Internal traineeship

Automation of VW transmission 02k-DNZ

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1 Introduction

This report regards my internal traineeship, the goal of the report is to look at the possibilities for automation of a manual transmission from Volkswagen, indicated by VW02k-DNZ. In this report will be studied if existing systems for automating a manual transmission are applicable to the Volkswagen gearbox. These systems are implemented in, for example the Alfa Romeo 147 and BMW M3, but also in rather small cars like the Opel Corsa and the Citroen C3. These existing systems will be examined and then one system will be chosen based on the following criteria; performance, cost and complexity. Because of the length of the report only the specific details of the chosen system will be included.
Then the specifications the gearbox has to comply with are presented and in the following chapters the shift time of the gearbox will be estimated. In the following chapters the performance of the gearbox and actuating systems will be determined by solving the torque equilibrium on the gearbox masses. Finally the performance of the gearbox and the actuation system will be compared with the desired specifications and some conclusions and recommendations will be made.

2 Choosing an actuation system

Now one of the before mentioned systems will be chosen, based on the following criteria:

- Performance
- Comfort
- Cost
- Packaging

2.1 Performance

The performance of a system is represented by its shift times. The faster a system can shift, the better its performance. However manufacturers are always vague under which conditions the given shift times are valid. This makes it difficult to predict how they will perform in a different gearbox and different conditions, which can be less favorable. The following table with shift times is therefore no more than a rough indication

Gearbox (car)	Min. shift time
BMW SMG II (M3 E46)	80 ms
Ferrari F1 (Maserati 4200GT)	80 ms
Ferrari F1 (360 F1)	150 ms
Ferrari F1 (Enzo)	150 ms
Bugatti Veyron (proposed)	200 ms
Ferrari F1 (575M)	220 ms
BMW SMG (M3 E36)	220 ms
Aston Martin Vanquish	250 ms
BMW SSG (3-series)	250ms (150ms for 1st to 2nd)
Alfa Selespeed (156 Selespeed) (old)	700 ms

Table 1: Shift times specified by manufacturer

In this table only hydraulically actuated systems are shown. In the following figure a comparison concerning an unknown hydraulically operated gearbox and the electromechanically operated gearbox “Easytronic” from Opel.

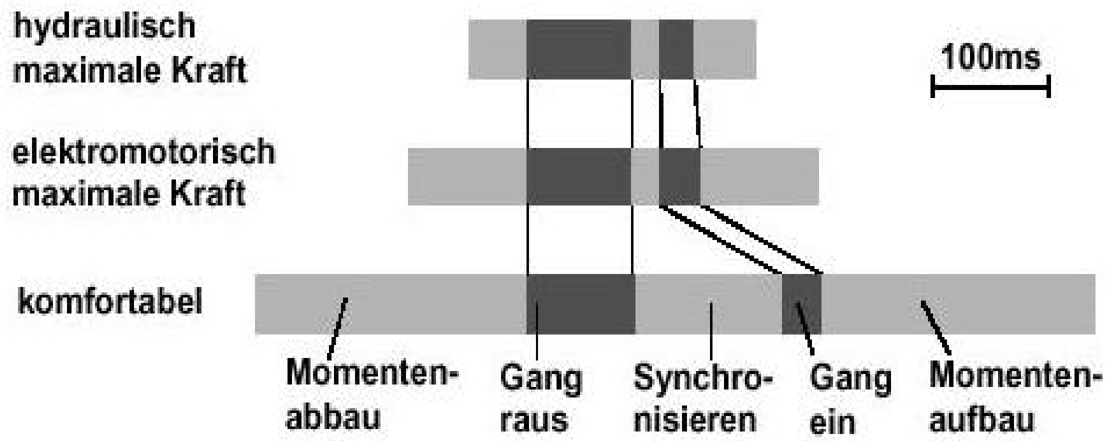


Figure 1: Shift time comparison hydraulic versus electromechanical

According to Figure 1, the electromechanical actuated systems are slower than the hydraulically actuated ones. The source of this picture doesn't mentions which hydraulic system is used in this comparison.
It can be concluded that for performance a hydraulic system is preferred and looking at Table 1 the fastest system is the SMG II implemented in the BMW, or the one the Ferrari F1 implemented in the Maserati 4200 GT. These systems are obviously shift the quickest.

2.2 Comfort

This is a very subjective parameter, but nevertheless it cannot be ignored. Since I have not been able to test myself, I have based my conclusions on the available information, which was sometimes provided by the manufacturer.
The BMW SMG II system, which had the best performance as we have seen, is also very uncomfortable. Due to the fast gear changes, the torque is not decreased gradually, but it is suddenly interrupted. This can be recognized by an excessive nodding movement of the heads of the people in the car and it is not comfortable.
Looking at Figure 1 again, we can see that especially the decrease of the transmitted torque is essential in the determination if the gear shift is experienced as comfortable. However if a gearshift takes too long, the driver experiences it as irresponsive and annoying. All of the automated manual gearboxes mentioned earlier have different characteristics varying from a sportive program to a comfortable or normal shift program. From Figure 5 we can see that an electromechanical actuated system shifts fast enough to comply with a comfortable gear change.

2.3 Cost

The used components in the automated manual gearboxes especially determine the price of such a system. The disadvantage of a hydraulic system is that it has quite a lot of components compared to an electromechanical one.

2.4 Packaging

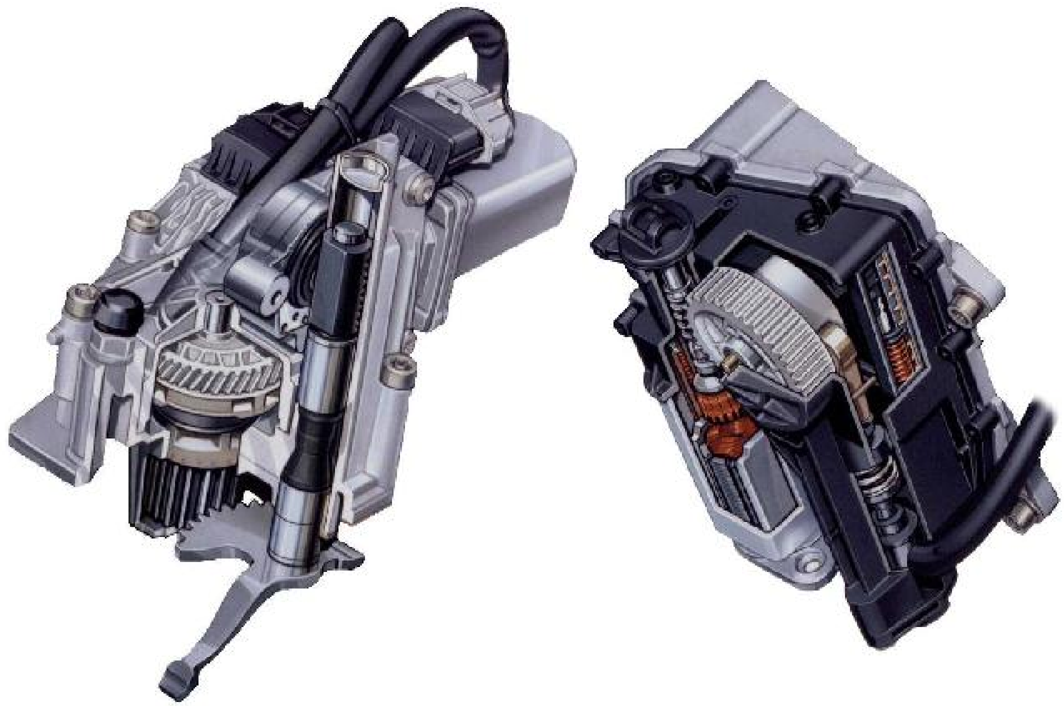


Figure 2: Left; shift and selector actuator, Right; clutch actuator

There are only two modules that have to be added to the gearbox and they are shown in Figure 2. These modules contain all the needed sensors, actuators and even the control unit, which is integrated with the clutch actuator as shown right in the figure above. All sensors which are not integrated in these modules but are required for the controls are already present in a manual gearbox. The control unit needs some of the signals from these standard sensors, so the control unit will have to be able to receive data from them.

2.5 Conclusion

An electromechanically actuated system is preferred mainly because its low cost and its compactness. It is also simple because it is a completely dry system, it cannot leak and eliminates difficult sealing problems.
Another reason for choosing the Easytronic system is that it is well documented compared to the Sensodrive solution of Citroen-Getrag, which is a comparable solution. A possible explanation why information concerning the Sensodrive system is rather scarce is that it is introduced recently, whereas the Easytronic system was introduced in 2000.

3 Opel Corsa – Easytronic

3.1 Reason for automation

Opel automated a manual transmission of the Corsa because they wanted to offer their costumers an extra bit of comfort. The actuators are powered by two electro motors. This is done because a system driven by two electro motors is a low cost solution compared with an hydraulic actuated system. The following parts are superfluous when an electromechanical system is compared with an electro-hydraulic actuation system; pump, accumulator and solenoids, this makes it less expensive. The extra sensors required for an automated system are integrated in the actuation modules, which makes it a very compact and simple add-on system.

3.2 Implemented modifications

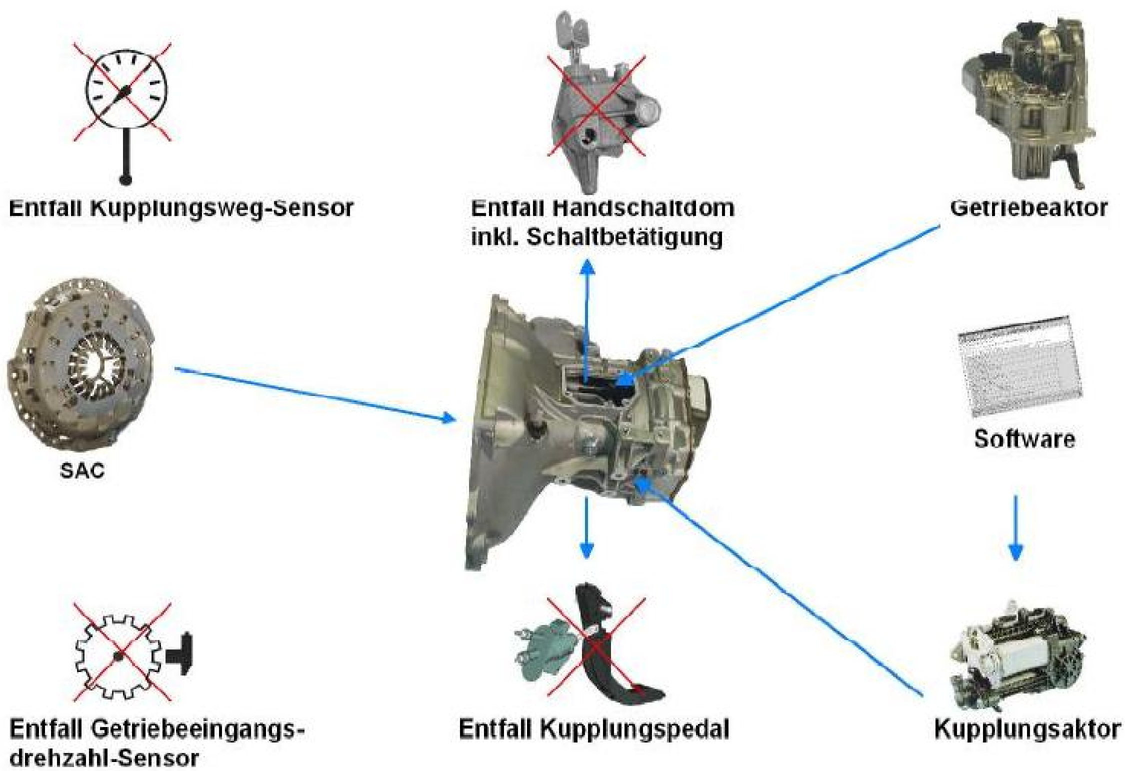


Figure 3: System components

In the figure above the modifications made to the gearbox when compared to the standard manual gearbox are graphically shown as well as the sensors that should be added when a hydraulic system was chosen. Opel removed the clutch pedal and there is no need of a sensor measuring the number of revolutions of the ingoing axle of the gearbox, or an extra clutch position sensor. The latter sensor is integrated into the clutch module.
However a different clutch is mounted, a so called Self Adjusting Clutch, from now on referred to as SAC. The advantage of such a clutch is that it compensated for wear. This results in a constant force during its life-cycle and makes it easier to control. Another advantage of the SAC is it requires less force to open. This can be seen in Figure 5.

