ABSTRACT

Besides the current development trends focusing on fuel economy and emission reduction in automotive transmissions, for example through hybridization or a higher number of gears and larger gear spread, analyzing and optimizing efficiency can make a major contribution to reaching future CO₂ limits. In cooperation with IAV GmbH, one of the leading development partners to the automotive industry, KISSsoft AG presents a method for analyzing, evaluating and optimizing transmission losses.

As a general rule, ascertaining the overall efficiency of an automotive transmission starts by abstracting calculation to the level of individual components and machine elements. After determining individual losses, they are summated to calculate overall loss. Analytical calculation processes already exist for individual elements such as bearings or gear teeth. They provide sufficiently accurate results to select concepts or improve existing systems.

For the purpose of more detailed calculations, loaded contact analysis can be used for spur gears. It allows the influence of the micro-geometry to be taken into account, which is especially efficient in an integrated environment such as the KISSsoft gearwheel calculation tool.

The present paper uses an IAV 7-speed dual-clutch transmission to explain the process for efficient calculation of the transmission’s overall efficiency. The primary focus here is on the automated generation of speed- and load-dependent power loss maps which can be used in subsequent cycle simulation. The modular approach to calculating individual losses provides the capability of performing detailed analyses by loss drivers and of validating optimization measures in a simple way. Based on KISSsoft and KISSsys software, the calculation method is therefore a helpful tool employed throughout the development process from the concept phase to the production layout stage.

KISSSOFT CALCULATION SOFTWARE

KISSsoft is a modular calculation system for designing, optimizing and analyzing machine elements. The scope of the software ranges from single machine elements right through to the automatic sizing of complete gearboxes. KISSsys is a KISSsoft system add-on that enables users to model complete gear units and powertrains.

The software includes internationally recognized calculation standards and a large number of design and optimization options. A special software feature is contact analysis that makes it possible to check the contact situation of the gears in a gearbox under load. As all functions
are integrated in a single package using the same user interface, it is easy to vary parameters and get immediate feedback about any resulting changes. In addition, built-in optimizations help to find the optimum macro- and micro-geometry. Shaft calculation identifies realistic, load-dependent gear misalignments. This process also takes nonlinear bearing stiffness into account.

In response to an increasing number of requests for calculations addressing the efficiency of powertrains, several methods have been implemented over the last years for analyzing the power losses of machine elements. This provides a sound basis for evaluating complete systems of machine elements, for instance gearboxes.

Combining IAV’s experience with calculation tools from KISSsoft resulted in a toolbox that can be used for practical cases. The calculations include various types of gear drives, such as helical gears, planetary gears, bevel gears, and worm gears that cover all possible gear train configurations. This power loss template is best used with a special variant generator that automatically calculates any number of design variants that take account of specific requirements in gear shift transmissions and provide a specific evaluation. This tool allows the user to select the best possible drive from among numerous alternatives according to criteria such as smallest size, highest strength and, of course, efficiency. In addition, existing gearboxes can be evaluated to see if performance and efficiency can be optimized without compromising on drivability or performance, bringing the experience from former designs to the new tool.

**Dual-Clutch Transmission IAV 7DCT280**

The method for systematically analyzing and optimizing transmission power losses is shown using the example of the seven-speed dual-clutch transmission IAV 7DCT280. This is a structurally optimized modular transmission in front-transverse design for compact and mid-size vehicles. Compared with current six and seven-speed transmissions, the mechanical components can be reduced to just one main shaft and one countershaft to provide the seven forward speeds. The package for the otherwise conventional second countershaft can be used on a modular basis to integrate an electric traction motor for various hybrid functions or the mechanical components of a classic reverse speed (Figure 1). [1]

In the hybrid variant, the electric motor is linked to the differential gear via a second output stage (Figure 1, left). A planetary gear set serves as a booster stage for the electric motor, giving the hybrid system high levels of wheel torque even at low vehicle speeds, and also helping to exploit better efficiency ranges. Synchronizer H provides the optional capability of disengaging the electric motor from the rest of the powertrain to protect the motor at higher vehicle speeds or in the event of a fault in the electric system. In the conventional transmission variant, these elements can be carried over, supporting the commonality concept that is aimed for. The mechanical reverse speed is provided merely by replacing the electric motor with a set of spur gears connecting the main shaft’s center gearwheel with the sun gear shaft of the planetary transmission (Figure 1, right).

