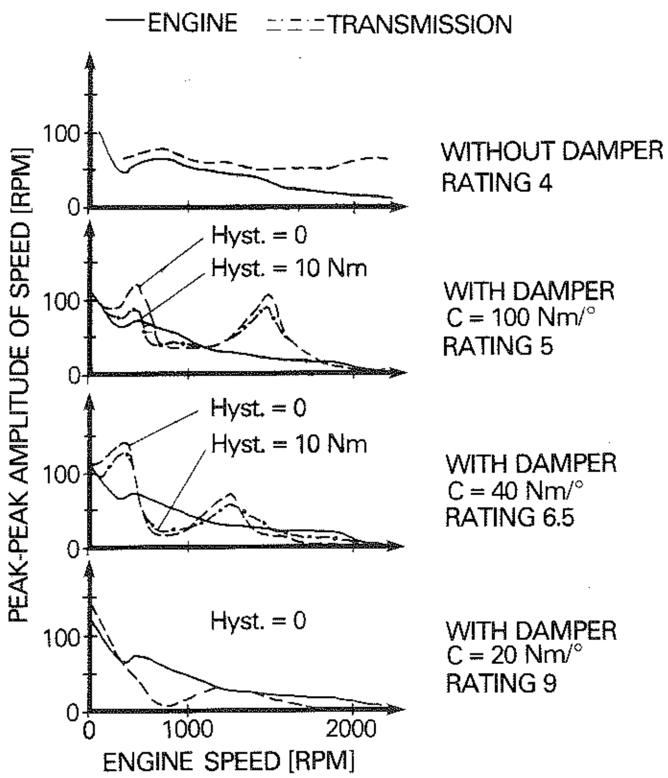

Figure 15:
Analysis of vibration measurements taken during drive mode in 30 km/h gear, full load, three cylinder engine

Viewed from the top down, the first graph shows a measurement taken without a torsion damper, followed by measurements with torsion dampers using 100, 40, and 20 Nm/° spring rates. The measurements taken at 40 and 20 Nm/° exhibit a deep isolation valley between the resonance points, which explains the "perfect" vibration patterns shown at the beginning. On the other hand, the resonance points are clearly defined. At these points, vibration performance is worse than without a damper!

Only the 20 Nm/° spring rate provides improvement over the entire speed range. The soft coupling between the transmission and the engine results in a situation where the transmission damping by itself is capable of suppressing peaks at the resonance speeds!

This means that designers will be able to concentrate on idle mode requirements when they select the torsion damper hysteresis.

Subjective noise ratings are indicated to the right of each of the graphs. The ratings indicate that this case is not particularly critical. However, experience has shown that a spring rate of about 25 Nm/° will generally eliminate noise in problem cases involving three-cylinder engines. Consequently, the conditions represented by the illustration are more or less generally valid.

We can use computer simulation to generate characteristic fields for the analysis of vibration behavior with respect to the essential damping and spring rate parameters in the 30 km/h gear. Figure 16 shows the results of this kind of analytical modeling. Again, the vibration amplitudes for engine and transmission speeds are represented as a function of engine speed.

Moving from the top to the bottom, the damper spring rate was reduced from 100 to 40 and finally to 20 Nm/°. Moving from left to right, the hysteresis was increased from 0 to 10 and then to 25 Nm. The far right column with a hysteresis of ∞ represents a clutch disc without a torsion damper.

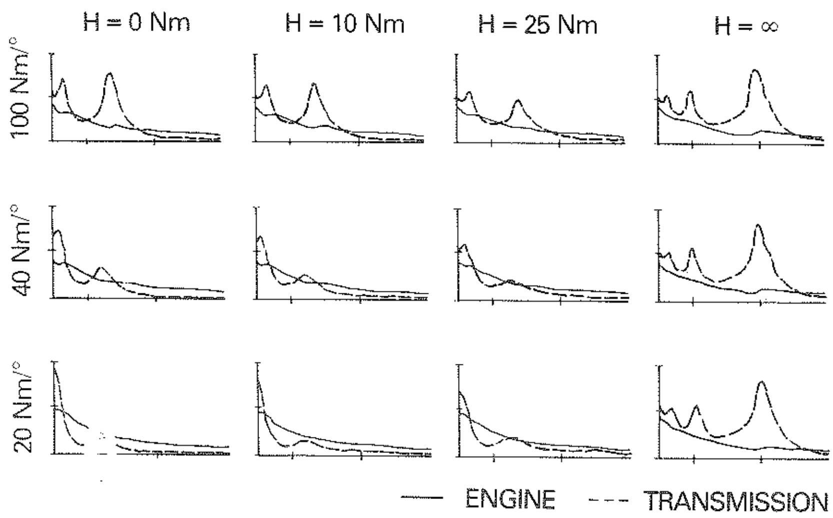

Figure 16: Computer vibration simulation for drive mode in the 30 km/h gear

Above all, the illustration confirms that at low spring rates, very low torsion damper hysteresis values can be used to prevent vibration peaks when unavoidably passing through resonance points.