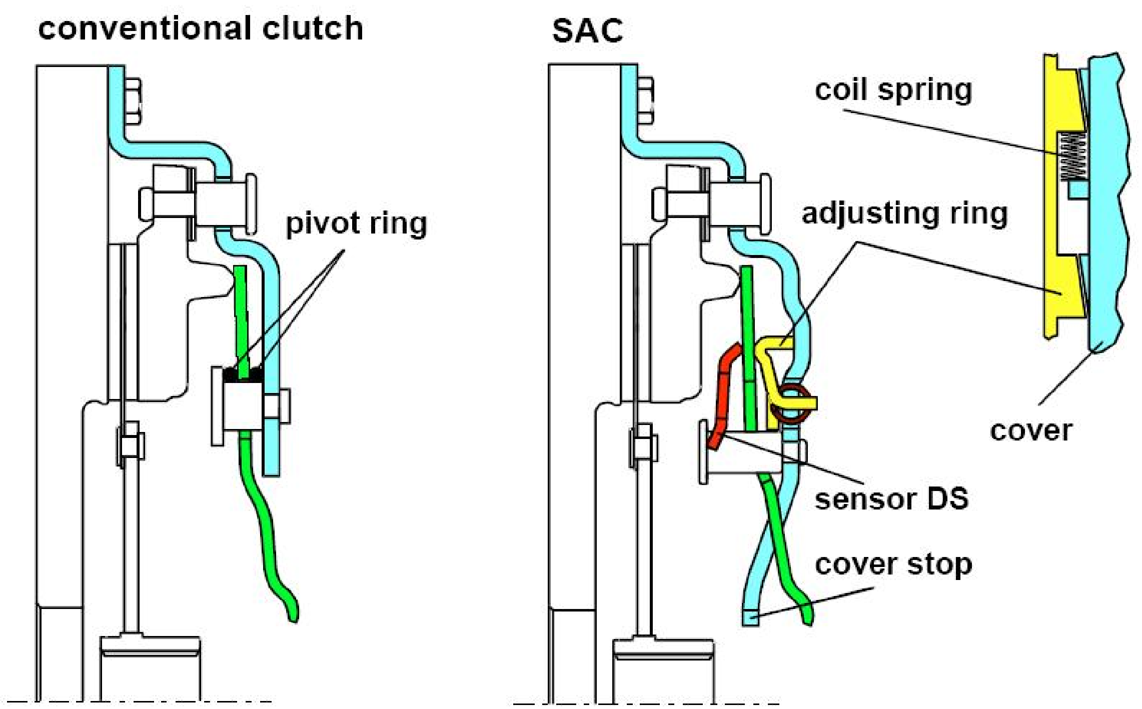


Figure 4: Self adjusting clutch working principle

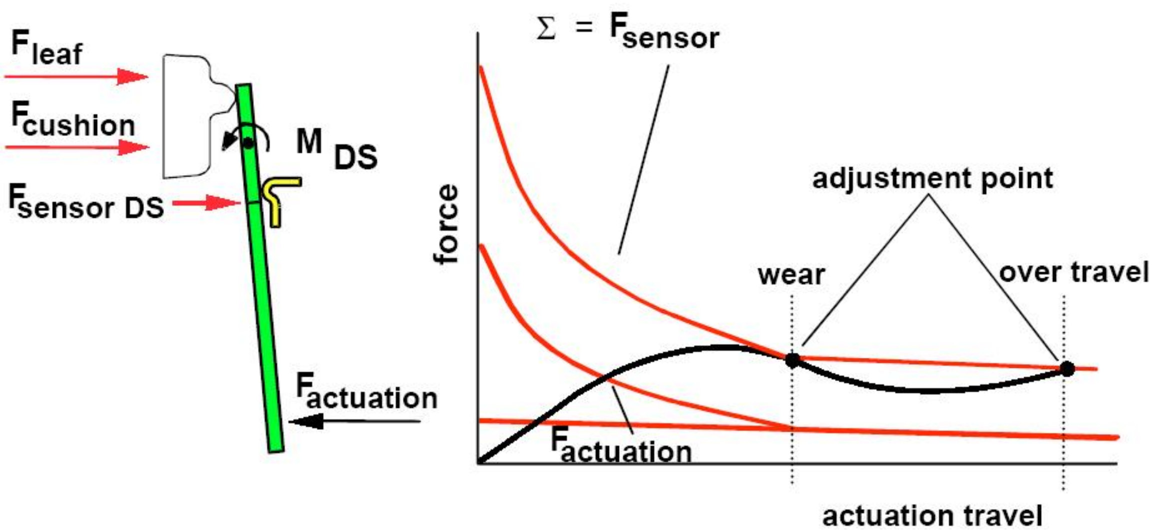


Figure 5: Clutch actuation force

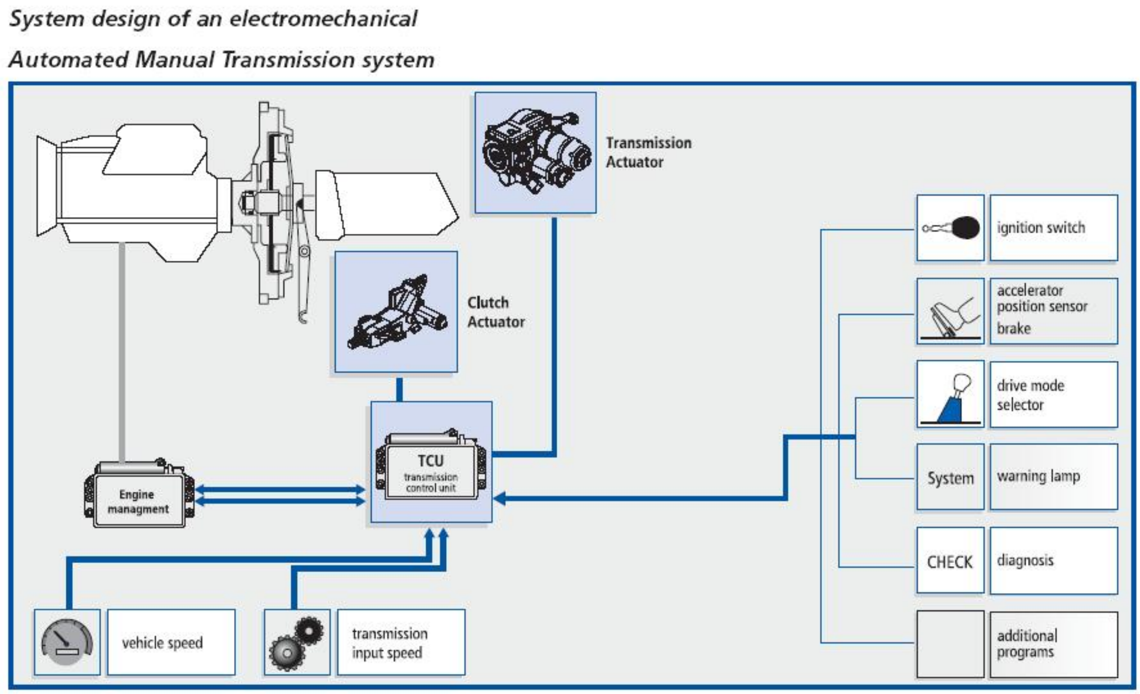


Figure 6: Block scheme of the easytronic system

In the block scheme in Figure 6 a schematic representation of the connections between the different components of the Easytronic system is shown.

3.2.1 Shift and selector actuator

The actuators of the system are working electromechanically as mentioned before. A disadvantage of such a system is its lower force density compared to a hydraulic system. By means of some innovative features this system can shift almost as fast. One of these features is a so called build in shift-elasticity. The advantages according to LuK are;

- Minimized free flight phases
- No stop of electro motor while synchronizing
- Constant shifting comfort
- Protection of transmission and actuators

The free flight phase is a part of the shift action where power is only needed to move the shifting rods. No forces other then friction and inertia have to be overcome. The electro motor does not have to stop and wait for synchronization because of the shift elasticity; however the motor runs at reduced speed until the gears are synchronized. An example of how the control signal could look like can be seen in Figure 10. The shift rod cannot move further until the rotating masses rotate at the same speed, however a force exerted on the shift rod is need to push the gear against each other. The increase in load practiced by the electro motors as they continue to rotate is represented in Figure 7. Because the electromotor did not have to stop it reaches its top speed again in the final phase of the gear shift so the gears will be locked faster.

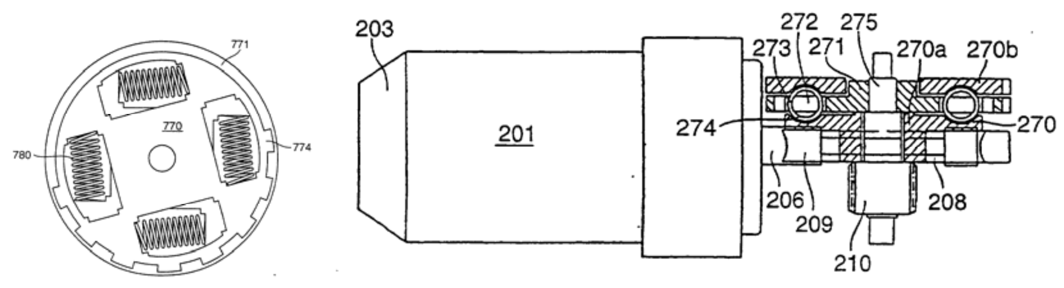


Figure 7: Left; effect shift elasticity, Right; implementation of shift elasticity

3.2.2 Actuation lay-out

The lay-out of the actuation part is shown in the next figure

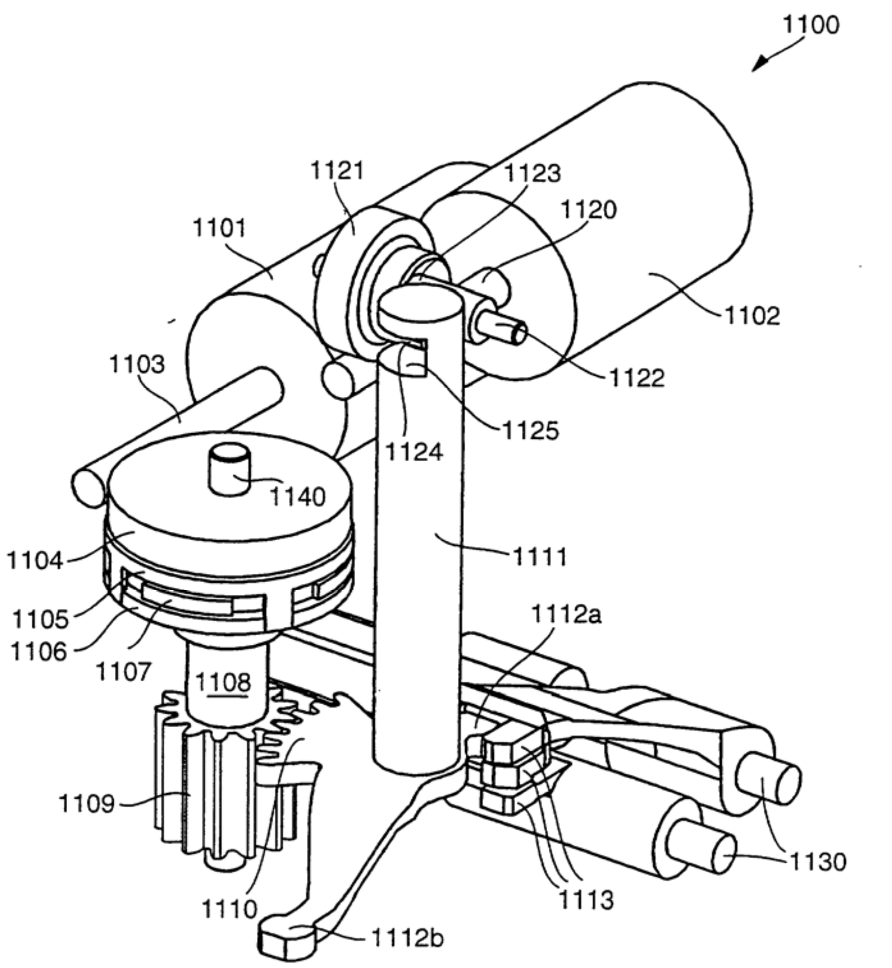


Figure 8: Lay-out actuation part

In this figure the lay-out is orderly represented and the components are easy to recognize. The component indicated with the numbers 1104 to 1107 are the implementation of the shift elasticity. Part 1104 is connected to shaft 1140 by means of the springs required for the shift elasticity, a detailed representation is given in Figure 7, so it is not connected rigidly to this shift. Worm wheels are fitted at the shafts (1103 and 1120) of the electro motors. Shaft 1111 performs the selection of the desired gear. At the top of this shaft a groove can be found, the shift finger fits into this groove and converts a rotation into a translation of shaft 1111. Parts 1112a and 1112b are the shift fingers, where 1112a operates the shift rods 1130 and part 1112b may operate for example a reverse gear.