Figure 2 shows the shift logic as well as the transmission’s overall gear ratios. The ratio series shows good stepping with a drive-off ratio of 15 and overall spacing of 7.5. With the standard ratio in the planetary gear set at -2.8, the reverse speed’s ratio of -14.9 is at a similar level to that of the first forward speed. In this configuration, the electric motor’s ratio to
the vehicle’s wheels is 11.5. This transmission structure is also shown to provide a high level of flexibility in its ratios, gear steps and spacing in return for slight modifications to the gear-set ratio steps. [1]

Further investigations regarding power losses will be done on the basis of the conventional variant with reverse speed and without hybrid functions.

![Hybrid transmission](image1)

**Figure 1:** Modular transmission system of the 7-speed hybrid DCT (left) and the 7-speed DCT with mechanical reverse speed (right)

<table>
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<tr>
<th>Speed</th>
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<th>Ratio</th>
<th>Step</th>
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**Figure 2:** Shift logic and exemplary ratios of the 7-speed dual clutch transmission

**MODEL SETUP AND POWER LOSS MAPS**

A calculation model under KISSsys (Figure 3, right) is used to compute transmission power losses. At this system level, all kinematic and kinetic interactions between the individual machine elements in the overall transmission are taken into consideration. This means that the most important prerequisite for automated generation of gear-dependent power loss maps is already fulfilled. In addition, the KISSsys model is complemented to include the loss modules. As in many cases a KISSsys model is already available for dimensioning of the machine elements, power losses can be calculated without major extra modeling work.
Figure 3: Considered power losses (left) and KISSsys calculation model (right)

The analysis claims to be able to calculate the overall transmission efficiency in an integrated way. This is why all relevant loss sources are factored in. These are losses produced by gear meshing, churning and ventilation, drag torque at the synchronizer units and disengaged multi-disk clutches, by radial shaft sealing rings, rotary unions, bearings as well as by the oil pump's power consumption (Figure 3, left). The calculation methods to ascertain the different types of losses are based on relevant standards, dissertations or publications. Furthermore, there is great flexibility to choose different approaches for calculating individual loss sources. The load-dependent losses caused by gear meshing are calculated on the basis of the proposal presented by Niemann/Winter [2]. In addition, contact analysis can be used to define the optimum micro-geometry using profile and tooth trace modifications. A multitude of calculation methods is available for load-independent churning, squeezing and ventilation losses. The rules defined in standard ISO/TR 14179-2 [3] are used to determine the losses occurring in the injection lubrication implemented in the dual-clutch transmission. The injection volume flows at the tooth contacts are assumed to be 1 l/min for the gear pairings and 2 l/min for the constant gear ratios respectively.

In the same way as for the gear teeth, a distinction is made for load-dependent and load-independent losses caused by the bearings. These are taken into account on the basis of the information provided in the SKF bearing catalog 1994 [4]. Alternatively, the new SKF 2004 method [5] can be used, which attributes frictional components in relation to their cause. The drag torques of non-actuated synchronizer units are determined from geometry data of friction surfaces based on Newton's law of viscosity. Measurements have shown that drag torque first increases in linear fashion with rising differential speed before leveling out at a constant value. This effect can be factored in by incorporating various influencing factors such as temperature, oil volume flow or air gap and by stipulating a limit friction torque. The approach presented by Rao [6] for ascertaining fluid shear in open multi-disk clutches is used to measure drag losses in the dual-clutch transmission. As friction torque largely depends on influencing factors such as spline, surface roughness, corrugation or oil supply, the values can be adjusted to any measured values available using correction factors.
The losses caused by the radial shaft sealing rings on the transmission intake side as well as by the differential on the output side are determined with the help of calculation rule ISO/TR 14179-2 [3]. Unlike the radial shaft sealing rings, friction torque on the rotary unions for actuating the clutches is load-dependent. With intake torque rising, actuation pressure and hence contact force acting on the piston rings increases.

