Efficiency-Optimised CVT Clamping System

Reduction of Fuel Consumption through Increased Slip?

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Introduction

Increasing fuel prices and the emission guidelines imposed through legislation cause high demands on automobile manufacturers and suppliers to further improve drive train efficiency.

ator, depending on the clamping forces. A new strategy to detect slip is introduced here.

Figure 1 shows the energy loss proportions for the MVEG cycle, taking the series-produced multitronic® as an example.

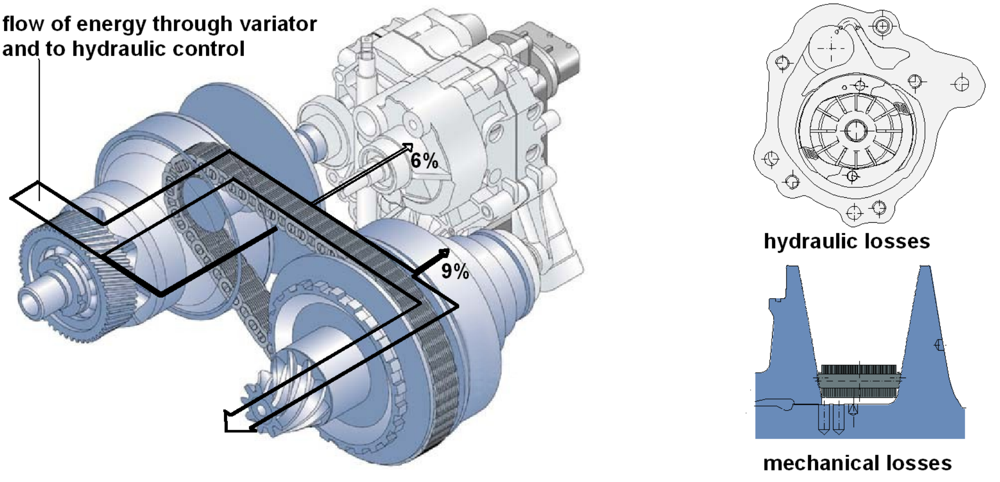

Fig. 1: Hydraulic and Mechanical Energy Losses in MVEG Driving Cycle

As a system supplier, LuK produces CVT components such as pulley sets, chains, hydraulic controls and dampers. All these components have a great influence on the transmission efficiency and consequently on the fuel consumption of the vehicle. The multitronic®, a joint development with Audi, shows that the transmission can make a significant contribution towards reducing fuel consumption. In comparison with the conventional 5-speed stepped automatic transmission, a consumption improvement of around 9% is achieved in the European MVEG cycle [1].

Despite the fact that the CVT has this advantage, work goes on to further improve the efficiency. Therefore this report will discuss in detail two components which contribute significantly to the energy losses. The first section deals with the losses incurred through the hydraulic control and with the possibilities to reduce them further. The second section deals with the mechanical losses in the variator, depending on the clamping forces. A new strategy to detect slip is introduced here.

The total energy loss from the variator and the hydraulics is approx. 15% of the total energy at the transmission input shaft. Further optimisation such as using a smaller pump and a modified clamping strategy for the variator should further improve the efficiency of the system.

Reduction of hydraulic drive power

Selection criteria for the pump size

The pump size is an important evaluation criterion in the selection of the hydraulic concept.

The hydraulic losses can, as shown in figure 2, be calculated through the theoretical delivery volume and the pressure.

low hydraulic losses =

low delivery volume **X** **low pump pressure**

- **pulley sets with LuK double piston**
- **clutch cooling with suction jet pump**
- **low leakage**
  - **pump with axial and radial clearance compensation**
  - **low number of valves**
  - **close clearance and small manufacturing tolerances**
  - **low number of solenoid valves**

- **small over-clamping (slip control)**
- **line pressure dependent on actual engine torque**
- **low back pressures**

Fig. 2: *Reduction of Hydraulic Losses*

Transmission concept

The pump size is determined by the transmission concept and by the transmission leakage. The transmission concept includes for example the choice of single piston or LuK double piston pulley design and the design of the clutch cooling with a suction jet pump, which allows the cooling flow of the high-pressure pump to double.

Depending on the driving conditions, the transmission requires quite different flow rates. The LuK system with double piston and torque sensor is outstanding in that the peak requirements for adjustment, clutch cooling and clamping of the variator equally determine the pump size. The large flow rate required by the pulley sets of conventional CVT's with single piston for dynamic transmission adjustments at hard braking can be reduced to 1/3 through the double piston system [2], [3].

Another important design criterion for dimensioning the pump is the clutch cooling. When driving off, the wet clutch requires a large cooling oil flow at low pressure to remove the energy generated from the slipping clutch. The periods when such a flow additional to that of the high pressure pump is required for clutch cooling are below 2% of the MVEG cycle. So it is not sensible to cover this peak requirement with the high-pressure pump alone. Instead, a low-cost, switch-on/switch-off suction jet pump is used. The suction jet pump requires no drive power, but uses the existing kinetic energy of the oil using the injector effect to increase the flow rate.