So much for the higher gears with final speeds of 20 to 40 km/h, which cause the greatest noise problems in tractors. In the lower gears, on the other hand, tractor transmissions fairly rarely cause noises because some of the gear pairs turn at greatly reduced speed. This also means that the effective inertia of the transmission will be considerably lower. Furthermore, as a result of the high reduction ratios, the transmission is de-coupled from the drive wheels with respect to vibration behavior.

Figure 17 shows the analysis of vibration measurements taken in drive mode in a typical 7.5 km/h gear using a three cylinder engine. Again, these graphs show engine speed variation entered as a solid line with reference to the engine speed on the abscissa and the transmission speed variation entered as either a broken or a dot-dash line.

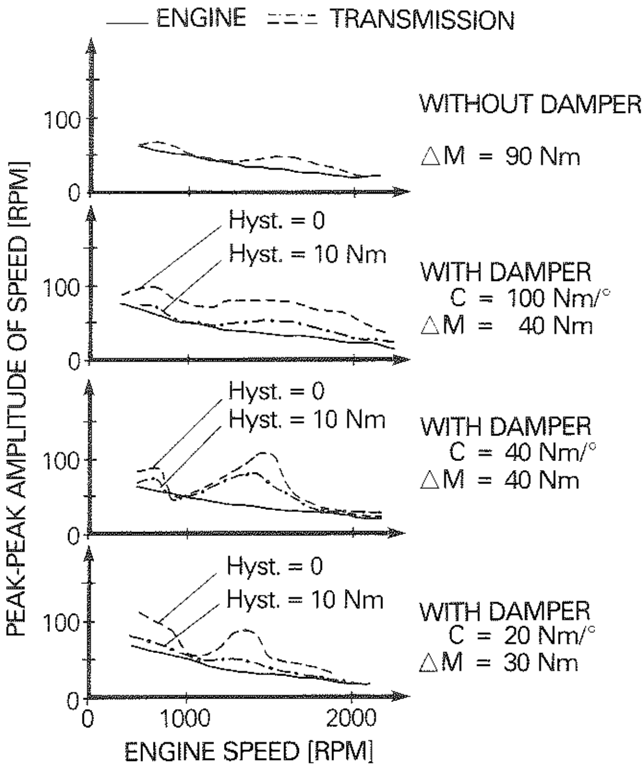

Figure 17:
Analysis of vibration measurements taken during drive mode in the 7.5 km/h gear, full load, three cylinder engine

Viewed from the top down, the illustration first shows a measurement taken without a torsion damper, then measurements for torsion dampers using 100, 40 and 20 Nm/° spring rates. There was no perceivable noise in any of these cases. The vibration curve without a torsion damper exhibits advantages in comparison to operation with dampers. However, the figures listed to the right of the graph sections, which indicate torque fluctuations Δ M at the transmission input, support the use of a damper. These values are measured in a range from 1200 to 2000 rpm. Without a damper, this amounts to 90 Nm, which in this case is approximately 40 % of the averaged torque of 200 Nm. Use of a damper can cut this value down to half or even a third this level. This factor certainly has an influence on the service life of the transmission, since the transmission has to operate in this mode for several thousand hours.

From the standpoint of vibration engineering, satisfactory performance realized without a damper is attributable in this case to the more or less serendipitous convergence of various parameters. As a result, the main resonance lies outside the operating speed range of max. 2500 rpm. The top computer-generated graph shown in Figure 18 illustrates this.

