Software for automized transmissions –

Intelligent driving

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Introduction

The use of software and electronics is an increasingly prominent feature of all areas of our lives. This trend is also reflected, and particularly so, in automotive engineering. LuK is moving forward with this development and has been involved for a considerable period in the automation of the drive train. In this field, control software is not an end in itself but offers tangible advantages that can be experienced in a practical sense.

LuK is concentrating on automation of the clutch. The progress made so far can be applied to a large number of systems, namely all those with clutches. In particular, these are automated manual transmission (ASG), double clutch transmission (DCT) and clutch-by-wire (CbW).

This paper presents investigations into the robustness of the software for double clutch transmissions and strategies for anti-judder control and slip control that LuK offers its customers. What is important here is that LuK does not supply simply a component but offers a "clutch system" comprising software and hardware including strategies that are modified and optimized in consultation with the customer for the specific application.

Robustness of the double clutch control

One example of the essential and successful close partnership between software and hardware is the control system for the double clutch. LuK is frequently asked about the robustness of the double clutch control system: How well can changes in clutch characteristics be detected? How well does the software respond to deviations from the ideal state? The background to these questions are the high demands for comfort that are placed on modern automated transmissions. Comfort must of course be ensured not just over short periods of time but continuously through the life of the vehicle.

The robustness of the dry double clutch is often questioned because it is subjected to thermal loads in extreme situations due to the absence of oil cooling, and its characteristics may change quickly in the short term. If these situations can be securely controlled, it offers long term advantages over wet clutches.

In wet clutches, the interaction between the oil and the friction partners is critical; this is influenced significantly by the additives in the oil. Ageing and extreme situations may place strain on the oil and break down these additives, leading to deterioration in its quality over time. The dry clutch in contrast practically renews itself since any damaged layers are worn away by normal wear and the clutch can thus maintain its characteristics over the long term.

LuK applies suitable strategies to master even the short term changes in the characteristics of the dry (double) clutch.

On the one hand, an important role is played by the adaptations that provide a suitable model of clutch behavior at anytime for the clutch control system. LuK is using a drive train observer. In control engineering, the term "observer" describes a model-based method for determining internal process information (cf. Figure 1). In the case of the clutch, the unknown internal process information is the torque capacities of the two clutches, which are either directly estimated or described using parameters such as touch points and friction values. In this way, changes in clutch characteristics are quickly detected and passed to the control system.

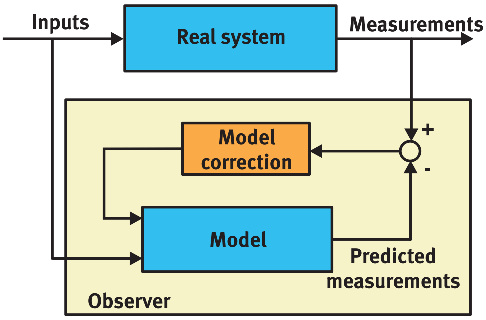

Figure 1 Concept of observer

On the other hand, robust strategies are required that can handle certain deviations in clutch parameters. LuK has carried out investigations using a prototype with a double clutch transmission in order to check the robustness of the control system.

Reference

In order to check robustness, a reference situation is first measured and assessed on an objective basis. A defined reproducible error is then introduced into the system and the reference situation is run and assessed again.

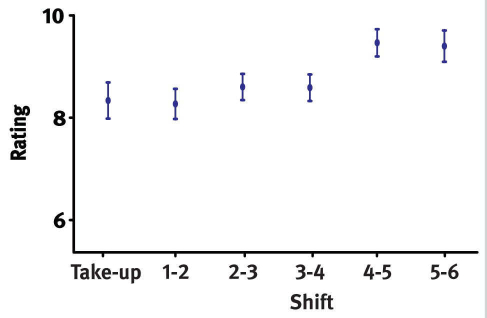

Figure 2 Rating of reference measurements

In order to keep the assessment objective, LuK uses the program AVL-Drive [1] in this example. The program processes several measurables such as longitudinal acceleration, vehicle velocity and engine speed. Based on these values, situations are classified for example, as take-up, powered upshifts and coasting. Physical parameters are then determined for these situations and converted into the established ATZ scores using various weightings. This gives a system that allows assessment of driving situations without having to rely on the subjective perception of a test driver.

The reference situation selected comprised take-up and powered upshifting to 6th gear with 30 % accelerator pedal. Figure 2 shows the results of the reference measurement. For the situations described, the mean value of the ratings as calculated by AVL-Drive and the standard deviation were recorded. The data summarizes 18 cycles. The scatter of the results is relatively small; as expected, gearshifts in the higher gears are rated better since, due to the lower ratios, torque changes have less effect on longitudinal vehicle acceleration.

Modified system

Physical manipulation of the clutch is very time-consuming and hard to reproduce. For this reason, the internal control model of the clutch was manipulated instead. This has approximately the same effect on the control system as a physical change since the internal model is the only information that the software has about the real clutch. In order to achieve a significant change in gearshift quality in the tests, the touch points of both clutches were offset by 1,5 mm. The overall clutch travel is about 7 mm. The touch point describes the actuator position at which the clutch is transmitting 3 Nm. The change in touch point thus corresponds to a parallel displacement of the clutch curve on which the transmissible torque is plotted over the travel. In this case, the clutch therefore transmits more torque than the control system anticipates. The “friction value” describing the mean pitch of this curve remains unchanged.