3.2.3 Clutch actuator

The clutch actuator is also operated by an electromotor. This actuator is also used in the Mercedes A-class, however the shift and selection actuator are not applied in this model. LuK developed these modules and Bosch supplied the electro motors. The motors are based on motors used to actuate door windows, and are developed further to meet the requirements for gearbox operation. The actuator consists of an electromotor, worm wheel, a gear wheel with cam and a piston plunger. The rotation of the electromotor is translated in a translation of the plunger, generating an oil flow and ultimately in the disengagement of the clutch. The used worm is self-locking, so no power is needed to maintain a certain position. According to publications from LuK the clutch actuator uses less then 10 W.

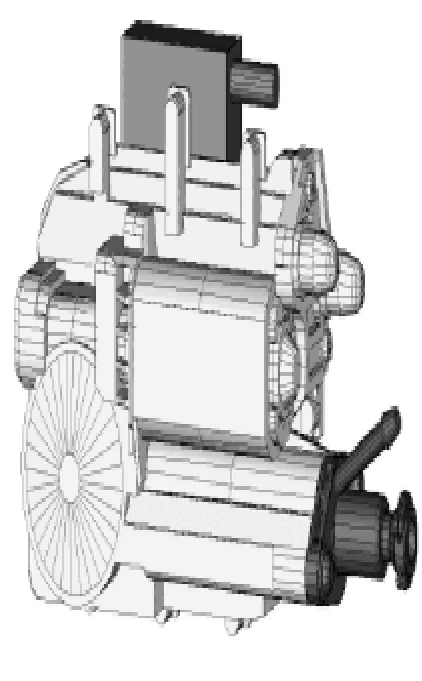


Figure 9: Clutch actuator

The most important components in Figure 9 will be discussed here. Component 101 represents the electro motor connected to the worm with number 112 by means of a shaft with number 102. This worm rotates gearwheel 113, the bearing of this gearwheel are indicated by number 114. A cam is mounted on gearwheel 113 and this cam translates the rotation of gearwheel 113 in a translation of the plunger 116. The oil moved by this plunger is situated in chamber 121 which is connected by tube 122 to a piston 123 that operates the clutch.

3.2.4 Easytronic control signal

From patent GB2313886 an example for a possible control signal is obtained. This control signal is shown in Figure 10. More information concerning the control of the Easytronic actuation system can be found in this patent.

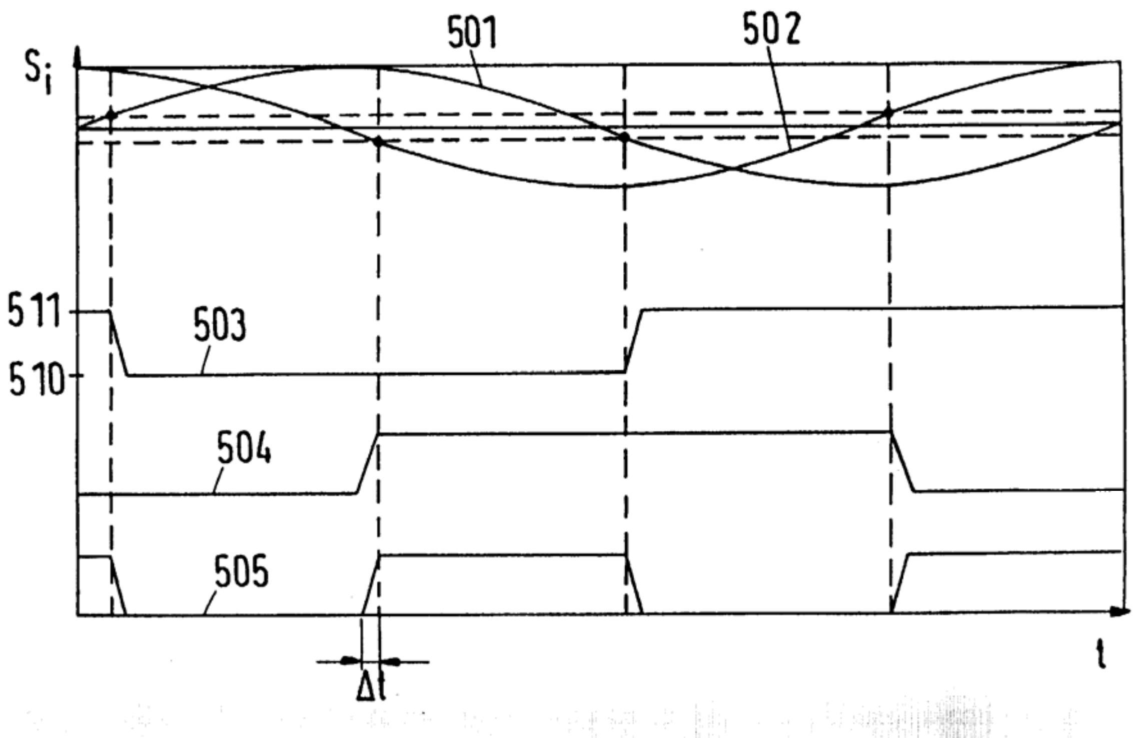


Figure 10: Easytronic control signal

4 Transmission 02k-DNZ overview

4.1 Transmission lay-out

In this chapter the transmission 02k-DNZ, represented in Figure 11, will be introduced.

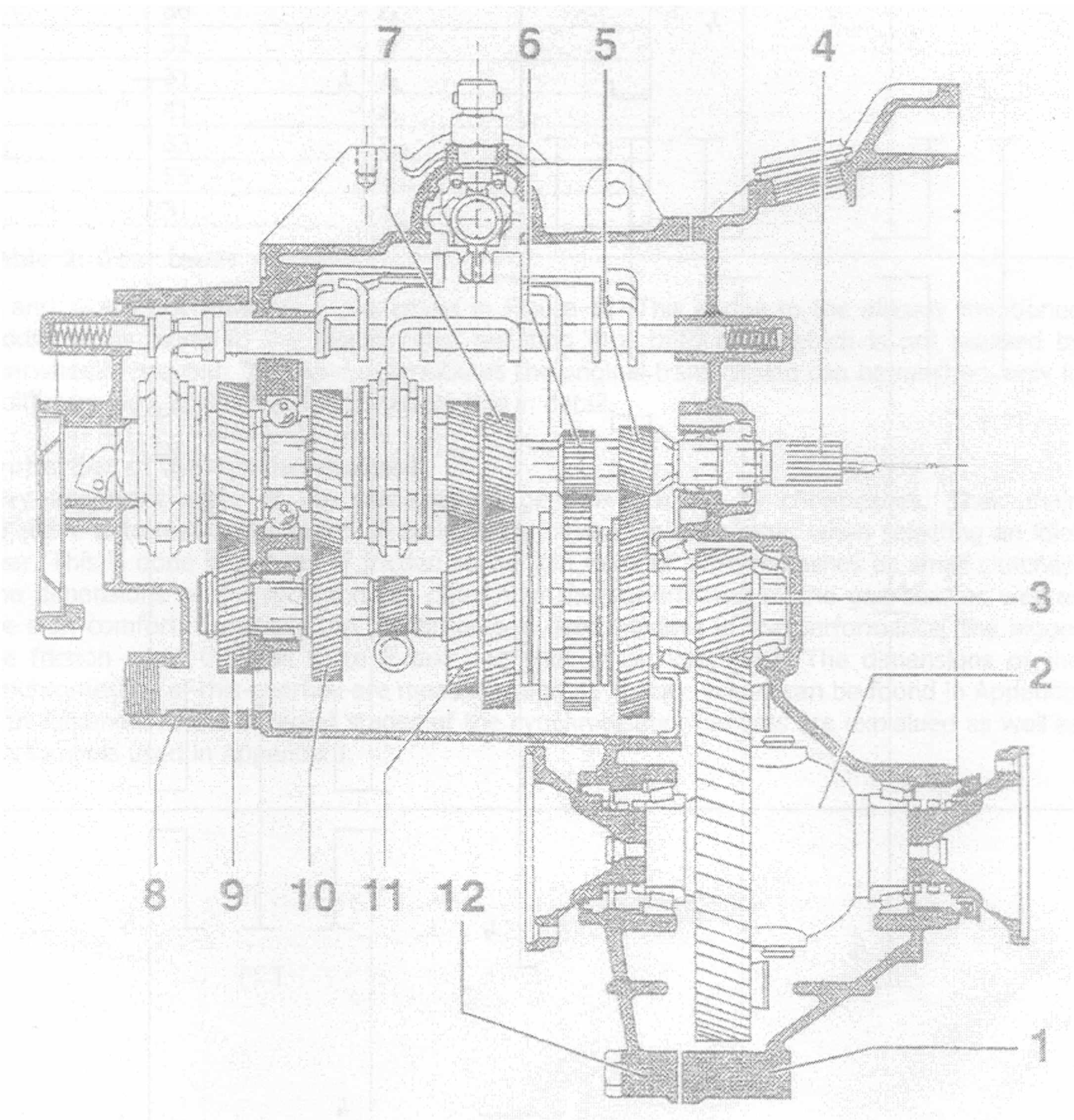


Figure 11: Cross-section view of gearbox 02k-DNZ

The gearbox shown above is used in the Volkswagen Golf and Bora as well as in the audi A3. Originally it was designed as a four-speed transmission, but a fifth gear was added afterwards. This is obvious when we take a look of the cross-section view above. The gearwheels of the fifth gear are added onto the original casing and an extra lid was added to cover the extra gearwheels and synchromesh.

In the following figure a schematic drawing of the gearbox is presented. Here it is more obvious which gearwheel is belongs to which gear. However a change is made compared to Figure 11. The third gear is replaced by a sixth gear, designed as an overdrive. So the schematic figure is not the same as the cross-section view in Figure 11.