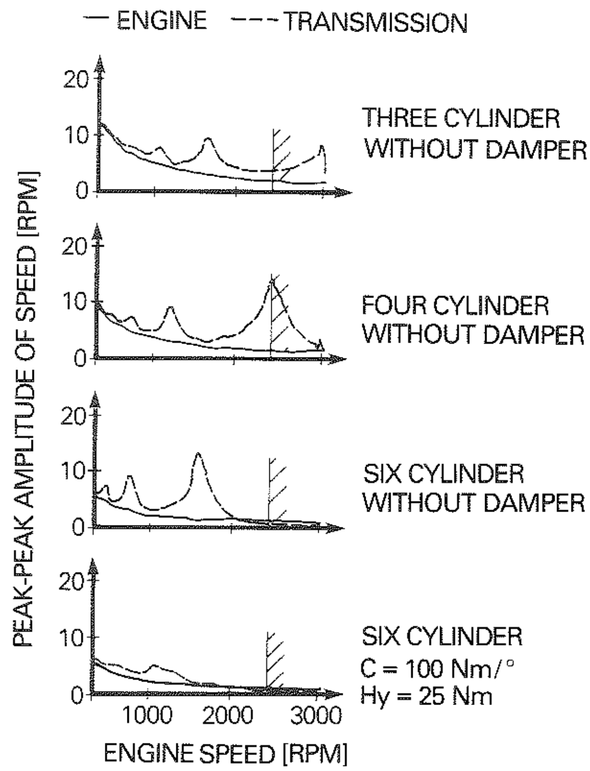

Figure 18:

Computer vibration simulation for drive mode in the 7.5 km/h gear

As an exception, the three cylinder engine has a positive effect in this case.

Computer calculation allows us to estimate the risk involved when using four and six cylinder engines. These engines shift the main resonance closer to or even entirely into the operating speed range. Even a relatively simple torsion damper with a 100 Nm/° spring rate such as the one shown in the bottom graph would reduce these amplitudes considerably.

To conclude this chapter on torsional vibrations during drive mode, it is necessary to take an additional look at engine excitation. Figure 19 shows engine excitation as the vibration amplitude of the engine angle for full load at 800 rpm.The value of 800 rpm was chosen because no noise is tolerated at this speed for tractors. As shown, thanks to positive marginal conditions, the problem can be solved. Certainly, engine vibration angles of up to 5°, which we see at the left end of the tractor range, require correspondingly long torsion characteristics.

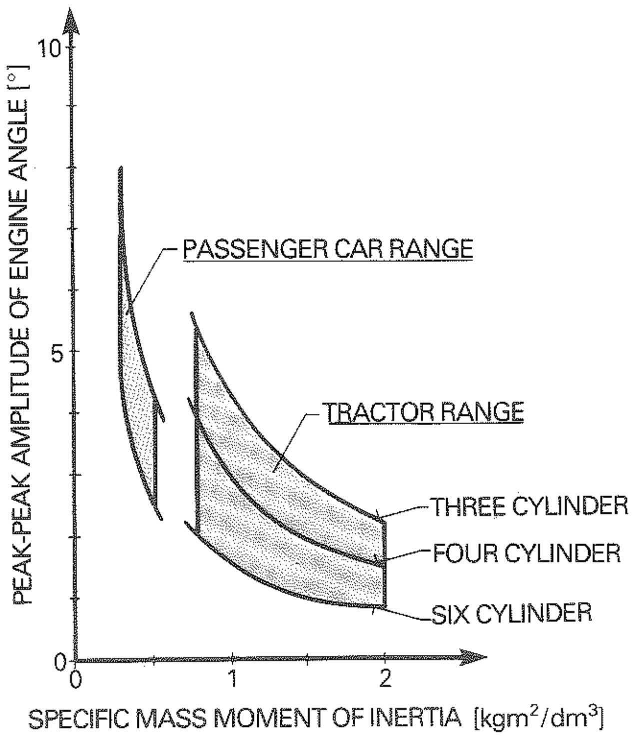

Figure 19:
Engine angle amplitude under full load at 800 rpm

Special Conditions in Power Shift Transmissions

Figure 20 shows a compilation of the most important characteristics curves used in the clutch discs LuK is supplying to the tractor market. Most of these discs do not feature separate predampers, and some of them have relatively long total torsional travel up to 35°.