After the touch point manipulation, the adaptations were deactivated in a first test and the reference situation was run for 6 cycles. The results are documented in Figure 3. The ratings and standard deviations are recorded again and the reference measurement is compared with the manipulated situation without adaptation. As expected, the ratings are significantly worse. It should be noted, however, that take-up quality is still rated as good

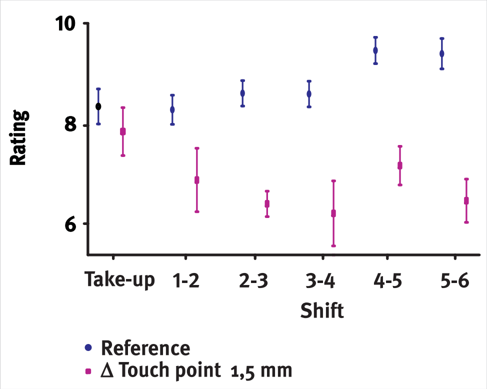

Figure 3 Rating with manipulated touch point and deactivated adaptation in comparison with original state

(better than 7). Due to the relatively long take-up and smart control, the take-up strategy can intervene to make corrections and compensate for the error. During the gearshifts this is hardly possible with the relatively short slip phases.

Adaptation

In the third stage of the robustness study, the adaptation of the clutch curves was reactivated. The results are documented in Figure 4. In addition to the reference ratings, the ratings are shown for 6 consecutive cycles. As expected, the rating of take-up is good from the beginning despite the touch point error. The other gearshifts are assessed as just as good as in the reference measurement. Only the first 1st to 2nd shift and the first 3rd to 4th shift are assessed as somewhat inferior, albeit better than the gearshifts without adaptation.

The reason for these different results is easily explained: During the take-up, which occurs with clutch 1, the adaptation can detect and correct the error in this clutch. The quality of the 1st to 2nd shift cannot benefit from this fully since it is essentially dependent on the accuracy of clutch 2. In contrast, the 2nd to 3rd shift is significantly dependent for quality on clutch 1 and is already very good. The 3rd to 4th shift, also influenced by clutch 2, is not yet good. The short adaptation time in the 1st to 2nd shift was not yet sufficient. In the case of the next shift on to clutch 2, the 5th to 6th shift however, the adaptation has corrected the error to such an extent that the original gearshift quality is restored.

These initial investigation results show clearly the performance capability and particularly, the speed of the adaptation. The take-up strategy can also react sufficiently quickly to clutch deviations due to the somewhat longer event. The entire system is very robust in relation to extreme changes at the clutch. It is clear that, through LuK's combination of mechanics and software, extremely robust systems can be projected.

Anti-judder control

A further aspect of intelligent interaction between the clutch and software is the prevention of clutch judder. The problem of judder has long been familiar to all engineers working in the drive train field, in both wet and dry clutches. Although considerable progress has been made in the development of linings and oils as well as design features, judder is still a live issue. One reason for this is that drive train efficiency is being continually improved, inherent damping is thereby reduced and sensitivity to judder increases as a result.

LuK is working intensively on the subject, since the demands on dry clutches in particular are steadily increasing. If the double clutch is to succeed as a replacement for the torque converter, it is necessary to achieve reliable prevention of judder.

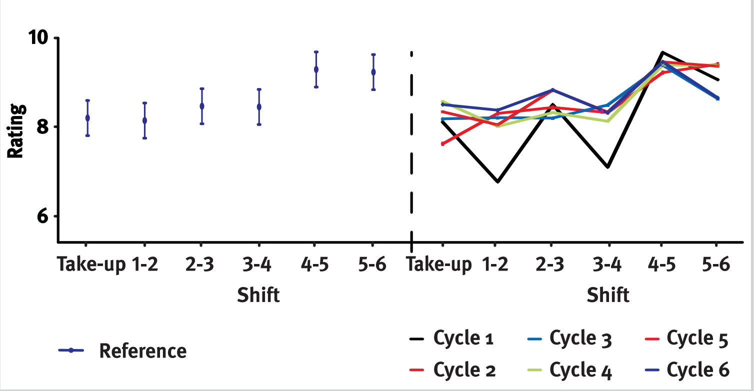

Figure 4 Trend for rating with activated adaptation in comparison with reference measurement

The approaches taken to solving this problem focus on current systems. Efforts must therefore be made in particular to control current clutches successfully using the currently available sensors and actuators. The second step, namely preventing judder when using more judder-sensitive ceramic clutches or with a widening of tolerances, can only be addressed after that and the cost issues must be critically reviewed.

Attention must be drawn here to an apparently trivial but important fact: in typical dry clutch applications, judder occurs only rarely and sporadically in few vehicles. Due to this very poor reproducibility, it is quite difficult to develop anti-judder control. Judder stimuli can of course be created using suitably prepared clutch discs, but these are generally much stronger when they occur than is typically found in vehicles. As a result, targeted development and checking of robustness in all situations is more difficult. In wet clutches, any judder can normally be attributed to damaged oil and, after the first occurrence, is normally reproducible.

Stimuli mechanisms

A detailed description of judder mechanisms is given in [2]. During judder, the rotary inertia of the disc and transmission vibrates with a slipping clutch against the vehicle wheels, or the vehicle mass respectively. The temporally variable clutch torque acts on the transmission input shaft. The corresponding differential equation, in slightly simplified form, is:

$$J \cdot \dot{\omega} + b \cdot \omega + c \cdot \varphi = 2 \cdot r \cdot F \cdot \mu \qquad (1)$$

The inertia of the transmission including the clutch disc is represented by *J*, while the drive train is characterized by the elasticity *c* and damping *b*. *F* corresponds to the clamp force, *r* to the friction radius and μ to the friction value. The typical natural frequency of this system, and thus of judder is approximately 10 Hz and 18 Hz in first and second gears respectively. The anti-judder control is restricted to the lower gears due to the significantly higher drive train natural frequencies in the higher gears, which require extremely dynamic control.