The power consumption of the oil pump depends to a large degree on the hydraulic concept implemented. This is why it is expedient to describe the pump's power consumption with the help of a map. The map can be defined using interpolation points and saved to KISSsys. Linear interpolation serves to ascertain power consumption at individual operating points. After reading in maps up to the third dimension, proprietary analytical approaches can be defined too. The first step consists in analyzing a concept with a constant-feed pump connected on the drive side to secure actuating pressure, lubrication and cooling oil demand. Although the IAV 7DCT280 uses a more efficient system, the following section will outline the potential improvement that can be achieved starting from this basic version often used.

As the transmission power losses will be used in subsequent cycle simulation, oil temperature is set to 60°C. This corresponds to a medium temperature between the cold start of the cycle and the hot temperature at the end of the cycle run.

After incorporating all relevant losses in the computation model, automated generation of power loss maps can start for all gears. The maps are generated incrementally based on a grid and a predefined engine speed and torque range for the internal combustion engine. The loss maps are output in the form of CSV result files which can easily be used for subsequent cycle simulation, for instance. In order to break down losses into individual components, a separate export file is created for each loss source in addition to the map for the overall transmission. By way of example, Figure 4 shows the maps for the seventh gear of the dual-clutch transmission investigated here.

The maximum efficiency is 94.1% in seventh gear at low intake engine speeds and high torque. With rising speed and decreasing torque, efficiency drops, as was to be expected. On the one hand, this trend can be explained by the progressive development of speed-dependent losses, for example due to bearing- and injection-related losses as well power consumption of the oil pump. The drop in efficiency at low intake torque is attributable to load-independent losses which constitute a constant variable at a defined engine speed.

Gear meshing losses exhibit a disproportionate increase at rising torque. This is because of the underlying physical model in which the tooth contact surface increases for rising loads. Investigations into the impact of oil viscosity show that in individual cases the use of highly viscous oil can improve overall efficiency. This trend can primarily be observed when gear meshing losses prevail at very low engine speeds as all other loss factors benefit from low oil viscosity.

Synchronization, injection and sealing ring losses are independent from load. In the case of the rotary unions, the influence is a result of torque in conjunction with the related increase in pressure acting on the clutch's actuating piston.

Starting from a minimum oil pressure, a linear correlation can be observed for the pressure increase of the ATF pump in relation to intake torque. As the constant-feed pump is coupled with the intake transmission shaft, the volume flow delivered increases in line with engine speed. In order to provide the dual clutch with sufficient cooling oil for driving off, the displacement volume is laid out for a delivery volume of approx. 20 l/min at 1200 rpm.
Figure 4: Power losses and efficiency maps of IAV 7DCT280 in 7th gear at 60°C oil temperature

CYCLE SIMULATION

The power loss maps shown in Figure 4 already permit individual losses to be stated for defined operating points. In order to evaluate the overall loss distribution in the consumption-relevant part of the NEDC, cycle simulation is required. If cycle simulation identifies loss drivers, it is possible to evaluate improvement measures directly in the cycle. This way, decisions can be taken straight away on the basis of CO₂ reduction predictions.

For the purpose of loss analysis, the IAV 7DCT280 is installed in a medium-class vehicle. The vehicle is driven by a gasoline engine delivering 140 kW output and a maximum torque of 265 Nm. Figure 5 summarizes the basic vehicle specification. Fuel consumption largely depends on the shift strategy. An economical way of shifting is selected for simulation. The
engine is consistently operated in the gear that provides the lowest fuel consumption in the combustion map, considering the transmission losses. Figure 5 shows the resulting operating points in the combustion map. This operating range is found at low engine speeds and higher torques.