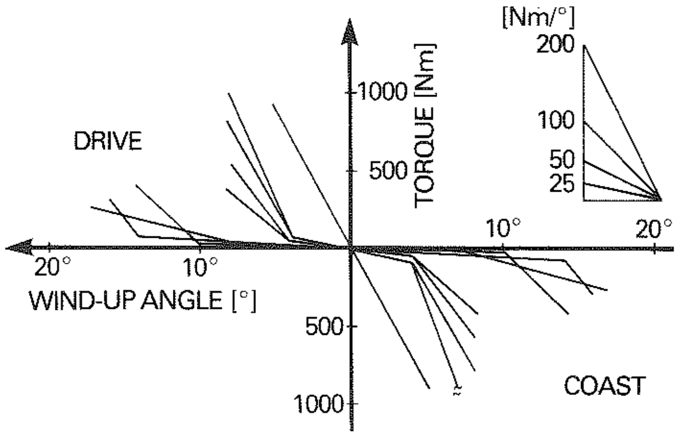

Figure 20: Torsion damper characteristics for typical tractor clutch discs

The illustration also shows the spring capacity that can be achieved using the space between the hub and the facings in the conventional clutch disc range from 225 to 350 mm ∅.

The shaded areas in Figure 21 indicate the position of the characteristic curves for these clutch discs. Damper characteristics designed for newer transmissions are also plotted on the graph.

These curves exhibit different dimensions. The common factor for the transmissions in question is that the main clutch itself is installed inside the transmission and the connection with the engine is achieved using a torsion damper.Obviously, this type of design places special demands on the torsion damper.

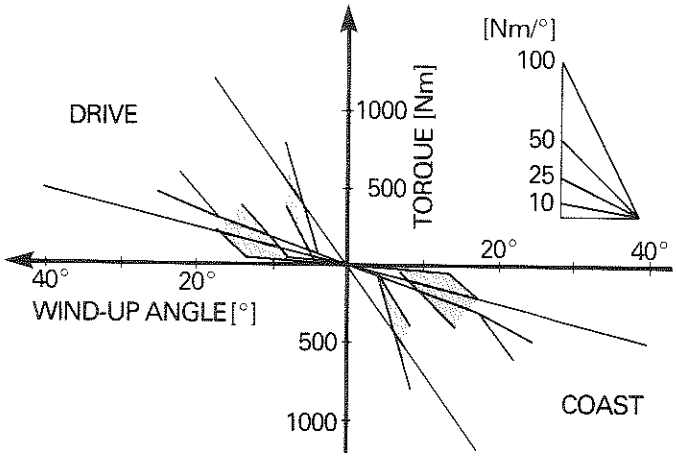

Figure 21: Torsion damper characteristics for newer tractor transmissions

Hydraulic Pump Drives

Let us start with idle mode. Because the damper represents the only connection with the engine, it is also subjected to load from the various hydraulic pumps located in the transmission. If we also account for drag torques for one or more wet friction clutches, we end up with a required torque capacity of 150 to 200 Nm for the torsion damper idle stage. The transmission might require a torsion damper rate of 2 to 5 Nm/° in this case in order to eliminate noise. This would result in unrealistically long torsional travel of 40 to 100° for idle mode. This suggests the serious possibility of introducing a second, independent shaft from the engine to the transmission for the purpose of driving accessories as is the case with conventional transmissions.

Alignment Error between the Engine and the Transmission

The transmission input shaft for this kind of transmission is not usually centered with respect to the flywheel, but its first bearing point on the transmission side will be relatively close to the flywheel. The misalignment between the engine and the transmission is unavoidable – in extreme cases we have to deal with a chain of up to 10 tolerances. Consequently, this misalignment will result in more pronounced friction variation within the damper compared to conventional transmissions. This situation can lead to excitations of the 1st order. Figure 22 shows this condition using sample measurements taken from a six cylinder engine. Transmission vibration has been significantly reduced, but during each third engine ignition cycle, that is, once per revolution, vibration, shown as a broken line, becomes more pronounced.

As a result, under these circumstances, we may have to use a damper design featuring low-friction centering, or even elastic suspension of the damper on the flywheel, in order to avoid these excitations.

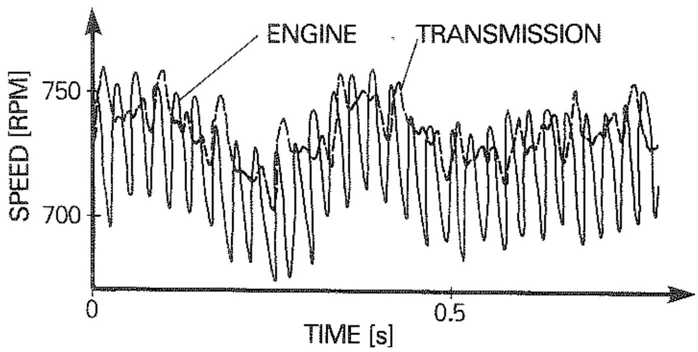

Figure 22: Vibration measurement in idle mode for a six cylinder engine; example of excitation of the 1st order