An essential distinction must be made between the two stimuli mechanisms in judder:

1. Self-induced judder
2. Forced / geometrical judder

In self-induced judder, the stimulus is derived from the friction value. Two principal friction value curves are shown in Figure 5. If the friction

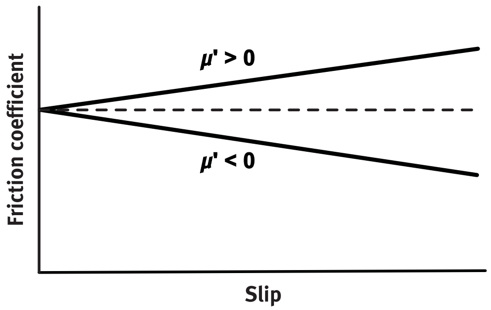

Figure 5 Idealized friction value curves

value curve, taken as linear, is inserted in formula 1, the stability condition (total damping of the system b^* must be greater than 0), can be easily used to derive the following condition for self-induced judder:

$$\mu' < -\frac{b}{2 \cdot F \cdot r} \qquad (2)$$

In this case, the system is stronger stimulated to vibration by the negative friction value gradients than can be compensated by the drive train damping.

With current modern linings, self-induced judder occurs only sporadically in volume use, for example if the linings are contaminated or damp. In the case of linings for extreme loads made from ceramic in dry clutches, however, the negative friction value gradient is a typical characteristic, leading to frequent judder. Even if self-induced judder occurs rarely in volume use, it is clearly felt by the driver and very disruptive. This judder is generally most critical during crawling and take-up, but can also be felt during shifting in low gears.

In geometrical or forced judder, formula 1 remains valid. However, the system is stimulated by an additional stimulus at the frequency of the engine speed, transmission speed or slip speed (or a multiple thereof). The periodic clamp force changes and the resulting torque fluctuations, arising from component deviations and angular misalignments, lead to forced vibrations. These forced vibrations are characterized by the wide, highly variable frequency spectrum, with the amplitude of judder vibrations at a maximum at the drive train natural frequency. Due to the variable frequency of stimulation, geometrical judder is significantly more difficult to control. Also, the torque amplitude applied to prevent judder must correspond to the stimulus amplitude and may become significant.

Sensor requirements

In order to control judder, it must first be detected. The LuK approach to anti-judder control is to detect judder by means of conventional or existing sensors. In many applications, it is already normal to use a transmission input sensor. Alternative sensor types such as torque sensors or longitudinal acceleration sensors generally offer no advantages despite the significantly higher cost.

When measuring speed, the most important boundary condition is that a current speed must be available to the anti-judder control for each calculation step. This means that, in the period between two control interrupts (T) at the relevant transmission input speed (n, in 1/min), at least one tooth must pass the sensor. If the number of teeth per transmission input revolution is described using Z, this gives

$$Z \geq \frac{60}{nT} \tag{3}$$

If judder control is to operate above a speed of 500 1/min and at an interrupt time of 2,5 ms, for example, this means Z=48 teeth per revolution of the transmission input shaft. Experience at LuK shows that the stated speed limit is normally sufficient. If judder vibrations below this limit – such as those during crawling while stationary or in the very lowest speed range – must be controlled, an additional, higher performance sensor system is required. In particular, additional trigger wheels are required because the transmission gears normally used as emitters do not have the necessary number of teeth.

Possible solutions

LuK has developed two different approaches for the active prevention of judder. The first is a feed-back control system in which judder is measured in each control interrupt and correction is calculated. The second approach is a feed-forward control system in which measurement is carried out over a judder half-wave and control is applied in the next half-wave.

In both approaches, it is absolutely necessary to calculate a "judder-free" filtered speed without phase retardation. Judder can then be determined from the difference calculated between the actual speed and filtered speed.

Figure 6 shows the principle of the anti-judder feed-back control system. The upper diagram shows the filtered mean speed and the real speed, both uncorrected and after anti-judder control. The lower diagram shows the nominal torque generated from the take-up strategy and the total torque arising from the overlaying of the anti-judder measure. As soon as the anti-judder control is activated (in the diagram, after the first half-wave), the actual deviation in the judder speed from the filtered speed is calculated at each control interrupt and a corrective torque is determined. The clear advantage of this strategy is that a relative rapid reaction to any deviation in speed is possible. This applies in particular if the stimulation frequency or stimulation amplitude changes quickly. This strategy is also effective with

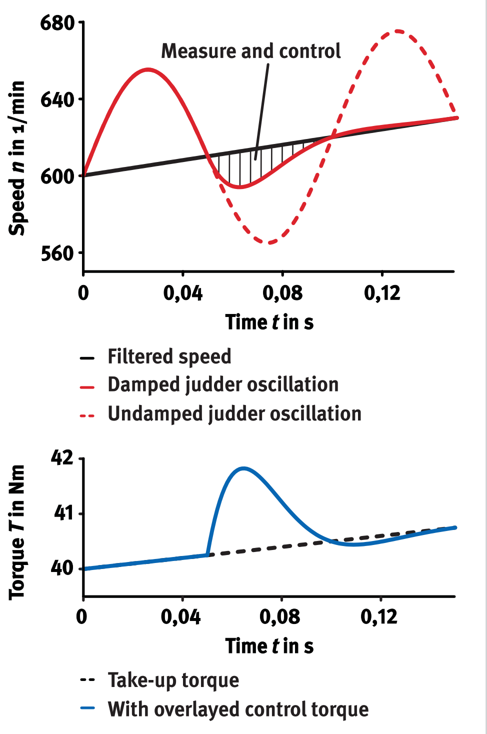

Figure 6 Feed-back control system for judder reduction

geometrical stimulation that varies with changing slip speed.

This advantage can, however, also prove to be a disadvantage. In particular, any disruption to the speed signal may lead to an undesirable change in torque at the clutch. In particular, high frequency disruptive signals and individual measurement and filter inaccuracies may be critical under certain circumstances. This can lead to increased judder stimulation, weaker judder damping and excessive load on the actuator system.