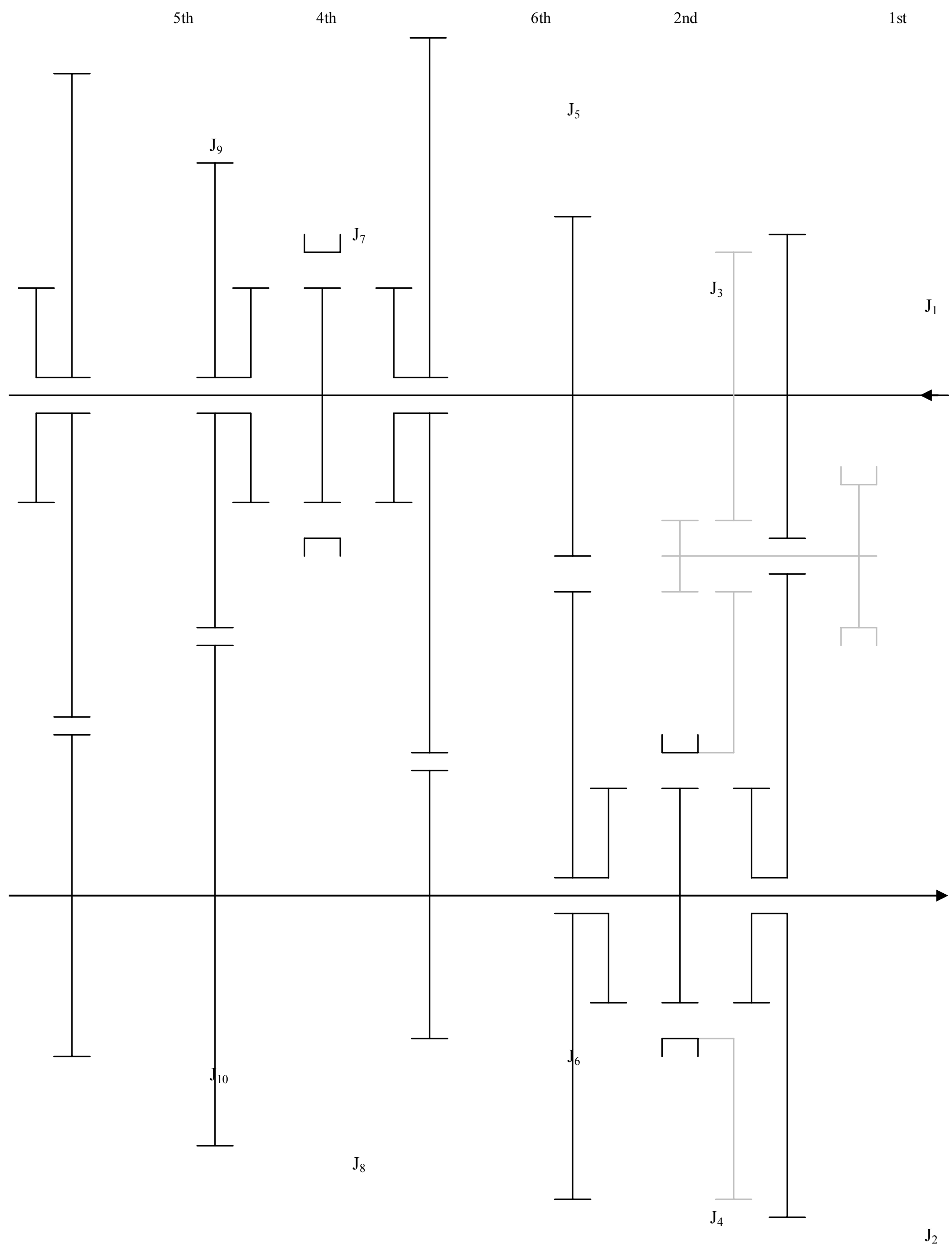


Figure 12: 02k-DNZ schematic

In the figure above the symbols used to indicate the inertia's are given. The indices will be used from now on to refer to the specified gearwheel. So gearwheel i is indicated by index i and the inertia belonging to the gearwheel will be represented as J_i.

The number of teeth on each gearwheel is given in the service manual of Volkswagen and are summarized in Table 2.

Gear wheel	Number of teeth	Symbol indicating number of teeth
J_1	11	z_1
J_2	38	z_2
J_3	18	z_3
J_4	35	z_4
-	28	z_5
-	36	z_6
J_7	32	z_7
J_8	31	z_8
J_9	41	z_9
J_{10}	33	z_{10}
J_5	55	z_{11}
J_6	31	z_{12}

Table 2: Gear teeth

z_5 and z_6 are not linked to a gearwheel in Figure 12. This is due to the already mentioned modifications made to the transmission with respect to the third gear. This gear will not be realized by gearwheels but in combination with a planetary gear set. However the same gear ratio as the original third gear is used, only in a different way. It takes to far to explain this in detail.

4.2 Properties of the synchromeshes

Very important parts of the transmission are the synchromeshes. Their main objective is to match the speeds of two rotating masses to each other. This is done by means of friction, so we can think of synchromeshes as small clutches. The dimensions of the synchromesh determine the performance of the gearbox, as well as the shift comfort. Especially the friction area is crucial for the performance, the bigger the friction area, the lower the shift force. The dimensions of the synchromeshes of this transmission are measured and its specific values can be found in Appendix H. In Appendix A. the different stages of the synchronization process are explained as well as the symbols used in Appendix H..

5 Required specifications

5.1 Time available

The required specifications of the actuation system are given in Appendix G. here a complete overview of the shift actions will be given.

Gearshift	Vehicle speed [km/h]	Duration [s]
1 -> 2	50	0,44
2 -> 3	90	0,93
3 -> 4	135	0,2
4 -> 5	180	0,2
5 -> 6	100	0,2

Table 3: Critical upshift times at corresponding vehicle speed

Gearshift	Vehicle speed [km/h]	Duration [s]
4 -> 1	40	1,62
6 -> 5	140	0,45
6 -> 4	80	0,47
6 -> 3	60	0,66
6 -> 2	60	1,23

Table 4: Critical downshift times at corresponding vehicle speed

From Table 3 and Table 4 we can calculate the corresponding angular velocity of the engine. This is done using the following equation:

$$\omega_{motor} = \frac{30/3{,}6}{r_{wheel}}\left(i_{diff} \cdot i_{gear}\right) \qquad \textbf{Equation 1}$$

The value of the wheel radius r_{wheel} as well as the differential ratio, i_{diff} and the gear ratio I_{gear} can be found in Appendix I. In the next section it will become clear why exactly these shifts will be discussed.

5.2 Finding the most critical shifts

First it will be shown that the gearshifts in Table 3 and Table 4 are the most critical ones. Using

$$\omega_{motor} = \frac{30/3{,}6}{r_{wheel}}\left(i_{diff} \cdot i_{gear}\right) \qquad \textbf{Equation 1}$$

and the data from Appendix I., the following two figures are made. The first figure represents the loss in angular velocity at up shifts and the following figure the increase in rotational speed when shifting down.

The shift-actions at which the largest change in angular velocity occurs, are the most critical ones. In these situations a lot of power has to be dissipated. However the change in rotational velocity alone is not the only parameter to determine which shift actions are the most critical ones. The available time for the gearshift is important as well.

6 Reduction of moments of inertia

6.1 Calculation of the inertia's

Because of the construction of a constant mesh gearbox the gearwheels are subject to different angular accelerations. In order to be able to use only one angular velocity for all the masses involved, they will be lumped to one axis. In this case this will be the outgoing axle.

$$J_1 = \left(D_i^4 + D_o^4\right) d \, \frac{\pi\rho}{32} \qquad \textbf{Equation 2}$$

D_i: Inner diameter [m]
D_o: Outer diameter [m]
d: Thickness [m]
ρ: Density [kg/m^3]

Using $J_1 = \left(D_i^4 + D_o^4\right) d \, \frac{\pi\rho}{32}$
Equation 2 and the data in Appendix H. results in the following table.

Part	Symbol	Inertia [kg m^2]
1st gear IS	J_1	1,957E-5
1st gear OS	J_2	1,551E-3
2nd gear IS	J_3	1.414E-5
2nd gear OS	J_4	9,219E-4
6th gear IS	J_5	1,341E-4
6th gear OS	J_6	4,878E-4
4th gear IS	J_7	4,241E-4
4th gear OS	J_8	1,613E-4
5th gear IS	J_9	3,666E-4
5th gear OS	J_{10}	1,927E-4
Clutch	J_C	4,086E-3
Ingoing axle	J_{IS}	1,729E-3

Table 5: Inertia's; OS means outgoing axle and IS ingoing axle

In the table above the calculated inertia's are shown. The dimensions of the gearwheels and axles and therefore the inertia's are based on the information of a picture from the service manual. This picture is shown in Figure 11. The dimensions of the gearwheels were measured in this drawing and the information is represented in Appendix I. Because the inertia's were calculated and their dimensions were extracted from drawings, the obtained values contain errors and should be considered estimates.
1st gear IS means the following: the gearwheel of 1st gear fixed to the ingoing axle. The inertia of the clutch is quite a rough estimate, because the drawing does not represent it very clear.

6.2 Determining the relative angular velocities

In the following table a survey of the angular velocities of the inertia's in each gear is presented. This is necessary to lump the inertia's with respect to one angular velocity.

Selected gear	J_1, J_3, J_{IS}, J_C	J_2	J_4	J_5	J_7	J_9	J_6, J_8, J_{10}
1	$\frac{z_2}{z_1}$	1	$\frac{z_3}{z_4}\cdot\frac{z_2}{z_1}$	$\frac{z_6}{z_5}$	$\frac{z_8}{z_7}$	$\frac{z_{10}}{z_9}$	1
2	$\frac{z_4}{z_3}$	$\frac{z_1}{z_2}\cdot\frac{z_4}{z_3}$	1	$\frac{z_6}{z_5}$	$\frac{z_8}{z_7}$	$\frac{z_{10}}{z_9}$	1
4	$\frac{z_8}{z_7}$	$\frac{z_1}{z_2}\cdot\frac{z_8}{z_7}$	$\frac{z_3}{z_4}\cdot\frac{z_8}{z_7}$	$\frac{z_6}{z_5}$	$\frac{z_8}{z_7}$	$\frac{z_{10}}{z_9}$	1
5	$\frac{z_{10}}{z_9}$	$\frac{z_1}{z_2}\cdot\frac{z_{10}}{z_9}$	$\frac{z_3}{z_4}\cdot\frac{z_{10}}{z_9}$	$\frac{z_6}{z_5}$	$\frac{z_8}{z_7}$	$\frac{z_{10}}{z_9}$	1
6	$\frac{z_6}{z_5}$	$\frac{z_1}{z_2}\cdot\frac{z_6}{z_5}$	$\frac{z_3}{z_4}\cdot\frac{z_6}{z_5}$	$\frac{z_6}{z_5}$	$\frac{z_8}{z_7}$	$\frac{z_{10}}{z_9}$	1

Table 6: z_i are the number of teeth on a gearwheel, indicated with its inertia

The clutch, ingoing axle and inertia's J1 and J3 are all fixed to each other, so they always have the same angular velocities. The rotational speeds stated in Table 6 are reduced with respect to the outgoing axle. This means that, if multiplied with the angular velocity of the outgoing axle, the actual angular velocity is returned.

6.3 Lumping the inertia's

When the transmission is shifted, as in Figure 12, to first gear with inertia's reduced with respect to the outgoing axle of the transmission, the corresponding lumped inertia is:

$$J_{red,1} = J_2 + \left(J_{IS} + J_C + J_1 + J_3 + J_4 \left(\frac{z_3}{z_4} \right)^2 \right) \left(\frac{z_2}{z_1} \right)^2 \qquad \textbf{Equation 3}$$

On the assumption that the output shaft (OS) and the components connected to it are not subject to any change of angular velocity during synchronization, their moments of inertia will be ignored. This is a valid approximation when the road has no gradient.

7 Calculating if the solution will work

7.1 Simplifications and assumptions

The following simplifications are made to use this calculation method

1. Oil temperature of 80 °C
2. The gearshift effort F is constant
3. The friction coefficient μ is constant
4. Torque losses T_V are constant
5. Friction torque T_R is constant
6. Change in angular velocity is constant

The errors resulting from the simplifications are largely offset in the calculation by the acceptable stress values in the synchromeshes. The acceptable stress values are obtained from Appendix H., and are derived from experience.

7.2 Required synchronization force

In this section calculations will be made to estimate if the actuation system can comply with the required specifications. The required specifications are a worst-case scenario, which means that shifts will be made when the engine is at maximum velocities so the axes in the transmission will have their maximum inertia.