Transmission leakage

A further important aspect for the design of the pump size is the internal transmission leakage. This consists of the following:

- Internal pump leakage
- Hydraulic control leakage
- Leakage of sliding seal rings
- Clearance leakage between the movable pulleys and the shafts

All these leakage's are highly dependent on temperature, pressure and tolerances. In a transmission, many component tolerances are included in the leakage factor. The probability of all these elements being simultaneously at a maximum is very low. Therefore, the maximum tolerances are considered through a statistical perspective, which ensures that the pump is not dimensioned larger than necessary. To calculate the transmission leakage, all individual leakage points are determined and combined in a balance sheet.

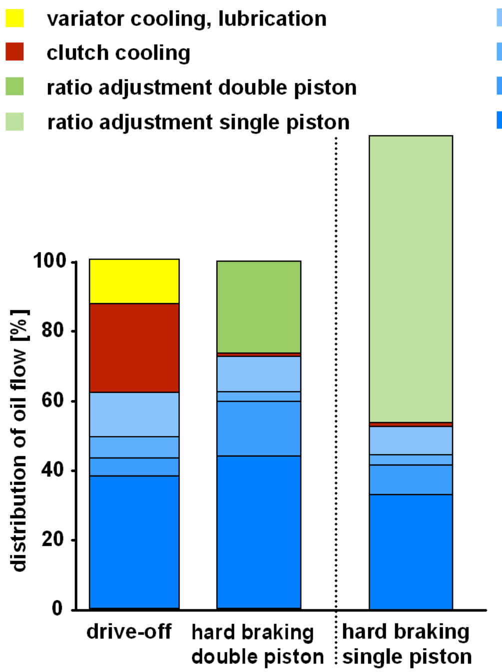

Fig. 3: *Distribution of Oil Flow for the Operating Conditions Driving off and Hard Braking*

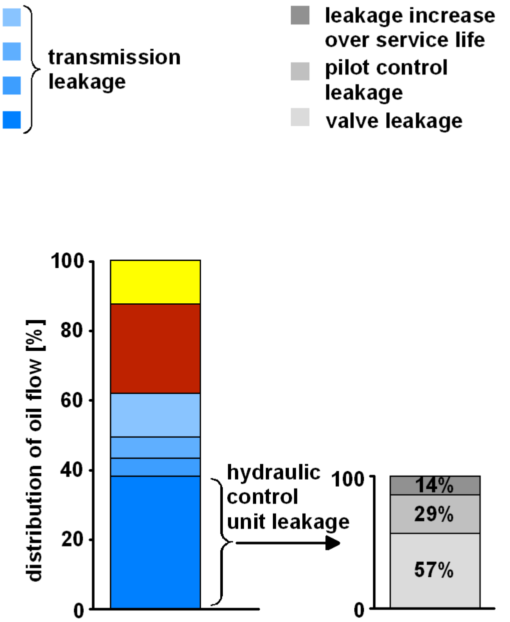

Fig. 4: *Leakage of the Hydraulic Control Unit in Driving off Operating Condition*

Figure 3 shows two critical operating conditions. Both conditions are based on the lowest realistic engine speed, the highest possible engine torque and a temperature of 100 °C, i.e. a low delivery rate of the pump at high system pressure and low oil viscosity.

In the drive off operating condition, the required flow is determined by the cooling oil flow requirement of the start-up clutch, whereas the hard braking mode is characterised by the high flow demand for the variator adjustment towards underdrive. The right column shows the advantage of the double piston system with regard to pump size. Despite the low pressure with a single piston system and the resulting lower transmission leakage, the pump has to be sized significantly larger due to the increased adjustment flow rate [2].

The total leakage, depending on the operating conditions, is up to 70% of the delivered flow.

The largest proportion of the leakage is in the hydraulic control unit itself. Regarding the drive off operating condition, the leakage of the hydraulic control unit is divided into three main blocks as shown in figure 4.

The following optimisation measures to reduce the leakage of the hydraulic control unit were investigated:

- Valve leakage
  - Reduction of valve diameter
  - Increase of overlap lengths
  - Reduction of valve clearance
  - Reduction to a minimum number
- Pilot control leakage
  - Use of 3/2-way control valves in place of 2/2-way control valves
  - Reduction in supply orifice diameters

- Leakage increase over service life due to wear in the bores
  - New coatings technology for optimum valve edge topography
  - Reduction of valve lateral forces through optimisation of flow control and pressure distribution

If all these measures are implemented, then the leakage of the control unit, as shown in figure 5, can be reduced by up to 25%.

The remaining leakage points such as pump, sliding seal rings and clearance between moveable and fixed discs were analysed and optimised through leakage-reduction measures.

The pump was provided with radial and axial clearance compensation [4]. This allows the pump leakage to be reduced to a minimum.