In order to compensate the stated disadvantages, LuK has developed as an alternative an anti-judder feed-forward control. In this system, the steps of judder detection, definition and setting of the correction torque are separate.

As the first step, the phase and amplitude of judder vibration are determined, see Figure 7. The size of the maximum deviation of the juddering input shaft from the filtered speed is determined as well as the time of this maximum deviation. On the assumption that the judder vibration is stable or changes only slowly, a correction signal can be calculated for the next half-wave.

The basis of this correction signal is a “prototype” for control of the actuator, see Figure 8. It is stored in the memory of the control unit and specially matched to the dynamics of the clutch actuator. For example at the end of the curve there is a flat area to prevent overshoot and thus stimulation of the vibration. The software needs only to scale the phase position, frequency and amplitude for this prototype in order to achieve optimum damping. The action is shown schematically in Figure 7.

Depending on the success of damping, judder is possibly detected again in the next half-wave. The major advantages of this feed-forward control approach are insensitivity to noise and the significantly lower actuator load. A further important advantage of the measurement method described is that the measurement frequency can be significantly reduced. It only needs to be high enough to allow detection of the phase and amplitude of the judder vibration reliably and with sufficient accuracy.

The remaining weakness of the feed-forward control to be eliminated is that it can only measure and become active every second half-wave. As a result, the frequency changes that (as mentioned) are typical of geometrical judder and very rapid changes in amplitude cannot be sufficiently controlled.

Further development of strategies at LuK is focused on a suitable combination of feed-forward and feed-back control in conjunction with

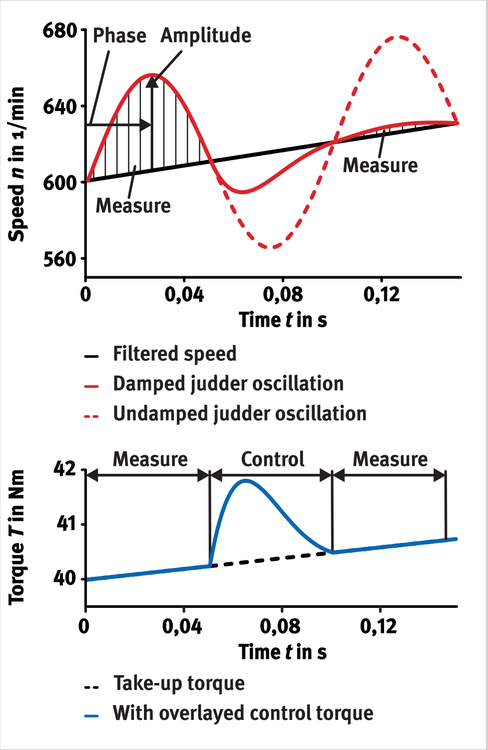

Figure 7 Feed-forward control system for judder reduction

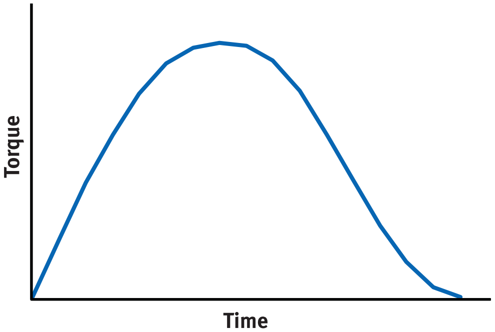

Figure 8 Prototype curve for anti-judder control

highly dynamic actuators and optimized signal processing. The aim of this development at LuK is to provide anti-judder control in automated clutches with minimal sensor requirements. The development of a suitable actuator system for foot-operated clutches that overlays anti-judder control on the manual operation of the pedal is also conceivable.

Results

LuK can present very positive results for both basic strategies. Figure 9 shows the result for anti-judder feed-forward control acting against self-induced judder. On the left is take-up without countermeasures, on the right is take-up with anti-judder control. In order to generate strong self-induced judder, a conventional clutch disc was fitted on one side with ceramic linings. This stimulation is unrealistically high but this very strong disruption can still be almost completely eliminated with the anti-judder control by means of few, small modulations of about 5 Nm.

Figure 10 shows the feed-back anti-judder control for geometrical stimulation, in which the stimulation changes with slip speed.

A conventional clutch disc was manipulated such that geometrical judder occurs. The left side shows take-up without anti-judder control, the right side take-up with feed-back anti-judder control. It can be seen how judder can be significantly reduced with a relatively small modulation but very high frequency. Only slight residual disturbances in the transmission input speed can be identified.

In both cases, the anti-judder strategy can improve judder behavior by 2 to 3 grades.