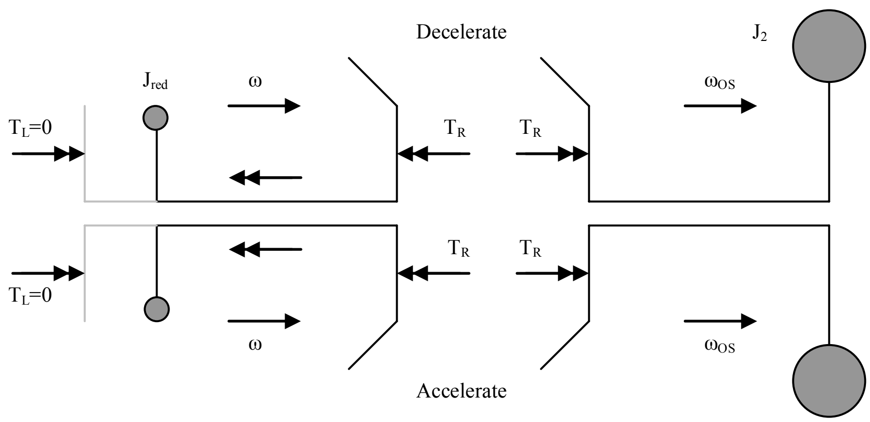


Figure 13: Synchronisation of two equivalent rotating masses

To estimate the shift times, the torque equilibrium for a synchronizer as in Figure 13 must be solved.

$$T_L + \frac{d\omega}{dt} J_{red} + T_V + T_R = 0$$

Equation 4

When the master clutch is fully opened, the load moment T_L is zero throughout the synchronizing process. The torque losses T_V are the result of bearing losses, oil churning losses, oil drag losses and oil compression losses. These losses are specific in each individual transmission and should be measured.
When shifting up, the gearwheel to be shifted is decelerated with the rotating masses reduced to its axis J_{red}. Friction torque and torque losses act in the same direction. When shifting down, the

gearwheel to be shifted is accelerated with the rotating mass reduced to its axis. Friction torque and torque losses act in opposite directions. With all this in mind $T_L + \frac{d\omega}{dt} J_{red} + T_V + T_R = 0$ Equation 4 reduces to:

$$T_R = -\frac{d\omega}{dt} J_{red} - T_V$$ **Equation 5**

For upshifts the term $\frac{d\omega}{dt}$ becomes negative, and now it works in opposite direction of T_V. With the friction torque the friction work can be calculated using

$$W = -\tfrac{1}{2}\omega t_r T_V - \tfrac{1}{2} J_{red}\omega_{\max}^2$$ Equation 6.

$$W = -\tfrac{1}{2}\omega t_r T_V - \tfrac{1}{2} J_{red}\omega_{\max}^2$$ **Equation 6**

Dividing the friction work by the permissible slipping time results in the average friction power, as in

$$P = \frac{W}{t_r}$$ Equation 7.

$$P = \frac{W}{t_r}$$ **Equation 7**

The permissible slipping time is defined as the time it takes to engage an idler gear. With the friction torque and the dimensions of the synchromesh the force practiced on the synchromesh sleeve can be obtained using the following equation:

$$F = \frac{T_r 2\sin\alpha}{d\mu}$$ **Equation 8**

Using $T_R = -\frac{d\omega}{dt} J_{red} - T_V$ Equation 5 to $F = \frac{T_r 2\sin\alpha}{d\mu}$ Equation 8 several plots can be made with varying permissible slipping time. When the permissible slipping time is known the corresponding values for T_r, W, P and F can be found. The maximum permissible slipping time available for each gear change is stated in Table 3 and Table 4.

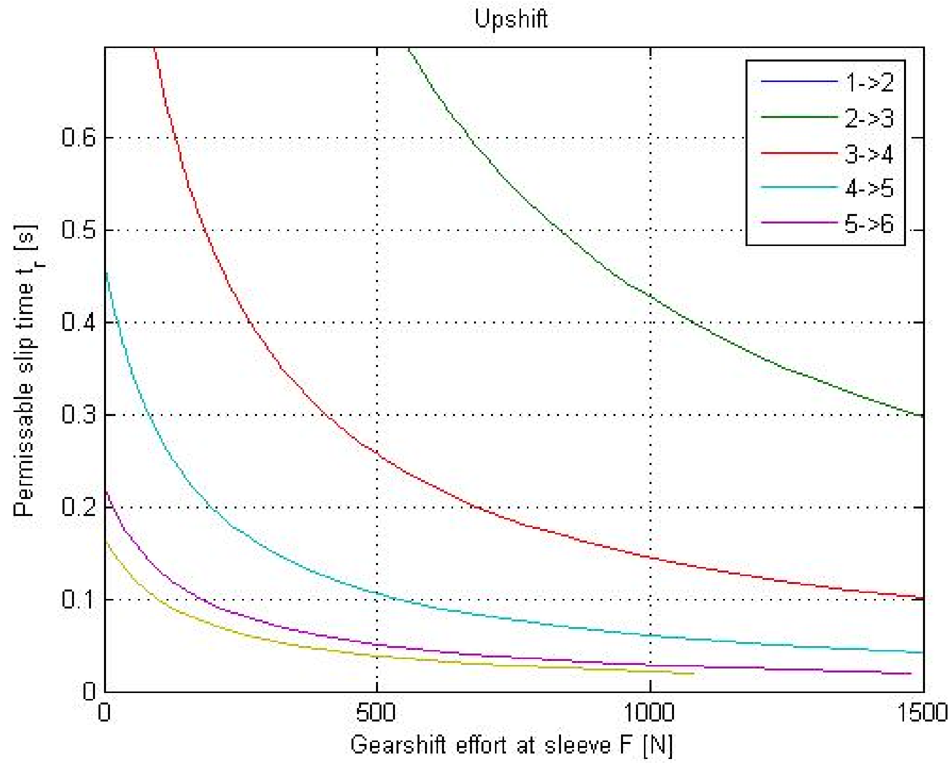


Figure 14: Upshift force

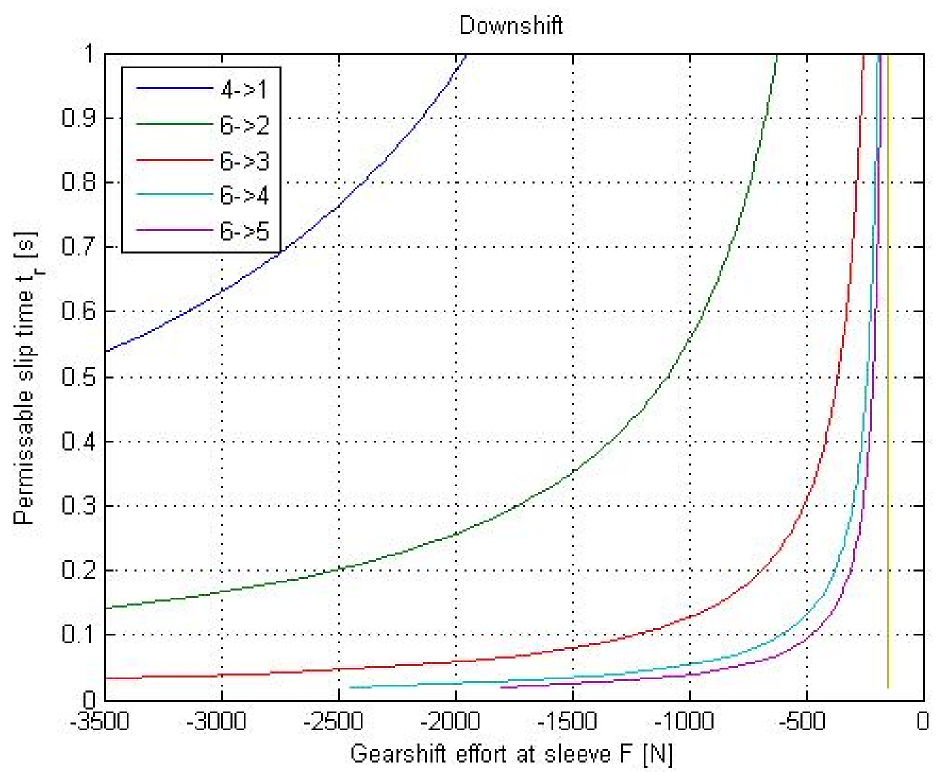


Figure 15: Critical down-shifts

7.3 Properties of the actuating system

Now we can calculate the maximum force the actuators can practice on the gear sleeves. The maximum torque can be determined by

$$M_n = \frac{P}{n}\frac{60}{2\pi} = 0{,}252Nm$$

Equation 9.

$$M_n = \frac{P}{n}\frac{60}{2\pi} = 0{,}252Nm$$

Equation 9

With the maximum torque and the transmission ratio of the actuation system the maximum force can be determined. The transmission ratio is 2500 rad/m, which is the total ratio including everything from rotation of the electro motor till translation of the gear sleeve. The transmission ratio is found in Appendix I..

$$F = M_n i_{actuation} = 630N$$

Equation 10

Now the maximum force the actuator can practice is known we can look at Figure 14 again, and make the following table;

Gearshift	Slip time [s]
1 -> 2	0,65
2 -> 3	0,22
3 -> 4	0,09
4 -> 5	0,045
5 -> 6	0,04

Table 7: Upshift slip times

And for downshifts:

Gearshift	Slip time [s]
4 -> 1	4
6 -> 2	0,95
6 -> 3	0,22
6 -> 4	0,1
6 -> 5	0,07

Table 8: Slip time for downshifts

The required synchronization force is one component that determines the shift time. The other component is the speed at which the actuation system can move from point A to point B, also referred to as the free flight phase. When for example first gear is engaged and the system must go to second gear, the gear sleeve must be moved twice the shift stroke. When a gear is engaged the shift rod and gear sleeve stand completely still. So the electromotor should accelerate then slow down, wait for synchronization (no full stop because of the shift elasticity), move further as fast as possible and then make a stop after second gear is engaged. The specifications of the electromotor are given in Appendix I. they determine the performance. One parameter is not presented yet and that is the maximum angular velocity, being 5000 rev/min which equals 524 rad/s.
The length of the shift stroke is 10 mm, from neutral position to when the gear is engaged, so when another gear is engaged, the stroke must be covered twice, being 20 mm. With the given

specifications of the electromotor the resulting acceleration is plotted in Figure 16 and is calculated with $M_n = J\frac{d\omega}{dt}$ Equation 11 .

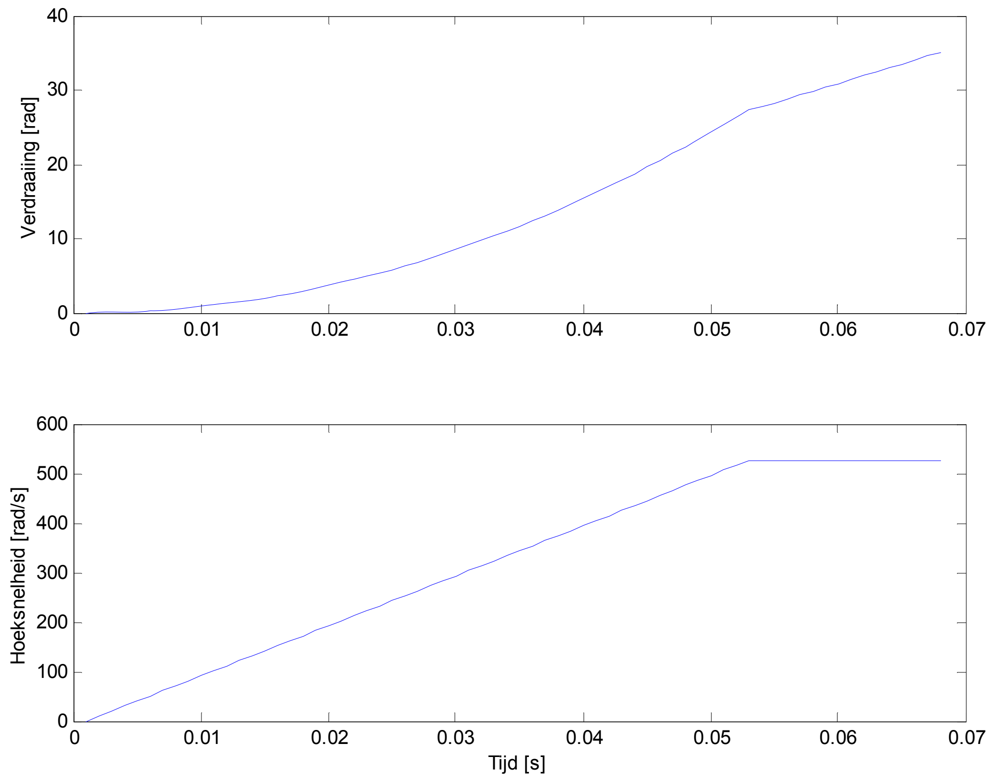


Figure 16: Actuation speed

The displacement is calculated with the following function:

$$M_n = J\frac{d\omega}{dt}$$ **Equation 11**

$$\frac{\partial\omega}{\partial t} = \frac{M_n}{J} = 10080\frac{rad}{s^2}$$ **Equation 12**

$$\omega = i_{actuation}s$$ **Equation 13**

After t = 0,052 s the maximum velocity is reached.

$$M_n = \frac{P}{n}\frac{60}{2\pi} = 0{,}252 Nm$$

Equation 9

states it takes 25 radials to move from the engaged position to neutral. From Figure 16 can be derived it would take 0,051 s to bridge this distance.
The time as stated above is the time required for a shift action, as in a shift from 1st to 2nd gear. When shifting from 2nd to 3rd gear a selection movement must be made as well. However the specifications of the selection motor are unknown. The reduction from rotation of the motor to displacement of the selection rod is also unknown and the only information about the required time can be derived from Figure 10. According to this figure it takes about 100 ms to complete this action. From this figure it can also be concluded that before initiating the selection procedure the shift-lever does not have to be in neutral. So it does not take an extra 100 ms when also a selection action is required. For the length of the shift time, no concrete information can be derived from the figure, because the circumstances are unknown.