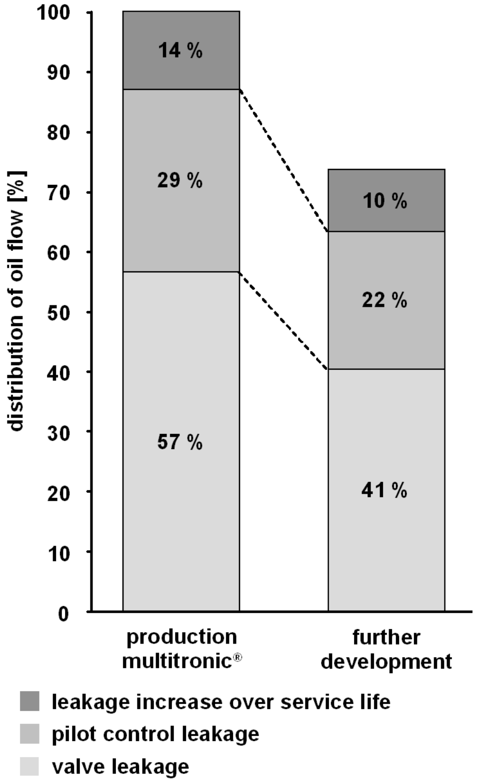

Fig. 5: *Optimisation of the Leakage of the Hydraulic Control Unit in Drive off Operating Condition*

The sliding seal ring leakage was reduced through the use of special lock geometry and new materials with minimal swelling.

The leakage between moveable pulley and shaft can be reduced through the reduction of clearance, increasing seal length and reduction of oval distortion under load by a more stiff design.

Selection criteria for pressure level of pump

If the system pressure increases, so does the leakage and the pump must be sized appropriately larger.

If lower system pressures are selected, then the clamping and adjustment cylinder areas are correspondingly larger, which in turn necessitates a larger pump if adjustment dynamics are to stay the same. An optimum point between pressure level and pulley cylinder areas is found using consumption simulations. The following boundary conditions were taken into consideration when selecting the pressure level:

- System pressure depending on engine torque and ratio
- Optimum point regarding pressure level and clamping cylinder areas
- Low safety factors for the clamping, e.g. variator slip control
- Minimum back pressure in the cooling circuit, to ensure that it does not determine line-pressure
- Low back pressure in the hydraulic control unit through large cross sections with few direction changes

Summary of hydraulic power requirements

Figure 6 shows a comparison between the pump size in the LuK hydraulic concept and other commercially available CVT systems. The engine torques to be transmitted greatly influence the pressures in the control system and consequently have a significant effect on the transmission leakage. The pump comparison was therefore carried out in relation to the engine torque.

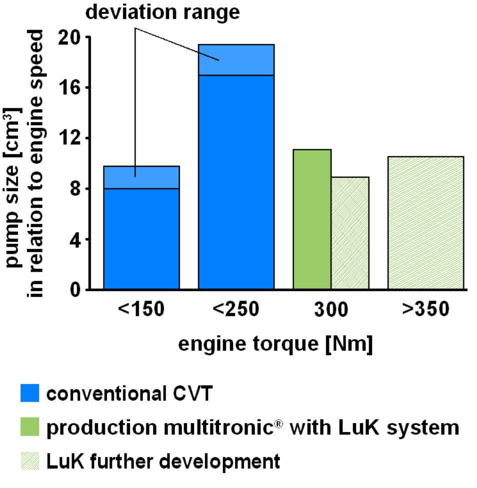

Fig. 6: *Comparison of CVT Pump Volumes*

The benefits of the LuK concept are clearly seen in the small pump, which for engine torques of over 300 Nm is only slightly larger in this design than a pump for a conventional CVT of less than half the engine torque. Even for the development of CVT's with higher transmittable torque, the pump size and consequent power demand will not rise steeply. The higher leakages which occur in transmissions through the higher transmittable torque are more than compensated for by the illustrated leakage reduction measures.

If all leakage reduction measures are implemented at the same rated torque, then the pump size can be reduced by around 20% in comparison with standard versions as shown in figure 6.

Reduction of mechanical losses

Determining slip

The power transfer in mechanical continuously variable transmissions takes place via frictional contact. To ensure the power transfer at the contact points, a sufficient clamping force is required. The minimum possible clamping force to be applied for correct operation is essentially influenced by:

- Quality of torque signal
- Fluctuations of frictional value
- Behaviour of control elements

Due to these influences and the fact that strong slipping of the variator will immediately destroy it, high over-clamping must be tolerated at times with a consequential loss in efficiency and increased component load.

The multitronic®, first introduced into the market two and a half years ago, achieved a significant improvement in efficiency in comparison with conventional CVT's [1]. The clamping force of the multitronic® is dependent on the variator input torque and controlled by a hydro-mechanical torque sensor [2] located directly on the primary pulley set, which provides a reduction of over-clamping.

To further increase the CVT efficiency, a further reduction of over-clamping is crucial.