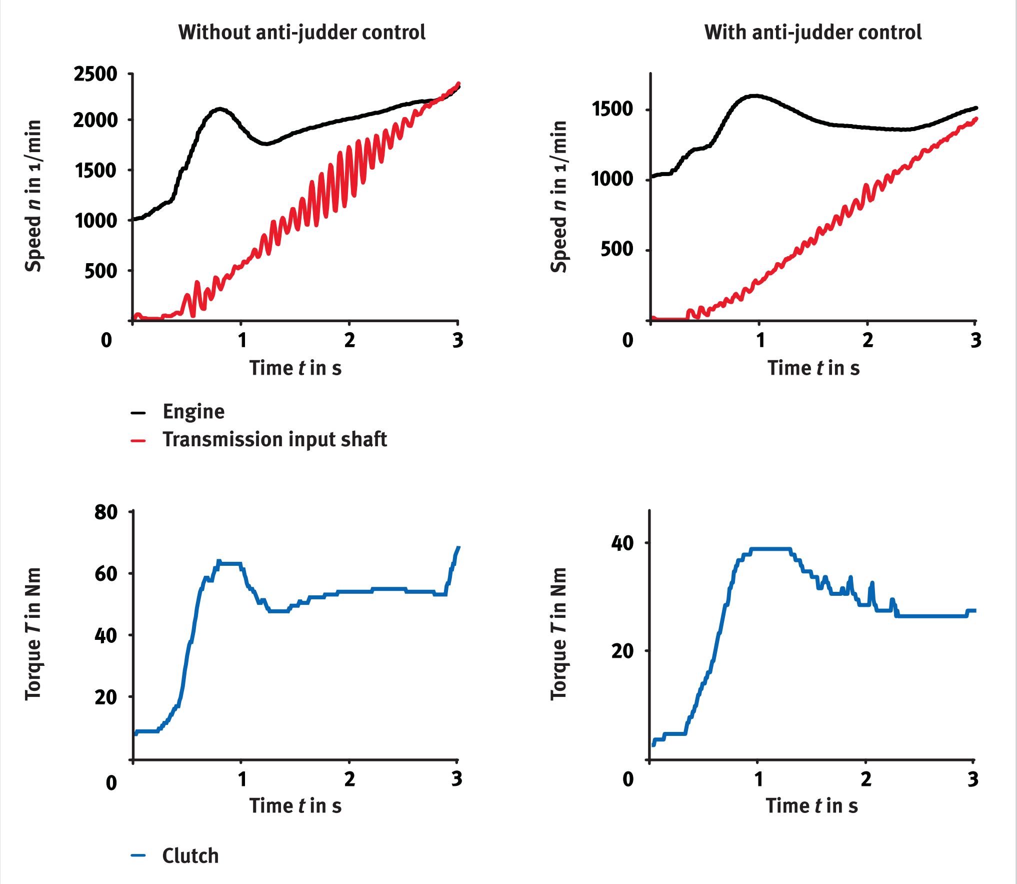

Figure 9 Feed-forward anti-judder control of self-induced judder

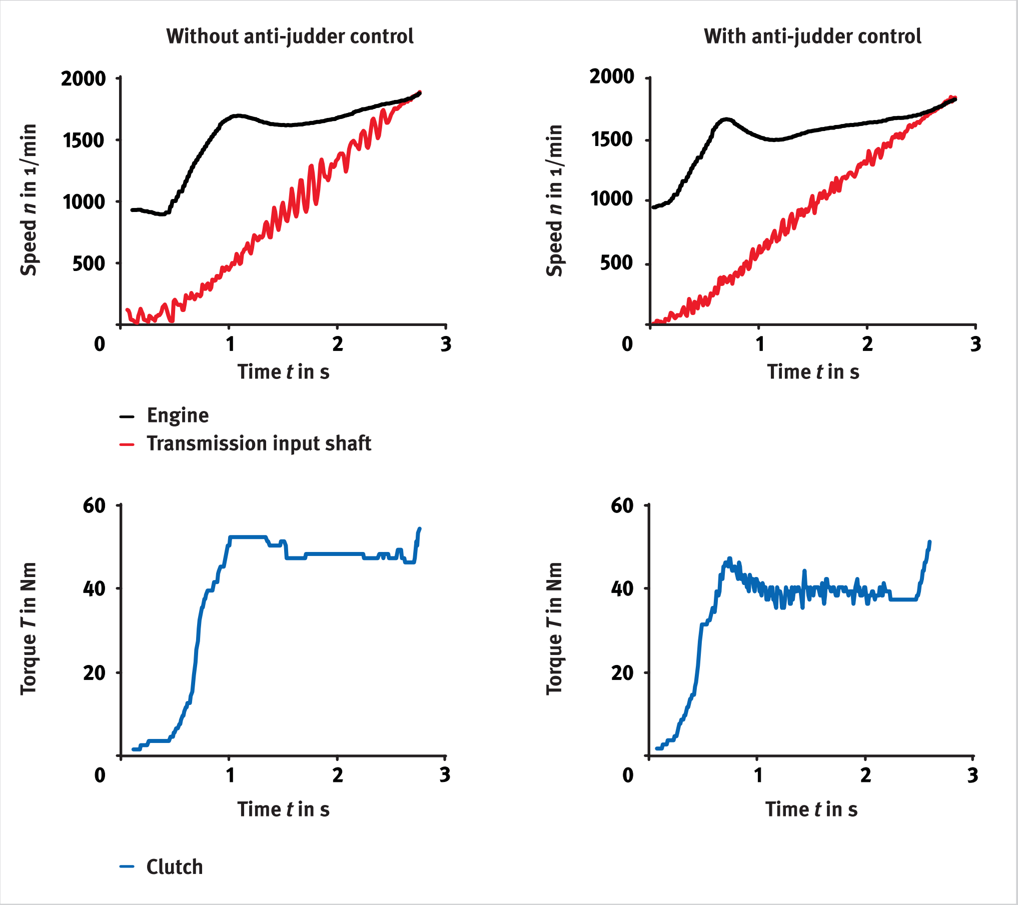

Figure 10 Feed-back anti-judder control under geometrical stimulation

Outlook

The results show that anti-judder control has very great potential. Despite limited fitting of sensors in transmissions and with the currently planned dynamic behavior of actuators, the typical judder problems currently occurring can be robustly overcome in future. The strategies will be rapidly developed to readiness for volume use.

In the future, the limits will be extreme applications such as in dry clutches with complete ceramic linings and a strong judder tendency or severe geometrical stimulation. These limits can only be overcome by additional efforts and costs for actuators and sensors.

Feed-forward anti-judder control in appropriate combination with feed-back control algorithms will further increase the comfort level of clutches in automated drive trains. In this case too, the software strategy will provide intelligent support to the hardware.