7.4 Synchroniser performance limits

In the preceding paragraphs the minimal shift time is calculated. However it is not sure the synchromeshes can dissipate the heat that is generated at these shift speeds. If the synchronizer has to process too much power, the tapers will become too hot, so the performance is determined by the thermal stress. This causes the material properties to change which results in a lower friction coefficient and thus a damaged synchronizer, which will not function properly any longer. By calculating the specific frictional work, which is defined as the absolute work divided by the various gross friction surface areas, the maximum permissible work can be defined.

$$W_A = \frac{|W|}{A_R}$$

Equation 14

With the gross friction surface defined as:

$$A_R = \pi \frac{d_N}{2}\sqrt{\left(9{,}9 \cdot 10^{-3}\right)^2 + \left(\frac{d_N}{2}\right)^2}$$

Equation 15

With the data given in Appendix H., this results in an A_R of 2,5E-3 m^2. The synchronizer ring is made of molybdenum. This material can deal with 0,53 J/mm^2, resulting in a maximum permissible work of 1325 W. Transient peak loads significantly higher than those given may be tolerated. The peak value for specific frictional work W_A in the synchronizer ring friction linings for molybdenum is 1,5 J/mm^2, allowing a work of 3750 W. Additional parameters are the permissible friction speed, power and contact pressure. All these values are calculated in a Matlab script in Appendix C..
Running this file leads to the following conclusion: the synchromeshes are too small to shift this fast. In the worst case scenario the maximum transmitted power for an upshift is 2,03E5 where the maximum power allowed is only $2{,}12 \cdot 10^{-3}$ as in Table 9. In this situation the synchromeshes will be damaged.

Unity	Permitted value
|W|	1,34E3
|P|	2,12E3
|F|	1,52E4

Table 9: Values of check_ist.m

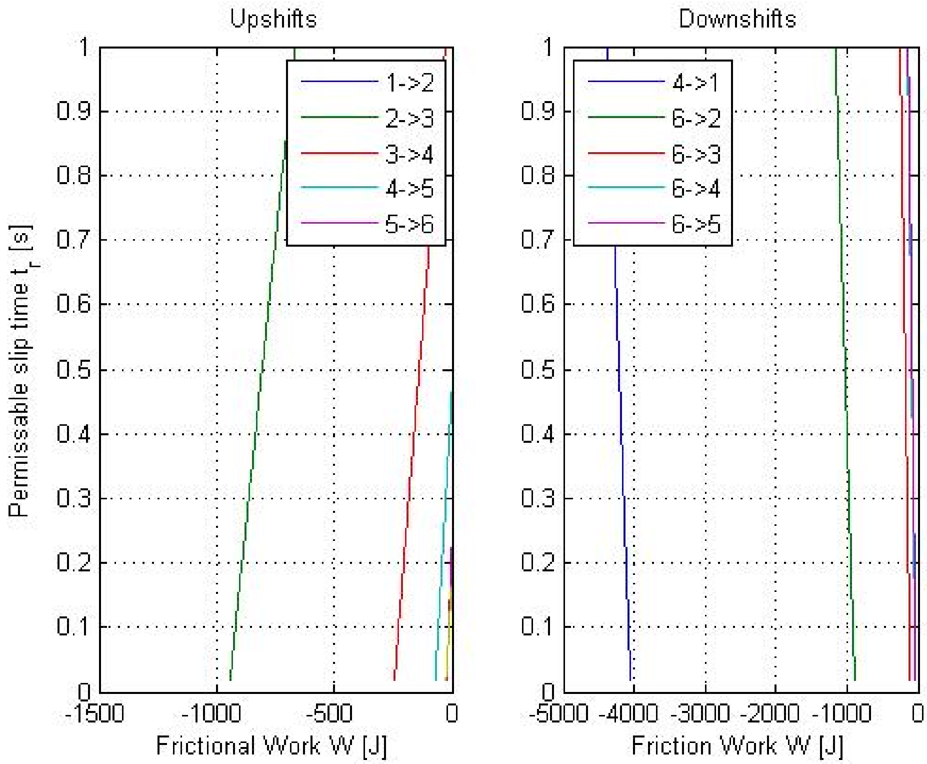


Figure 17: Left; upshift work, Right; downshift work

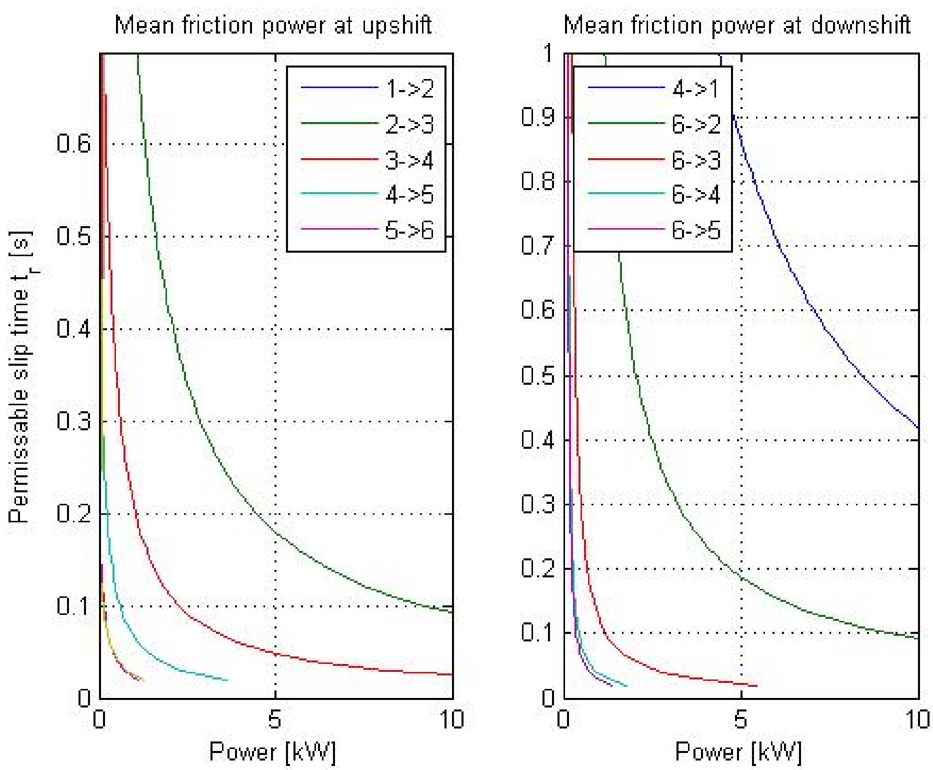


Figure 18: Left; upshift power, Right; downshift power

In Figure 17 and Figure 18 the values in the table are very obvious as well as the impact on the permissible slipping time. At upshifts the only problem is the shift action from 1^{st} to 2^{nd} gear, limiting the permissible slipping time to a minimum of0,4 seconds. When shifting down the thermal performance of the synchronizer is a bigger limitation, especially when shifting from 4^{th} to 1^{st} gear.

8 Interpretation of the results

First we will compare the required shift times with the calculated shift times of the actuation system. We can combine Table 3 with Table 7 and Table 4 with Table 8, however this is not a fair comparison because Table 7 and Table 8 represent only the synchronization time. The time the actuator needs to move from the engaged to the chosen idler gear must be added.

Gearshift	Minimum achievable slip time [s]	Desired slip time [s]
1 -> 2	0,701	0,44
2 -> 3	0,371	0,93
3 -> 4	0,141	0,2
4 -> 5	0,196	0,2
5 -> 6	0,091	0,2

Table 10: Comparison achievable and desired upshift times

Gearshift	Minimum achievable slip time [s]	Desired slip time [s]
4 -> 1	4,151	1,62
6 -> 2	0,201	0,45
6 -> 3	0,22	0,47
6 -> 4	0,151	0,66
6 -> 5	0,221	1,23

Table 11: Comparison achievable and desired downshift times

When changing to a higher gear there are not a lot of problems, only changing from first to second gear takes too long. When shifting to a lower gear however a problem occurs when shifting from fourth to first gear. This takes very long and this is due to the high inertia of the gearbox, which is a result of the first gear ratio.
Now that the minima of the shift times in the worst-case scenario are calculated and the minimum shift times looking at the thermal stress expressed by permissible work and power, these results should be compared. The minimum slip time for the calculated gear changes were given in Table 7 and Table 8. The minimum shift times when looking at the thermal stresses are shown in Figure 18.

Gearshift	Slip time [s]
1 -> 2	0,39
2 -> 3	0,11
3 -> 4	0,03
4 -> 5	Less then 0,03
5 -> 6	Less then 0,03

Table 12: Minimum thermal slip times for upshifts

Gearshift	Slip time [s]
4 -> 1	4
6 -> 2	0,5
6 -> 3	0,05
6 -> 4	Less then 0,03
6 -> 5	Less then 0,03

Table 13: Minimum thermal slip times at downshifts

When looking at these tables we can conclude that the thermal restrictions are not exceeded yet. When we take a look at the minimum thermal slip times for all gear changes and compare them with the required specifications, the following remarks can be made.

A stronger electromotor can be used to increase performance since the synchronizers do not carry their maximum load yet. However shifting from fourth to first gear can not be much faster and a more powerful electromotor would cause damage to the friction material of the first gear. The remaining gear changes meet their desired specifications. These calculations are done in case of a worst case scenario, so when driving normal and shifting with a more likely engine speed, like for example at maximum torque of the engine, the Easytronic system would be applicable, with an exception for the fourth to first gear change at maximum engine speed.

9 Conclusions and recommendations

9.1 Conclusions

An electromechanical actuation system has some obvious advantages compared to a hydraulic actuation system, especially in the field of cost and packaging. Its lower power density is its main disadvantage, but we have seen from the calculations that it should perform quite well.
More over it is possible to increase performance, which is desired at upshifts. The increase in performance can be obtained by fitting a more powerful electromotor. However thermal restrictions have to be taken into account, since the performanceof the synchromeshes will be met.

9.2 Recommendations

Some important parameters used in the calculations and which are stated in the specification sheet in Appendix H. might differ from reality, and these parameters should be checked. Also the parameters of the specific work, power and contact pressure are not from this transmission so they could be wrong.

10 Bibliography

1. Antrieb und Getriebe – Aral
2. De complexe aandrijflijn – Kluwer technische boeken
3. Automatische Fahrzeuggetriebe – Springer-Verlag
4. Automotive Transmissions – Fundamentals, selection, design and application – G. Lechner, H. Naunheimer, Springer
5. De elektronische versnellingsbak, Zelfstudieprogramma 221 – Constructie en werking
6. Technology survey on smartness added to automotive manual transmissions – J.D.W. de Cock
7. Das tribologische Verhalten von Syncrhonisierungen unter Berücksichtigung
8. Beanspruchungskollektivs – Dipl. –Ing. Tobias Lösche 1997
9. Getriebe in Fahrzeugen 2001 – VDI-Berichte 1610
10. Patent number GB2316723
11. Patent number GB2313886
12. Patent number WO03087632
13. Patent number DE19725816
14. Patent number WO03087628
15. Patent number WO03081091

Appendix A.