Figure 5: Vehicle data of NEDC simulation and operating points in the engine map

Figure 6, left shows the distribution of summated energy losses after running a NEDC simulation. With a share of more than one third, the ATF oil pump accounts for the single largest part of the overall losses. This trend can be explained by the oil volume dimensioned to deliver sufficient oil supply for the drive-off engine speed. At rising engine speeds, we observe that the constant-feed pump delivers too much oil. At 19.0%, bearing-related losses are more or less in the same order of magnitude as gear meshing losses, which account for 18.1%. Load-independent squeezing losses at the gear teeth resulting from injection lubrication are shown to be 4.1%. Comparing drag losses between the clutches and the synchronizers, we see that synchronizing losses prevail. This distribution can be attributed to the fact that no gear is pre-engaged in simulation. In order to define the missing kinematic degree of freedom, both clutches need to have the same speed. This boundary condition is based on the assumption that the speed of the disengaged clutch tends to approach the speed of the engaged clutch due to drag torque. Only in gears 1 and 7, a relative engine speed prevails in the disengaged clutch because of the synchronizers being actuated repeatedly (Figure 2). The sealing-related losses can be divided as follows: 4.6 % for the radial shaft sealing rings and 6.4% for the rotary unions.

Figure 6: Distribution of energy losses in the NEDC over components and speeds
Figure 7: Distribution of average power losses and efficiency in the NEDC over speeds

Besides evaluating the individual losses, gear-dependent energy distribution is of interest (Figure 6, right). The main part results from the seventh gear. This trend can be primarily explained by the high time share of 41.6% that the seventh gear has in the NEDC.

Figure 7 shows the gear-dependent efficiency, mean power requirement and mean power loss. As well as can be expected, efficiency is poor in lower gears because of the high transmission ratio and low power requirement. The loss distribution in seventh gear reveals that the increase in gear meshing losses caused by two additional meshing processes in the winding path plays a minor role. However, an increase in speed-dependent losses at the bearings and synchronizers is worth mentioning. They result from the comparatively high gearwheel speed which correlates with the rising vehicle speed in seventh gear. The mean overall transmission efficiency based on the NEDC is 92.9%.

OPTIMIZATION MEASURES

The distribution of transmission losses in Figure 6 shows that the actuating concept, the gear teeth, bearings and synchronizers harbor the greatest optimization potential for reducing CO₂ emission. Hence, the following section defines various measures in a step-by-step approach and evaluates their effect in the NEDC.

The hydraulic concept with an ATF constant-feed pump analyzed before is not designed in relation to actual demand. A reviewed hydraulics concept uses an electrically driven pump to supply cooling and lubricating oil. In addition, consideration is given to electromechanical actuation of the clutches and gear actuators. This combination permits demand-controlled actuation and cooling oil supply in relation to the torque prevailing or the output transmitted respectively. Because of the operating principle, no rotary unions are required in the case of electromechanical clutch actuation as they are replaced by engagement bearings.

With a view to reducing gear meshing losses, the aim is to modify the design philosophy. The basic design exhibits a distribution between contact and overlap ratio of \( \varepsilon_\alpha = 2 \) and \( \varepsilon_\beta = 1 \) in order to meet the acoustic requirements. As the power loss mainly depends on the contact ratio, a new design with a distribution of \( \varepsilon_\alpha = 1.5 \) and \( \varepsilon_\beta = 1.5 \) is produced. With the same overlap ratio, a noticeable increase in efficiency is to be expected at the cost of acceptable acoustic disadvantages.
The next step in reducing the gear meshing losses consists in elaborating an alternative distribution between gearwheel transmission ratios and final drive transmission ratio. Using the transmission variant generator in KISSsys, a solution with lower final drive transmission ratio and hence higher gearwheel ratio is identified for the IAV 7DCT280.

In order to harness the potential for reducing bearing-related losses, a fixed/free bearing based on an adjusted bearing arrangement of the differential with taper roller bearings is foreseen. This arrangement consists of a deep groove ball in conjunction with a cylindrical roller bearing.

One possible measure to decrease drag losses in the non-actuated synchronizers consists in the reduction of one friction lining. In order to accommodate the resulting higher stresses, a carbon material can be used instead of the conventional sinter lining.

Figure 8 summarizes the results of the individual measures as a function of the mean power loss of the base variant operated in the NEDC. Losses caused by the oil pump are reduced considerably from 156 W to 69 W. The extra power required for electric actuation is low at 17 W. Furthermore, the rotary unions and their losses amounting to 28 W are replaced by the release bearings with their extra losses of 13 W. All in all, an improvement of 85 W can be achieved on average by substituting the constant-feed pump.