Application of slip control

The strategies for controlling slip using the physical relationships are well understood [3] and have been tested on many prototypes. Slip at the clutch isolates the drive train from irregularities in rotation of the internal combustion engine. The control system sets the necessary slip speed by a small modulation of the clutch torque. The slip can and should be relatively small, normally less than 80 1/min. In general, “partial slip” is set under which the clutch alternates at high frequency between gripping and slip for very short periods in response to the irregularities of the internal combustion engine.

Following basic development, a defined and standardized method is now necessary for matching or “application” of slip control for new vehicles. Efficient and robust application across many projects will only be possible by means of a systematic approach.

Application can be subdivided into 3 phases:

- definition of vibration isolation targets
- design and tuning in simulation
- validation and optimization on the vehicle

Isolation targets

The vibration isolation targets must be defined in close consultation between the customer and LuK. Customers as well as LuK can draw on long experience in the field of vibration analysis and drive train assessment. Accordingly, there are already quantitatively well-defined requirements in many cases. LuK can provide support by means of its own estimates.

Since the acoustic characteristics of vehicles and transmissions may vary considerably, adjustment to the target vehicle is always necessary. The so-called “add-on slip controller” from LuK has proved particularly efficient in the vehicle test. A vehicle with a conventional clutch actuation system is equipped with a clutch robot specially developed by LuK. Vehicles can thus be measured in steady state conditions. Various slip speeds are set, allowing on the one hand quick and simple assessment of vibration isolation while on the other hand ensuring comparison with objectively measured criteria.

Theoretical design

Theoretical analysis and simulation are decisive for efficient and targeted design of slip control systems. This is the only way of achieving optimum solutions in a limited time.

The essential input values for this process are the isolation targets described above, the specific stimulation of the relevant internal combus-

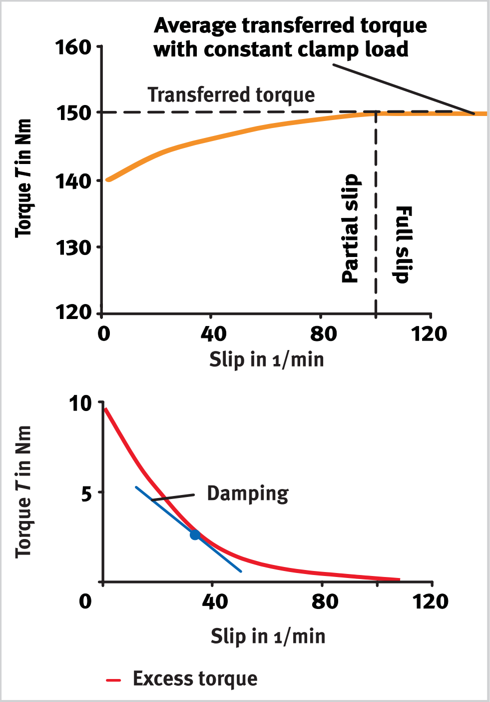

Figure 11 Slip-dependent excess torque and the resulting damping

tion engine and the characteristics of the drive train. The design objective is to achieve both robust function, especially stability of the slip control system, and minimal slip leading to the lowest possible consumption and minimal wear. The results of the optimization are control parameters, the slip map and where necessary, clutch damper adjustment.

One of the most important system characteristics in analyzing the drive train in slip, especially in partial slip, is the damping of the drive train by slip in the clutch. At this point, reference should be made to the LuK Symposium 2002 [3] where it was shown how, due to changes in the slip and grip phases in partial slip, the torque transmitted varies as a function of the slip speed, see Figure 11, upper half. A different view of this curve now used more commonly at LuK is shown in the lower half of Figure 11. As a function of slip speed, it is shown how much stronger the clamp force on the clutch must be if partial slip instead of full slip is to be set. The clamp force is given as an excess torque in Nm. The pitch of this excess torque curve describes the damping at the specific slip present. The smaller the slip, the stronger the damping and thus the self-stabilization of the slipping system. This was discussed in detail at the LuK Symposium 2002.

In order to assess the control quality of the whole system in partial slip, the vehicle and its drive train, clutch, actuators and control system can be computer-simulated. In addition to the control parameters, the dynamics of the clutch actuator and the finite torque resolution play a decisive role in this non-linear system. Together with hysteresis and the damping in partial slip as described above, each overall system has a different set of dynamics.

However, all systems are characterized by the fact that they are non-linear. In control technology and system dynamics, various theories cover such systems. For stable non-linear systems – and such a one is to be designed here – they predict a so-called limit cycle: The system output oscillates with a certain amplitude and frequency about the target value.

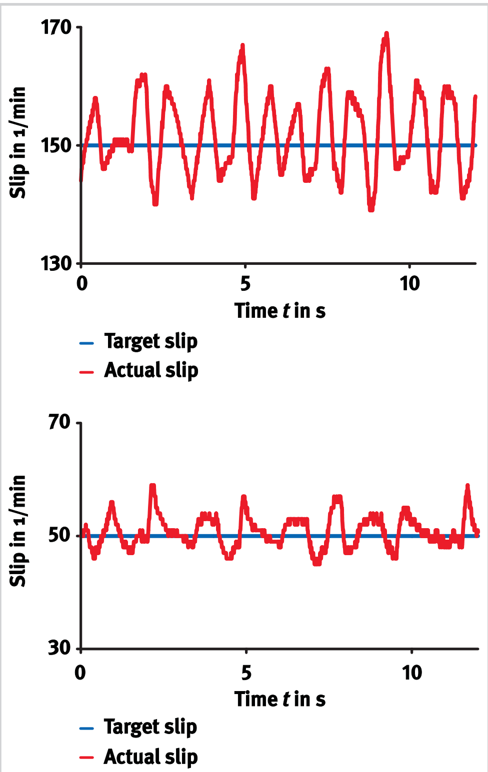

Figure 12 Simulation of the limit cycle at various slip speeds

The simulation models at LuK are now so detailed that this oscillation can be seen in their results, see Figure 12. The simulation result in the upper half shows the low frequency oscillation of actual slip about the nominal slip value of 150 1/min used in the simulation. This oscillation in the range of 1Hz is not problematic in acoustic terms. The vibration amplitude, approximately 10 1/min in this case, is essentially dependent on the damping, i.e. the size of the partial slip. The simulation at a nominal slip of 50 1/min in the lower half shows, as expected, significantly smaller amplitudes of the order of 5 1/min. The amplitude can be regarded as control accuracy and is a decisive value for slip design. The permissible level of deviations is discussed in detail below.