Effects of Positioning the Start-up Clutch inside the Transmission

Given a certain order of magnitude, slip will prevent the transmission of vibrations to downstream links in the kinematic power flow chain. In conventional transmissions, the main clutch is situated on the engine flywheel, that is at the beginning of the transmission train. During the start-up phase, the engine is de-coupled from the transmission so far as vibration behavior is concerned. However, if the start-up clutch is located in the middle of the transmission, then the upstream transmission components vibrate with the engine during the start-up phase, while the effective inertia can be reduced considerably because the downstream components are disengaged from the engine during this phase. This results in transmission noise during the start-up phase. Figure 23 shows a typical vibration measurement during this kind of start-up.

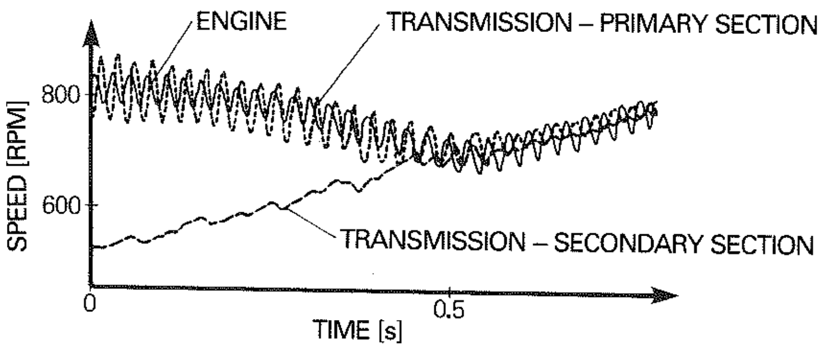

Figure 23: Vibration measurement during start-up with the master clutch situated in the middle of the transmission

This measurement also shows that in stable condition after completion of the start-up process, the front and rear ends of the transmission vibrate with significantly different amplitudes. The discussion of natural modes already made reference to this eventuality. Consequently, it is very important to exercise care when taking measurements.

Given the relatively low inertia for the transmission components upstream of the clutch, a spring rate in the magnitude of 10 – 20 Nm/° may be required for operation above natural frequency. With this spring rate, it is necessary to achieve approximately the rated engine torque, which again can lead to damper wind-up angles of 20 to 50°.

Starting the Engine with a High Transmission Inertia

Starting the engine with a high transmission inertia is a home-grown problem, to use a popular phrase. We didn't have this problem until we started installing dampers in the transmission. At transmission inertias of, for instance, 0.2 to 0.6 kgm^2, starting the engine can involve passing through the resonance with torque peaks of several 1000 Nm or can even make starting impossible. It is questionable whether this specific problem can be resolved at all using a damper.

Power Shift

Power shift causes internal strain in the transmission, which leads to considerable angular motion among the individual transmission components and can manifest itself in extreme noises. An interesting problem. Contrary to previous assumptions, the damper can exercise either a positive or a negative influence, although it is actually in the wrong position to do so.

At this point, our current measuring technology and our options for documenting vibration conditions are probably stretched to the limit. Curves for speeds or accelerations associated with drastically differing noise intensities look almost the same.

This topic must be analyzed more thoroughly in order to find an explanation.

Summary, Forecast

In the case of conventional tractor transmissions – meaning essentially manually shifted machines with the main clutch situated on the engine flywheel – efforts to eliminate noises using torsion dampers installed in the clutch disc have been successful in almost all cases. However, we cannot overlook the fact that in contrast to passenger cars, we can profit considerably from relative low engine irregularity and high internal transmission damping. Changes in these two parameters will determine the future development of the torsion dampers that will be installed in clutch discs.

New designs that integrate the main clutch into the transmission will require significantly higher torsion damper capacity. It is even possible that we may need to use the arc spring technology developed for the dual mass flywheel in order to implement these solutions. At any rate, the high wind-up angles prevent the use of conventional dry dampers.

In addition to solving individual problems, LuK is also faced with the task of producing standard components in this area, which are urgently needed because of the fairly low quantities we experience in the tractor market.

In spite of their complexity, it is possible to use computer models to simulate the vibration behavior of tractor transmissions. It is impossible to use these calculations to predict noise behavior. However, they do provide decisive information on the physical analysis of vibrations and thus can lead to speedier solutions. Furthermore, they offer an appropriate means of estimating critical boundary values for functional parameters during later field application.