The reviewed gear teeth layout with a contact ratio of $\varepsilon_0 = 1.5$ allows gear meshing losses to be reduced by 28 W. As axial forces increase as a result of the simultaneous rise in the helix angle, the benefit is partly offset by the bearing-related losses that are 5 W higher.

Shifting the transmission ratio between the gearwheel and final drive is capable of reducing gear meshing losses by another 5 W. Furthermore, this measure has a positive impact on gearwheel speeds. The sum of bearing-related, injection-related and synchronizing losses is reduced by 11 W.

At 3 W, the benefit of the fixed/free differential bearing is minor compared to the version with taper roller bearings. The reason is the low speed on the output shaft producing comparatively low power losses.
Ultimately, the carbon synchronizer linings can make a 12 W contribution to reducing power losses in the NEDC.

If all optimization measures are implemented, the power losses can be reduced on average by 32% from 436 W to 297 W. Under these circumstances, fuel consumption is lowered by 0.11 l/100km, corresponding to a reduction in CO₂ emission of 2.5 g/km. However, a case-by-case evaluation is necessary for each application to assess whether the reduction in power loss warrants the potential extra costs caused by the optimization measures.

**Optimizing micro-geometry**

Above and beyond the optimization measures described above, the gear meshing losses can be further reduced by appropriate gear teeth modifications. This measure is carried out for each gear wheel pairing independently from the measures implemented so far. The load-dependent contact analysis in KISSsoft is used for optimization. Among others, the analysis ascertains friction torque at each meshing position based on contact simulation. As local losses depend on sliding speed and friction force, profile modifications and a resulting redistribution of meshing forces can reduce friction losses.

Figure 9 describes the general approach. The curves for absolute and specific sliding show that the maximum values can be found at the start and end of gear meshing respectively. The normal force curve juxtaposes the results obtained with the unmodified and optimized teeth geometry. A comparison of the corresponding power losses reveals the considerably lower sum in the form of the area below the curve showing the optimized gear teeth.

![Figure 9: Sliding speed, local power loss and normal force](image-url)
KISSsoft provides a specific layout function allowing the optimum correction to be identified. This function varies modifications within predefined limits, ascertains all possible combinations of up to three sets of modifications and carries out a contact analysis for each variant. At the same time it is possible to vary load as contact behavior strongly depends on load. The results are visualized in clear form in radar charts, allowing several parameters to be reviewed simultaneously. This enables the engineer to determine the optimum combination of modifications for the case at hand. Figure 10 shows two radar charts, the left-hand one depicting efficiency, the right-hand one the transmission error. Here, variant 3:2 would constitute a good compromise exhibiting high efficiency (99.6% instead of 99.06%) and moderate transmission error.

![Radar charts for efficiency (left) and transmission error (right)](image)

**Figure 10:** Radar charts for efficiency (left) and transmission error (right)

**SUMMARY**

In cooperation with IAV GmbH, KISSsoft AG has developed a method for analyzing and evaluating transmission losses. Using the 7-speed IAV 7DCT280 dual-clutch transmission, this paper explains the efficient calculation, evaluation and optimization of transmission losses.

Calculation of transmission power losses is based on a KISSsys model. On this system level, all kinematic and kinetic interactions between the individual machine elements are taken into consideration. Hence, the essential prerequisite for automated generation of gear-dependent power loss maps is fulfilled.

The maps of all relevant loss sources are used for subsequent cycle simulation on the basis of the NEDC. A breakdown of individual losses allows loss drivers to be identified systematically. Alternatives are suggested for selected components and then evaluated in the cycle. Furthermore, contact analysis provides the capability of determining appropriate profile modifications in order to increase gear meshing efficiency in a targeted way.

The process chain presented in this paper is suitable for both efficient evaluation of different transmission concepts and detailed optimization studies on existing transmissions. The calculation method based on KISSsoft and KISSsys is a helpful tool which is employed in the entire development process from the concept phase right through to the production layout stage.
REFERENCES