The assumptions made and models used in the simulation have been verified on real systems. Measurements on a LuK prototype with slip control (Figure 13) show very good agreement with the simulation of this system as shown in Figure 12. The limit cycle of similar amplitude and frequency can be easily identified. Since simulation at LuK is now so refined that reality and simulation are in very close agreement, slip control can essentially be designed through simulation and calculation.

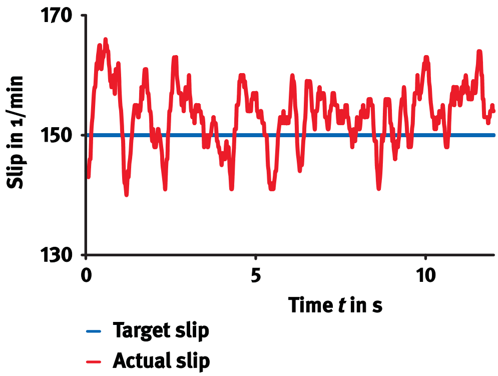

Figure 13 Measurement of the real limit cycle in the vehicle

A further important step is that the occurring limit cycle can now be calculated in an analytical form at LuK. By means of the harmonic balance theory, it is possible to generate this representation for given systems. The decisive factor is not just that the amplitude and frequency can be predicted but that, beyond this, a statement can be made on stability. In Figure 14, the achievable

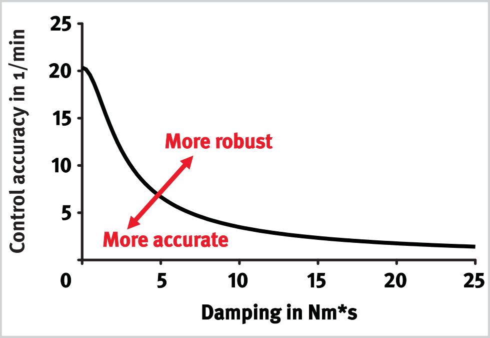

Figure 14 Achievable control accuracy for real systems

control accuracy is presented arising from this formula in relation to the damping of the system (i.e. indirectly in relation to the slip). The analysis includes torque resolution, inertia, control frequency, system hysteresis and other values. Ideally, the smallest possible control error is required, corresponding to the flattest curve possible. In principle, this can be achieved by increasing the controller gain, but robustness is reduced in this case. The practicable limit, i.e. the distance from the stability limit, also known as the "stability margin", must be defined by the experienced engineer such that there are sufficient reserves for variations across vehicles and vehicle life.

How can these analyses of control accuracy be used in design? Once a theoretical slip map has

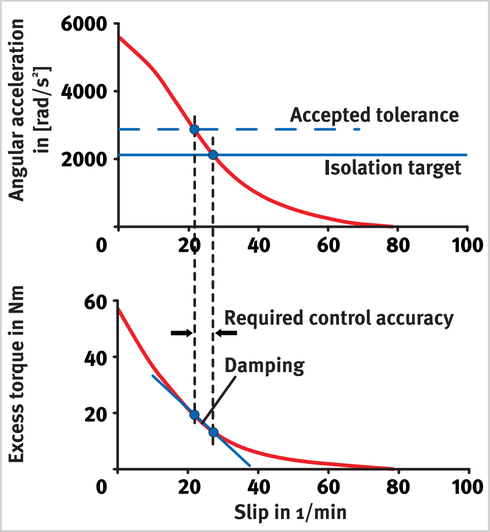

Figure 15 Isolation tolerance and damping for an operating point

been calculated on the basis of the isolation targets, the permissible deviations from nominal slip must be checked. For each operating point, the acceleration amplitude remaining at the transmission input is plotted over the slip, see Figure 15, upper half. A permissible tolerance is defined for each isolation goal. When the isolation goal and an acceptable isolation deviation to be defined in the vehicle is entered, this gives the permissible slip range. This slip can also be interpreted as a control accuracy to be achieved in all cases. Furthermore, the effective damping at this operating point can be read off the excess torque curve presented above (Figure 11), see Figure 15, lower half.

Finally, this requisite control accuracy is entered for all operating points against damping in the diagram for achievable control accuracy (Figure 14). The result is shown in Figure 16. The engineer can thus compare the achievable control accuracy (continuous line) with the requisite control accuracy (operating points entered). These operating points should be on or above the achievable control accuracy. If this is not the case, as in the example shown, there are various possibilities for optimization:

First, the nominal slip speed can be increased. In this case, even larger control deviations do not lead to clutch sticking and the associated deterioration in noise behavior. This is only possible to a limited extent since, as slip increases, damping decreases (cf. Figure 15, lower half) and the operating points in Figure 16 thus creep upwards (the required control accuracy decreases), but also to the left (lower damping due to greater slip).