Synchromesh dimensions

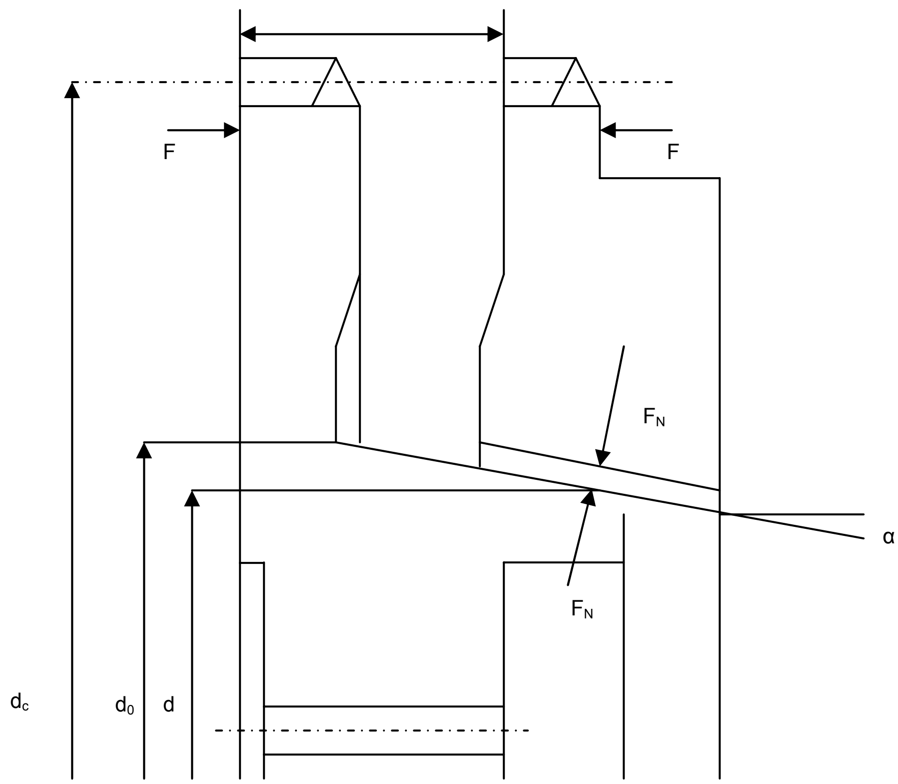


Figure 19: Schematic representation of a synchromesh

The symbols used in Figure 19 are:

d	Effective diameter
d_0	Nominal diameter
d_C	Clutch diameter
F	Gear shift effort
F_N	Normal force
s	shift movement at the gearshift sleeve
α	Taper angle

The steps in the synchronization process

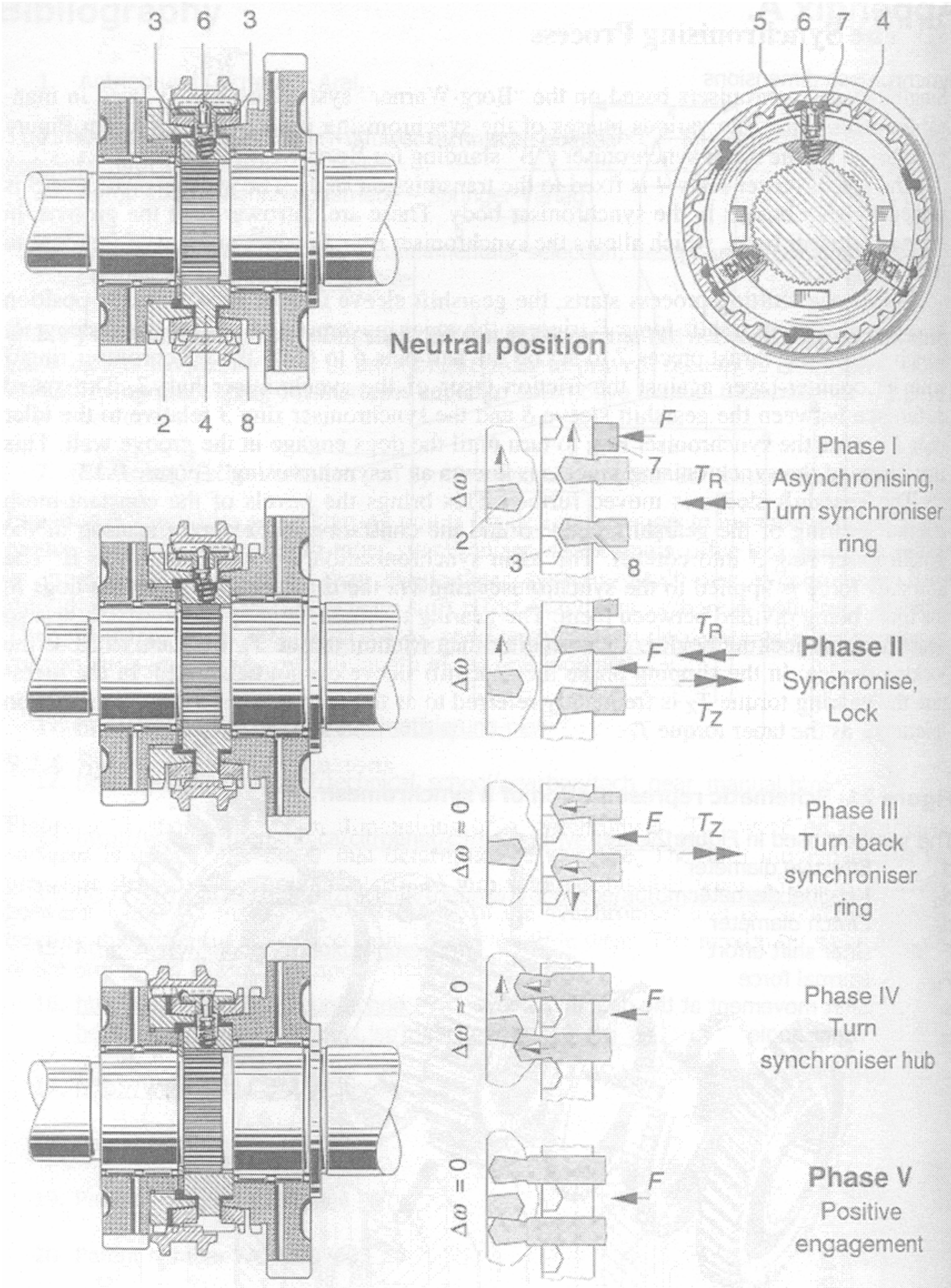


Figure 20: Syncrhonisation process

Appendix B.

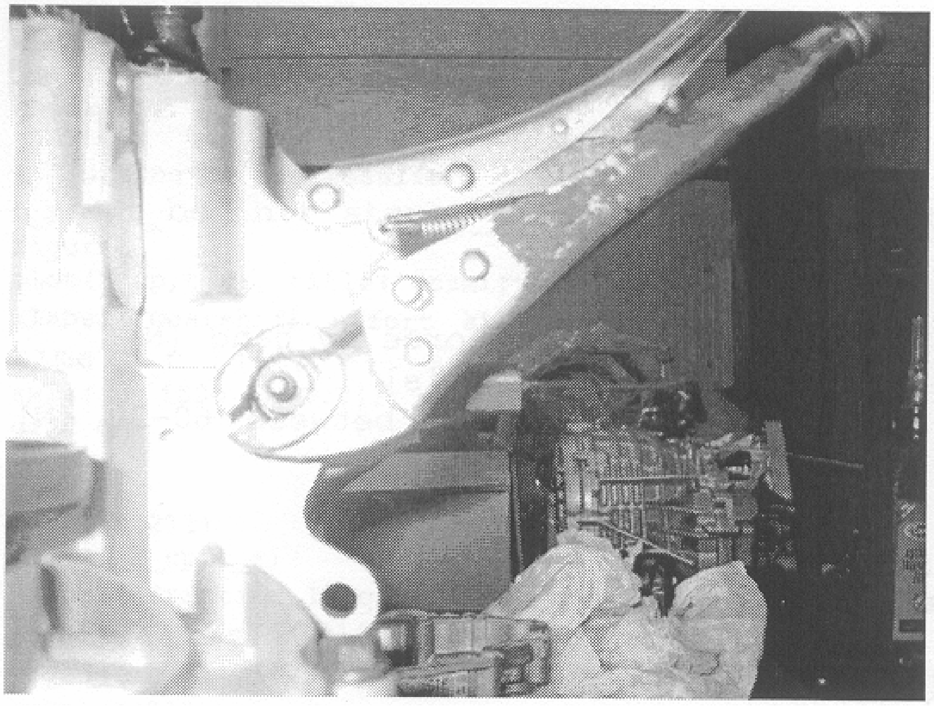


Figure 21: Position 1

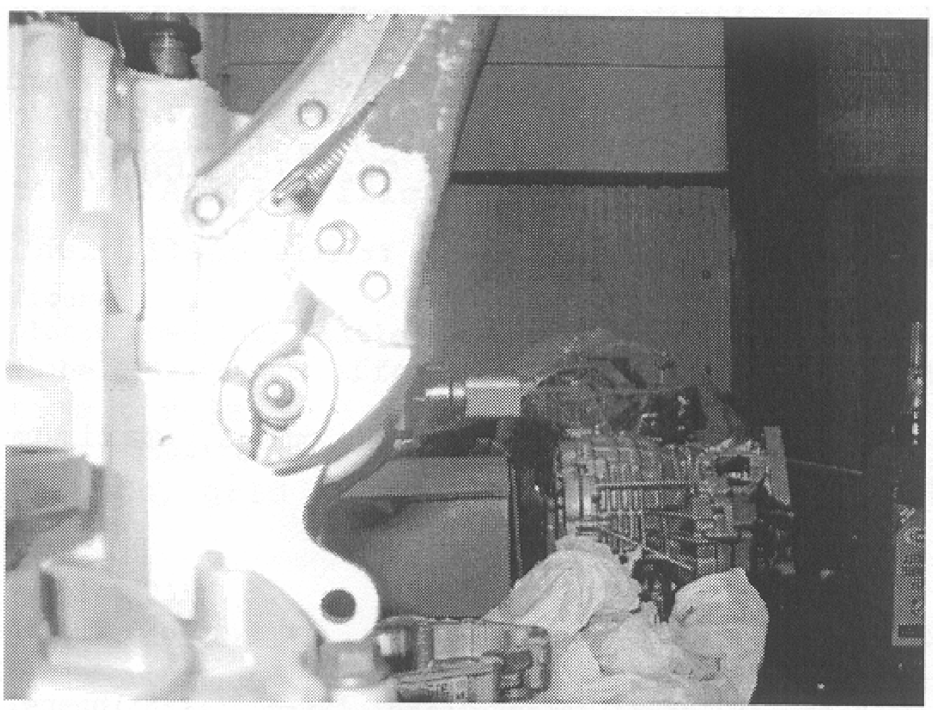


Figure 22: Position 2

Appendix C.