But only a small optimum range exists between undesirable over-clamping and destructive slipping.

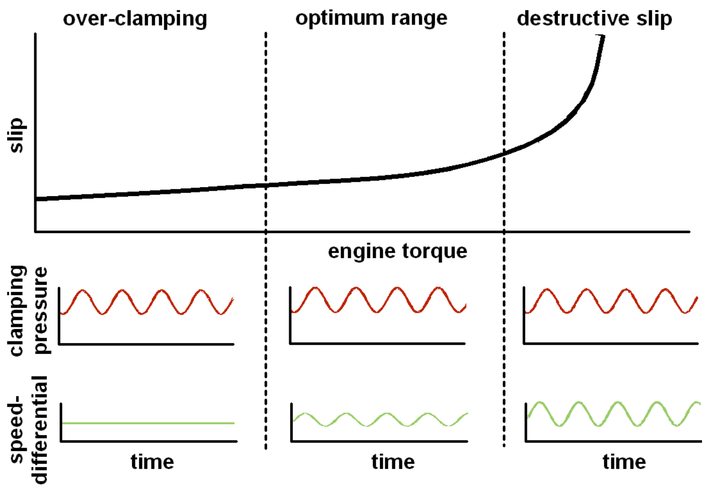

Fig. 7: Principle of Slip Detection (Schematic)

Therefore, a measured variable is necessary, which will be called the control variable in the following. The control variable is used to determine whether the clamping force is within or outside of the optimum range. Optimum in this context means that the functionality of the variator is not limited during its service life, but at the same time a high efficiency is achieved.

The variator slip is a possible control variable to determine the range between the actual clamping force and the slip threshold, at which the variator will be irreversibly damaged.

While developing a method to determine the slip, various physical principles which provide feedback on the slip were investigated, such as measuring the temperature of the cooling oil spray from the pulley surface. Most promising proved to be a modulation of the variator clamping pressure.

The top of figure 7 schematically shows the increase of the variator slip dependent on the engine torque at a constant clamping pressure. In the over-clamping range, the slip is small. In the optimum range, the slip is already slightly higher. With a further increase in torque, the slip increases disproportionately and irreversible damage to the variator system occurs.

Slip in this context means the gliding between chain and pulleys which is necessary for transmitting power.

The process of pressure modulation is shown in the bottom illustrations of figure 7. During this process the clamping pressure is periodically modulated and the pulley set speed differential is measured. In the over-clamping range, the pressure modulation causes no reaction of the speed differential. However, as slip increases, a speed oscillation occurs, with its frequency corresponding to that of the clamping pressure modulation and its amplitude increases with increasing slip. This produces a value which alters depending on the variator slip.

Selection of modulation frequency

Essential criteria for the selection of the modulation frequency are:

- Hydraulic and mechanical natural frequencies
- Comfort and acoustic characteristics
- Variator adjustment behaviour

Figure 8 shows the ideal frequency range for the modulation, where no discomfort would be expected and where there are few disturbance frequencies to interfere with the modulation.

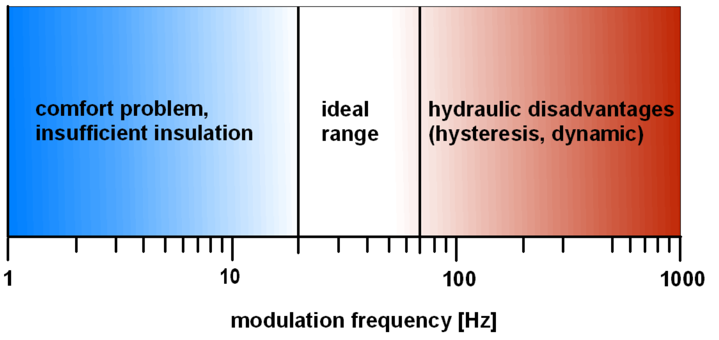

Fig. 8: Selection of Modulation Frequency

Evaluation process

The method of measuring slip as described in figure 7 was tested through measurements.

Figure 9 shows two measurements with pressure modulation on a chain type CVT. The measured signals of clamping force and speed differential are shown over time.

The same average clamping pressure was used for both measurements, which according to the torque sensor arrangement equates to a transmission input torque of about 60 Nm. The transmission input torques in the measurements are quite different with values of 30 Nm and 100 Nm. This produces the extreme relationships shown in figure 7.

The speed of the primary pulley set in both measurements is 3000 min^{-1}, the variator ratio is 2. This results in an average speed differential of 1500 min^{-1}. The speed differential at both torque niveaus show oscillations; however, there are frequencies present which do not correspond to the modulation frequency. Furthermore, despite differing input torques, the differences of the speed differentials given in figure 7 are not visible.

The reason that the signals only vary insignificantly is due to the strong noise contribution, which covers the desired signal. This means that in comparison to the noise, small desired signals must first be made visible through an evaluation process.