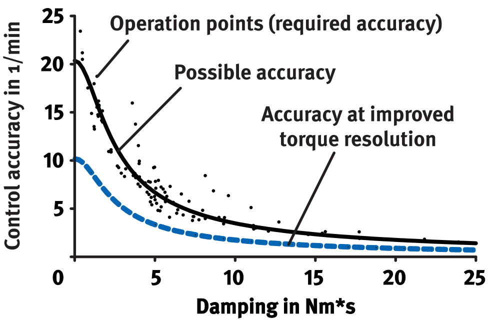

Figure 16 Assessment of stability and control accuracy for all operating points

Alternatively, attempts must be made to improve the system in terms of the stability boundary. Ideally, this can be achieved by increasing the torque resolution of the clutch control system, as shown by the broken line in the example. The complete system, controller and slip map is thus designed using iterative development exclusively in simulation.

Validation in the vehicle

The final stage of application for a vehicle is optimization and validation in the vehicle. The quality of control and isolation must of course be finally checked by testing. The decisive fac-

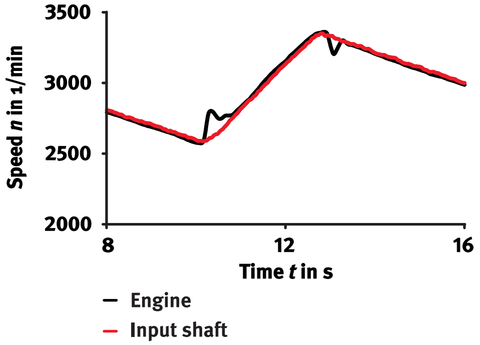

Figure 17 Vehicle measurement of slip control with load reversal

tors in our experience are the quality of engine torque signals, matching of load reversal damping and the behavior of the idle controller. In particular, it is almost exclusively in the vehicle itself that the load reversal behavior can be determined and matched.

With appropriate controller and system design as well as matching in the vehicle, very good slip control results can be achieved (Figure 17). It is also clear that Tip-In and Back-Out can be damped and controlled to an optimum degree.

Power analysis

In addition to pure slip analysis, the power introduced into the clutch also plays an essential role in the overall assessment of slip control. Consumption and wear issues must also be taken into consideration. The significant factor is however the power introduced into the clutch that leads to increased temperature. The clutch considered in the example can withstand a mean power input of 750 W in continuous use. The power introduced by slip control is clearly dependent on the slip and the transmitted torque. Power analysis is aided by the diagram in Figure 18. It shows the slip map, in other words slip plotted over engine speed as a function of current torque, and the power introduced for the specific torque and slip. The horizontal line indicates the maximum permissible continuous power for the clutch.

For most applications, situations can be conceived in which the current power is briefly above the continuously permissible power. Depending on the application, however, the limit value may be exceeded considerably; for example in engines that have high torque at low speeds in combination with large irregularities, thus making high slip speeds necessary. The example shown presents an application in which the permissible continuous power is exceeded at relatively low torques due to the high slip to be set. With a torque of 200 Nm at 1700 1/min, the permissible power is exceeded by almost 30 %.

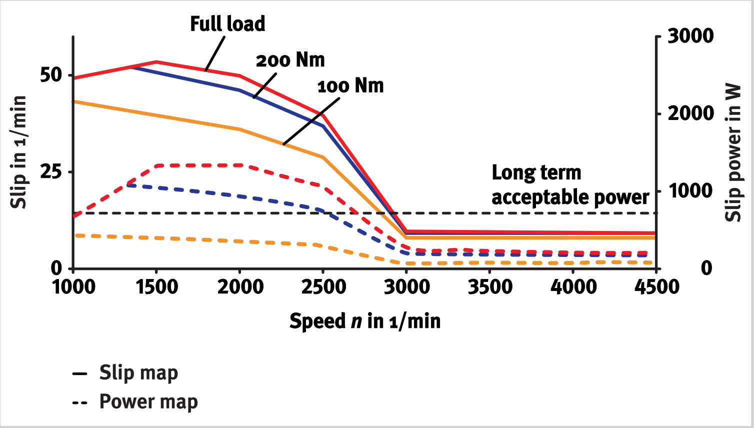

Figure 18 Slip map and mean introduced power

Based on this diagram, it must be discussed with the customer whether the application can be securely achieved only with slip control or whether additional isolation measures such as an external damper are required. The requirements of the customer in relation to handling of the critical situation are decisive. It must be decided in particular whether limits on function such as drive train isolation are acceptable if the permissible power is exceeded for a longer period. The most important protective measures that can be taken in this situation are as follows.

Restriction of the slip power

In this case, the slip power is continuously restricted with increasing temperature. This is achieved by a reduced nominal slip, where the drive train isolation of course decreases under high load.

Shift point displacement

In gear selection in the automatic program, the corresponding situations can be detected (such as a long hill climb) and the gearshift curves varied accordingly to select gears that give reduced slip and/or lower torques at the clutch.

Forced downshift

In the manual mode too, such gearshifts are possible in principle but difficult for the driver to understand.

The increasing severity of the measures shows that there are applications that can be achieved exclusively with slip control, dispensing with a damper similar to Dual Mass Flywheel (DMF), only on the basis of certain restrictions.

Conclusion

Automation places increasing demands not only on the single clutch in ASG or clutch-by-wire but also on the double clutch. The software can and must provide decisive support in order to make the whole drive train system more user-friendly and robust. As an expert partner to the automotive industry, LuK has therefore worked intensively on this subject for a long time.

The examples in this paper show a selection of current results:

- Control of the double clutch transmission in response to changes in the clutch is extremely robust.
- Anti-judder control and management can considerably improve the judder behavior of the clutch, especially at take-up.
- Slip control can be adapted to vehicle-specific conditions quickly and routinely and will be widely used in future.

Thanks to the integration of our knowledge by means of software, the automated clutch systems from LuK give our partners a decisive competitive advantage.

Literature

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