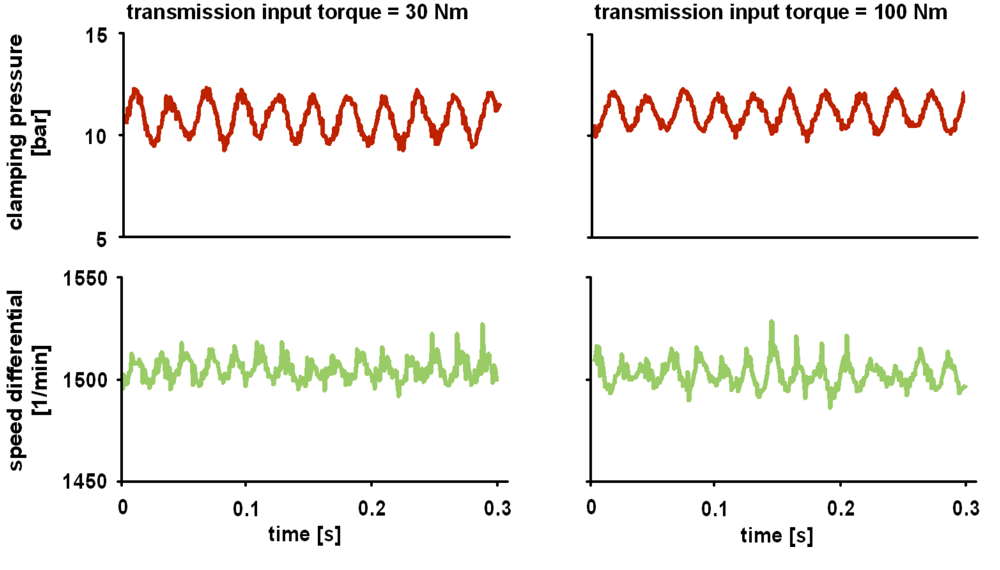

Fig. 9: Measurement Results at Two Different Torques

The lock-in process for indicating a desired signal from a very noisy background is well known. The process is explained in the following, with the functionality proven by the above shown measurements.

As shown in figure 10, the clamping force signal is first converted to a squarewave with a value ranging between -1 and +1.

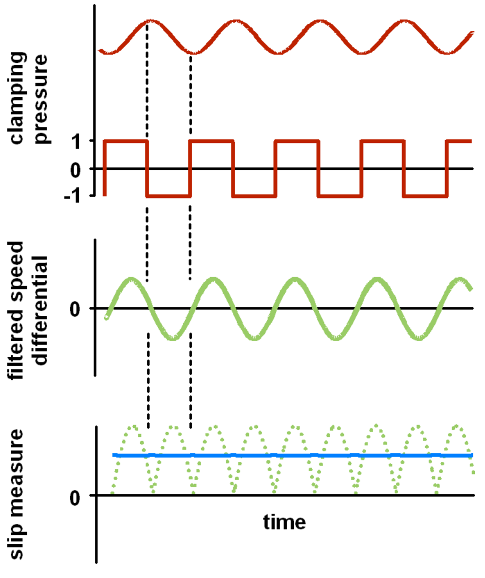

***Fig. 10:** Lock-In Strategy Using Ideal Signals*

This can be realised by using a bandpass filter on the raw signal and generating a square-wave signal in relation to the phase.

The speed differential signal is also filtered with a bandpass filter. If the square-wave signal is now multiplied with the filtered speed differential signal, the negative ranges of the speed differential are transferred to the positive range. The process described can also be called in-phase rectification, as only those ranges which have the correct phase to the clamping force are rectified. All interference elements whose phase is not constant to the clamping signal are randomly multiplied.

The higher-frequency interference elements are suppressed through the subsequent low-pass filtering of the rectified signal, so that the desired signal becomes visible. A comparison between the desired signal and the slip at a geometrically blocked variator shows that the desired signal depends on variator slip. Therefore in the following it is called slip measure.

The measurements shown in figure 9 have been evaluated using the process described.

The resultant slip measure is shown in Figure 11. Significant differences can now be clearly noticed in the slip measure between the signals for the input torque of 30 Nm and 100 Nm, in contrast to the speed differentials plotted over time.

Figure 12 shows the slip measure of several evaluated measurements in relation to the engine torque.

The measurements were carried out with a chain CVT, powered by a combustion engine, at three different variator ratios. The average clamping force was almost constant. The symbols illustrate the measurement points and their deviations.

At all three ratios there is a significant dependence of the slip measure on the engine torque. The gradient of the graph at high over-clamping is at first flat and increases progressively into the area of less over-clamping.

The transmitted torque in the multitronic® at the specified clamping force is also given.

The system comprising of pulley sets and chain was evaluated after the measurements. No damage was detected despite a slip measure of up to 16. This means, that at least a brief period of higher slip than realised in the multitronic® is permissible. At the current state of testing, it is not possible to assess which slip measure can be sustained long-term.

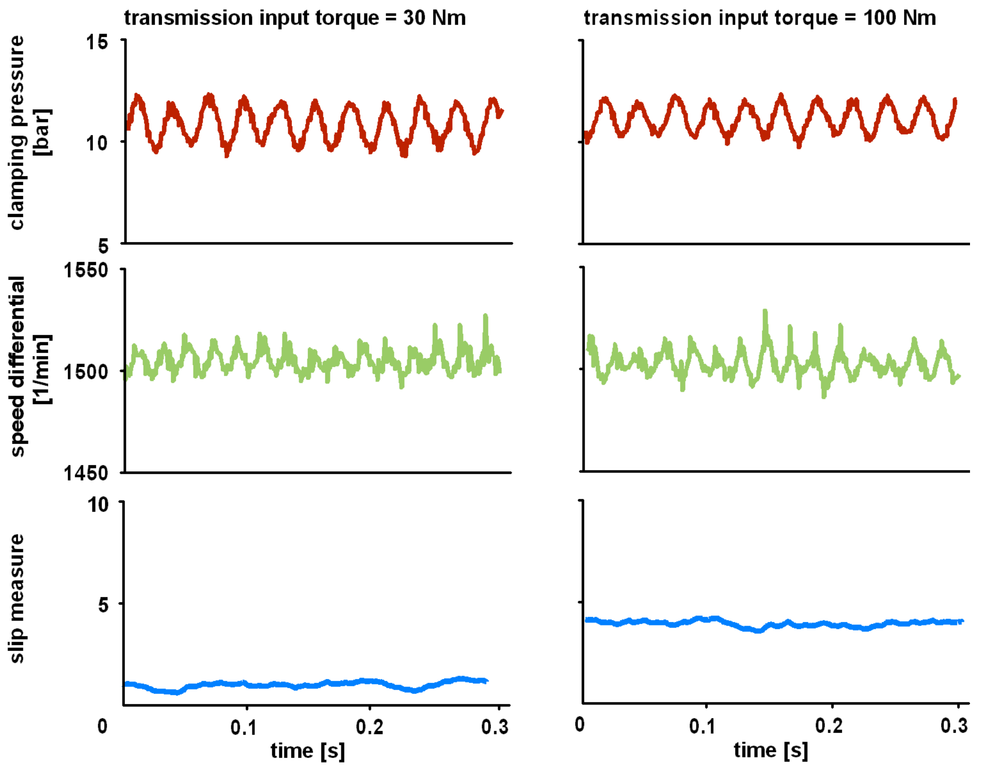

Fig. 11: *Lock-In Evaluation of Measurements*

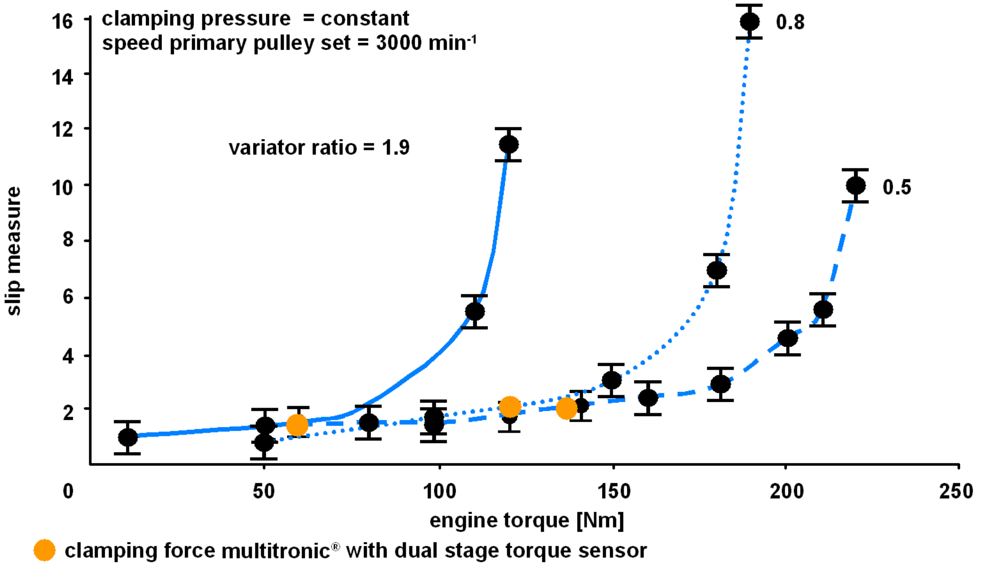

Fig. 12: *Measurements at Transmission Test Bench with Combustion Engine*

Influence on Efficiency

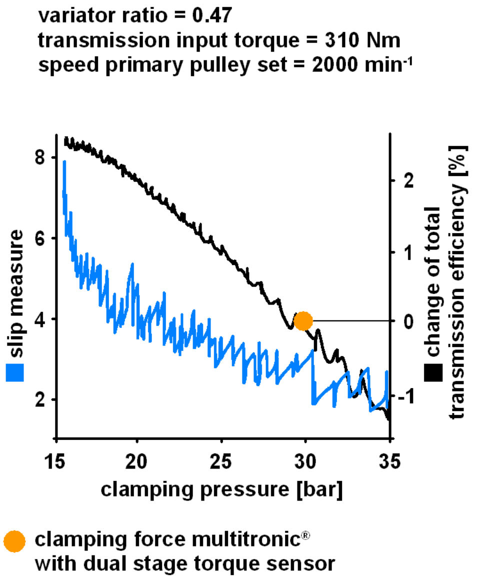

Fig. 13: *Measured Total Transmission Efficiency*

Figure 13 shows the effect of over-clamping on the total transmission efficiency.

During measurement the transmission input speed, the variator ratio and the transmission input torque were held constant.

As the clamping pressure falls, the efficiency, based on the clamping pressure of the multitronic®, initially increases. Despite an increase in the variator slip, the total transmission efficiency increases. This is both due to reducing the hydraulic losses through reduced line pressure and also due to lowering the variator losses occurring through the reduced over-clamping.

On further reduction of the clamping force, the total transmission efficiency increases only slightly or remains constant. The slip losses are no longer over-compensated for by the power gain from the reduction of the over-clamping and the hydraulic losses.

The slip measure, calculated according to the lock-in process, continually increases with decreasing clamping pressure.

Slip control

If the slip signal is used as a control variable for a slip controller, then the possibility exists in principle to operate the transmission in the optimum efficiency range. The variator shaft loads are also lowered due to the reduced clamping force, which can for example be used for an increase in ratio spread, because smaller shaft diameters are possible.

As the clamping force requirement alters slowly, an open-loop control combined with an adaptation instead of a closed loop slip control is also conceivable.

Hydraulic concept

Controlling slip causes very high demands on the hydraulic control:

- Low hysteresis in the control sequence
- High dynamics in the pressure build-up
- High stability concerning hydraulic oscillation
- Very good system knowledge dependent on pressure and temperature, to be able to design the slip control for stability in all operating conditions
- High reproducibility of pressure build-up and release.

Figure 14 shows the section of the hydraulic control unit responsible for the clamping load system.

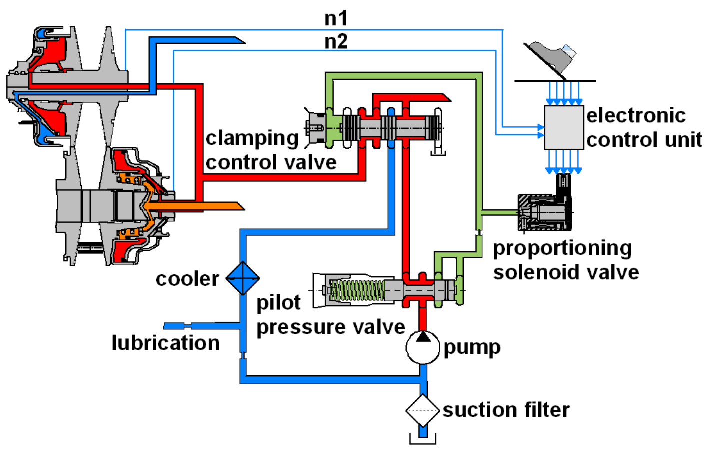

***Fig. 14:** Slip-Controlled Clamping System*

Extensive simulations and calculations have been carried out for the design of the hydraulic system. These simulations for example enable illustration of the influence of the hysteresis on the slip control (figure 15).

Both the simulations and the measurements have shown that with equal valve activation, the pressure amplitudes decrease as the valve hysteresis increases. The slip measure also is reduced simultaneously.

LuK has developed a special valve actuation in order to keep the valve hysteresis to a minimum. The hysteresis of the proportioning solenoid valve and of the clamping valve is reduced to a minimum by introducing specific pressure oscillations to the entire system via the pilot control circuit. These pressure oscillations are in a frequency range of between 70 and 100 Hz and are therefore above the modulation frequency for the slip detection. They have no influence on the pressure amplitudes of the slip control. Figure 16 shows the relationship between electrical current, pilot control pressure and clamping pressure.

The necessary form of the overlay amplitudes in the electrical current is determined by the subsequent system. As the system features non-linear characteristics, excitation amplitudes of various levels are necessary in the electrical current to achieve a constant pressure amplitude through the characteristic curve.

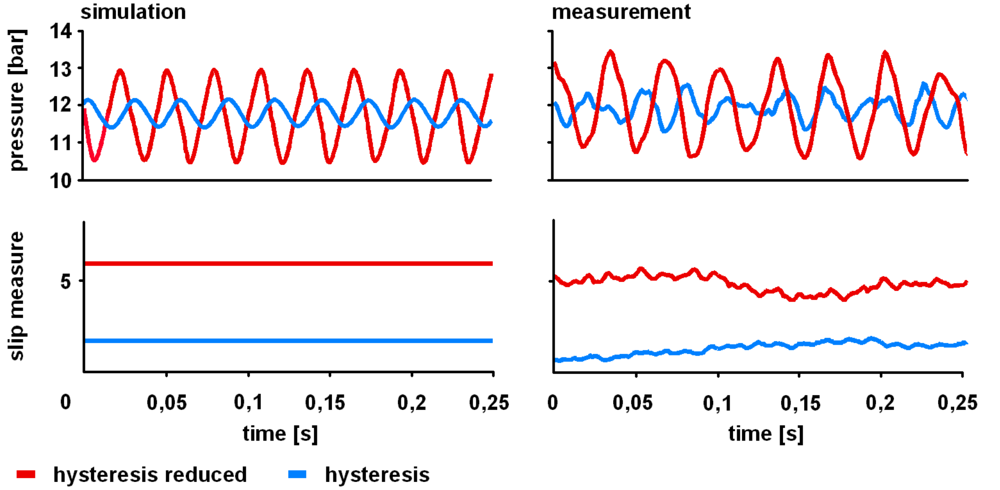

***Fig. 15:** Influence of Hysteresis on the Slip Measure*

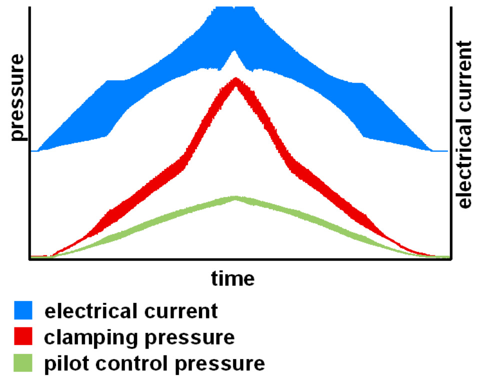

Fig. 16: LuK Valve Actuation with Overlay

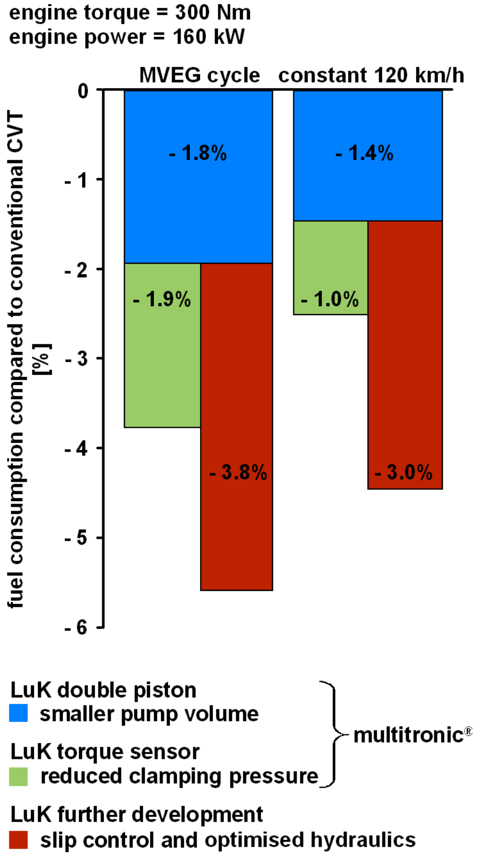

Fig. 17: Consumption Benefits

Summary

An important development target of modern transmission technology is the reduction of fuel consumption. This requires measures to further increase the transmission efficiency. The multitronic® currently in production already presents a very good standard with regard to efficiency and fuel consumption, as shown in figure 17.

Through concentrated further development of the hydraulic control in conjunction with new ideas for realising a slip control, there is undoubtedly further potential to reduce fuel consumption.

Indeed, the initial measurements indicate that the potential to further increase the efficiency exists and is achievable.

References

[1] Nowatschin, K.; Fleischmann, H.-P.; Gleich, Th.; Franzen, P.; Hommes, G.; Faust, H.; Friedmann, O.; Wild, H.: multitronic® – Das neue Automatikgetriebe von Audi, ATZ 102 (2000), no. 7/8, p. 548 - 553, no. 9, p. 746 - 753, [multitronic® – The New Automatic Transmission from Audi, ATZ worldwide, no.7/8, p. 25 - 27, no. 9, p. 29 - 31.]

[2] Faust, H.; Linnenbrügger, A.: CVT Development at LuK, 6th LuK Symposium 1998.

[3] Englisch, A.; Faust, H; Friedmann, O.: Innovative System for Clamping and Adjusting of a Chain Variator; Proceedings of the Global Powertrain Congress (GPC), Detroit (U.S.A.) 2001.

[4] Lauth, H. J.; Scholz, T.; Agner, I.; Webert, D.: Flow Demand Controllable Pumps – Reduced Power Consumption of Steering, Suspension and Transmission Systems, 7th LuK Symposium 2002.