Synchromesh_sti.m

```
% Calculation of the shift force

clear all, close all, clc, format long

% Gearbox specific parameters
sti
rev_change_sti

% General and estimated parameters
T_V = 2;                % [Nm]  Torque losses
F_Hperm = 100;          % [N]   Permitted hand shift force
t_Rperm = 0.25;         % [s]   Permitted slip time
F_kies = 60;            % [N]   selection force (30-60)

% Slip time
t_begin = .02;          % Start time
t_stap = .01;           % time step size
t_eind = 1;             % End

% are the tapers self-locking?
if tan(alpha) >= mu
    disp('No self-locking occurs')
else disp('Self-locking of the tapers')
end

% Calculating opening torque
%T_Z = F_a*d_C/2*((cos(beta/2)-
mu_D.*sin(beta/2))/(sin(beta/2)+mu_D*cos(beta/2)));
% T_Z = 1/2*F_a*d*acos(beta/2) vereenvoudiging, negeren van de frictie
coefficient mu_D

for u = 1:2 %u == 1 upshift, u == 2 downshift
    % maximum difference in sliding velocity
    v_max = omega_max(:,u)*d_N/2;    % also possible with omega

    T_r = [];
    W = [];
    P = [];
    Ps = [];
    for t_r = t_begin:t_stap:t_eind
        % Required friction torque
        T_r_tmp = -J_red.*omega_max(:,u)/t_r-T_V;
        T_r = [T_r,T_r_tmp];
        % Friction work per shift action
        W_tmp = -1/2.*omega_max(:,u)*t_r.*T_V-
1/2.*J_red.*omega_max(:,u).^2;
        W = [W,W_tmp];
        % Mean friction power
        P_tmp = W_tmp/t_r;
        P = [P,P_tmp];
        % Power transmitted momentarily to the synchromesh
        %Ps_tmp = T_r_tmp.*omega_max(:,u);
```

```
        %Ps = [Ps,Ps_tmp];
    end

    % Results in the required axial force belonging to a certain slip
time
    F = T_r*(2*sin(alpha))/(d*mu);

    % Correct for the number of friction tapers
    for l = 1:(length(ratios)-1)
        F(l,:) = F(l,:)/j(l);
    end

    if u == 1
        T_r_up = T_r; W_up = W; P_up = P; Ps_up = Ps; F_up = F;
    end
    if u == 2
        T_r_down = T_r; W_down = W; P_down = P; Ps_down = Ps; F_down =
F;
    end
    clear F P W T_r Ps
end

Check_ist

% Plotten the results
t_r = t_begin:t_stap:t_eind;    % Time axis for plotting
figure
plot(F_up,t_r), title('Upshift')
xlabel('Gearshift effort at sleeve F [N]'), ylabel('Permissable slip
time t_r [s]')
axis([0 1500 0 0.7]), legend('1->2','2->3','3->4','4->5','5->6',1),
grid on

figure
subplot(122)
plot(W_down,t_r)
xlabel('Friction Work W [J]'), title('Downshifts')
axis([-5000 0 0 1]), legend('4->1','6->2','6->3','6->4','6->5',2), grid
on
subplot(121)
plot(W_up,t_r)
xlabel('Frictional Work W [J]'), ylabel('Permissable slip time t_r
[s]'), title('Upshifts'), axis([-1500 0 0 1])
legend('1->2','2->3','3->4','4->5','5->6',1), grid on

figure
subplot(121)
plot(abs(P_up)/1000,t_r)
xlabel('Power [kW]'), ylabel('Permissable slip time t_r [s]'),
title('Mean friction power at upshift')
axis([0 10 0 0.7]), legend('1->2','2->3','3->4','4->5','5->6',1), grid
on
subplot(122)
plot(abs(P_down)/1000,t_r)
xlabel('Power [kW]'), title('Mean friction power at downshift')
axis([0 10 0 1]), legend('4->1','6->2','6->3','6->4','6->5',2), grid on
```

```
figure
plot(F_down,t_r), title('Downshift')
xlabel('Gearshift effort at sleeve F [N]'), ylabel('Permissable slip
time t_r [s]')
axis([-3500 0 0 t_eind]), legend('4->1','6->2','6->3','6->4','6->5',2),
hold on, grid on

figure
plot(T_r_down,t_r), xlabel('Friction Torque T_r [Nm]'), axis('tight'),
hold on
plot(W_down,t_r)
xlabel('Frictional Work W [J]'), axis('tight')
% *(1)  Das tribologische Verhalten von synchronisierungen unter
Berucksichtigung des Beanspruchungskollektivs KSN 97 LOE
% *(2)  blz 246 Lechner & Naunheimer
```

Appendix D.

Sti.m

```
% Parameters voor versnellingsbak Volkswagen 02K DNZ in combinatie met
STI
scale = (65/23)*1e-3;              % [-]       schaal van de tekening

rho = 7800;                        % [kg/m^3]  dichtheid materiaal
gebruikte tandwielen

% Versnellingsbak specifieke parameters
mu = .1;                           % [-]       gemiddelde
wrijvingswaarde in het konusvlak *(1)
alpha = 11.42*pi/180;              % [rad]     halve kegelhoek van de
synchronisers
beta = 115*pi/180;                 % [rad]     kegel hoek / opening
angle van de tanden op de synchronisers
d = 54e-3;                         % [m]       effectieve diameter
(halverwege wrijvingsopp synchro)
d_C = 70e-3;                       % [m]       clutch diameter (v.d.
synchromesh waar de dogs zitten)
d_N = 55e-3;                       % [m]       nominal diameter
% voor definities van bovenstaande drie zie *(1) blz 234
i_actuation = 2500;                % [rad/m]   Total gear ratio of the
actuation part
h = 65e-3;                         % [m]       hartafstand tussen
prise en pignont as
omega = [650 650 650 650 565 332];  % [rad/s] toerental waarbij
geschakeld wordt (worst-case maxvermogen@ 6200 rpm, maxkoppel@ 3200
rpm)
s = 10e-3;                         % [m]       Weg die de gearshift
sleeve aflegt 10-13 [mm]
j = [1,1,1,1,1,1];                 % [-]       Allemaal enkele
wrijvingsvlakken volgens Christian
z = [11,38,18,35,28,36,32,31,41,33,55,31];  %   3e versnelling wordt
6e, tandwielen omdraaien nog niet gedaan
ratios = [z(2)/z(1) z(4)/z(3) z(6)/z(5) z(8)/z(7) z(10)/z(9)
z(12)/z(11)];   %   Wordt gevraagd door synchromesh.m
i_diff = 68/18;                    % [-]       overbrenging
differentieel

% Gemeten waarden uit tekening

% Traagheid ingaande as
L_as = [29,4,8,1,11,10,9]*scale;    % van rechts naar links op de
tekening plus synchromeshes
D_as = [7,9,9,14,28,9,28]*scale;    % idem ook inclusief de
synchromeshes
D_as_i = 4*scale;                  % inwendige diameter, uitsparing
voor pen van de koppelingsbediening
J_IS = (sum((D_as.^4).*L_as)-D_as_i^4*sum(L_as))*(pi/32*rho);
%Traagheid Koppeling
D_clutch = [0.03 0.14 210e-3];      % gedeeltelijk uit easydata
D_clutch_i = [0.02 0.03 0.14];      % uit easydata
d_clutch = [0.04 0.011 0.0007];     % ook uit easydata
```

```
J_C = (sum((D_clutch.^4).*d_clutch)-
sum((D_clutch_i.^4).*d_clutch))*(pi/32*rho);
% Traagheid J1
d1 = [6,3]*scale;
D1 = [12,9]*scale;
J(1) = sum((D1.^4).*d1);
% Traagheid J2
d2 = [5,1]*scale;
D2 = [38,30]*scale;
J(2) = sum((D2.^4).*d2);
% Traagheid J3
d3 = [5]*scale;
D3 = [12]*scale;
J(3) = sum((D3.^4).*d3);
% Traagheid J4
d4 = [5]*scale;
D4 = [34]*scale;
J(4) = sum((D4.^4).*d4);
% Traagheid J5
d5 = [5]*scale;
D5 = [21]*scale;
J(5) = sum((D5.^4).*d5);
% Traagheid J6
d6 = [5]*scale;
D6 = [29]*scale;
J(6) = sum((D6.^4).*d6);
% Traagheid J7
d7 = [5]*scale;
D7 = [28]*scale;
J(7) = sum((D7.^4).*d7);
% Traagheid J8
d8 = [5]*scale;
D8 = [22]*scale;
J(8) = sum((D8.^4).*d8);
% Traagheid J9
d9 = [5]*scale;
D9 = [27]*scale;
J(9) = sum((D9.^4).*d9);
% Traagheid J10
d10 = [5]*scale;
D10 = [23]*scale;
J(10) = sum((D10.^4).*d10);
% Traagheid van J1 t/m J6 reduceren, omdat ze hol zijn
d_i =
[sum(d1),sum(d2),sum(d3),sum(d4),sum(d5),sum(d6),sum(d7),sum(d8),sum(d9
),sum(d10)];
D_i = [4,5,4,5,4,5,4,5,4,5]*scale;                          %
binnendiameters J1,J2..,J10
J_i = (D_i.^4).*d_i;
J = (J-J_i)*(pi/32*rho);                                    %
corrigeren van traagheden en opslaan in een rij

%Berekende waarden van de diameters en hierbij behorende traagheden ter
controle
for k = 1:2:10
    Db(k) = 2*(z(k)/z(k+1)*h)/(z(k)/z(k+1)+1);
    Db(k+1) = 2*h/(z(k)/z(k+1)+1);
```

```
end
Jb = ((Db.^4).*5)*(pi/32*rho);

% Gereduceerde traagheden bepalen zodat er met 1 hoeksnelheid gerekend
kan worden
J_red(1,1) =
J(2)+(J_IS+J_C+J(1)+J(3)+J(4)*(z(3)/z(4))^2)*(z(2)/z(1))^2;
J_red(2,1) =
J(4)+(J_IS+J_C+J(1)+J(3)+J(2)*(z(1)/z(2))^2)*(z(4)/z(3))^2;
J_red(3,1) =
(J_IS+J_C+J(1)+J(3)+(J(4)*(z(3)/z(4))^2+J(2)*(z(1)/z(2))^2))*(z(6)/z(5)
)^2;
J_red(4,1) =
(J_IS+J_C+J(1)+J(3)+(J(4)*(z(3)/z(4))^2+J(2)*(z(1)/z(2))^2))*(z(8)/z(7)
)^2;
J_red(5,1) =
(J_IS+J_C+J(1)+J(3)+(J(4)*(z(3)/z(4))^2+J(2)*(z(1)/z(2))^2))*(z(10)/z(9
))^2;
J_red(6,1) =
(J_IS+J_C+J(1)+J(3)+(J(4)*(z(3)/z(4))^2+J(2)*(z(1)/z(2))^2))*(z(12)/z(1
1))^2;
```

Appendix E.

Actuatie.m

```
% Actuatie berekeningen
% Gegevens van de actuator

n = 5000;                        %[omw/min]
m = .813;                        %[kg]
V = 162;                         %[W/kg] Vermogensdichtheid
omega_max = n*2*pi/60;           %[rad/s]
J_schalt = 25e-6;                %[kgm^2]
P = 163*m;                       %[W]
tau = 7.46e-3;                   %[s]
phi = s*i_actuatie_totaal;       %[rad] hoekverdraaing motor

M = P/omega_max;                 %[Nm]
max_accel = M/J_schalt;          %[rad/s^2]
t_v_max = omega_max/max_accel;   %[s]
F_shift = M*i_actuation;         %[N]

t_tmp=0;
t = [];
omega = [];
omega_tmp = 0;
k = [];
k_tmp = 0;
while k_tmp < phi
    if omega_tmp < omega_max
        omega_tmp = max_accel*t_tmp;
        k_tmp = omega_tmp*t_tmp;
    else
        k_tmp = omega_max*t_tmp;
    end
    omega = [omega,omega_tmp];
    k = [k,k_tmp];
    t_tmp=t_tmp+.001;
    t = [t,t_tmp];
end

subplot(211)
plot(t,k)
ylabel('Angle [rad]')
subplot(212)
plot(t,omega)
ylabel('Angular velocity [rad/s]'),xlabel('Time [s]')
axis('tight')
```

Appendix A.

Rev_change_sti.m

```
% Opschakel acties, 1->2,2->3,3->4,4->5,5->6, sequentieel

%Opschakelen
for k = 1:(length(ratios)-1);
    omega_up(k,1) = omega(k)*(ratios(k+1)/ratios(k))-omega(k);
end
omega_up = [0;omega_up]; % toevoegen van een nul om dat 0->1 met de
koppeling gebeurt

% Te bepalen terugschakel acties
% 2->1 3->1 4->1
% 3->2 4->2 5->2 6->2
% 4->3 5->3 6->3
% 5->4 6->4
% 6->5
v_car = [30,25,40,50,50,60,60,80,80,60,100,80,140]/3.6;
g = [2,3,4,3,4,5,6,4,5,6,5,6,6];
r_tire = .316;

%Terugschakelen
omega1 = [];
for k = 1:3;
    omega_engine = v_car(k)/r_tire*(ratios(g(k))*i_diff);
    omega1_tmp = omega_engine*ratios(1)/ratios(g(k))-omega_engine;
    omega1 = [omega1;omega1_tmp];
end
t1 = [0.73; 0.92; 1.62];
dwdt1 = omega1./t1;
omega2 = [];
for k = 4:7;
    omega_engine = v_car(k)/r_tire*(ratios(g(k))*i_diff);
    omega2_tmp = omega_engine*ratios(2)/ratios(g(k))-omega_engine;
    omega2 = [omega2;omega2_tmp];
end
t2 = [0.46; 0.72; 0.98; 1];
dwdt2 = omega2./t2;
omega3 = [];
for k = 8:10;
    omega_engine = v_car(k)/r_tire*(ratios(g(k))*i_diff);
    omega3_tmp = omega_engine*ratios(3)/ratios(g(k))-omega_engine;
    omega3 = [omega3;omega3_tmp];
end
t3 = [0.35; 0.55; 0.65];
dwdt3 = omega3./t3;
omega4 = [];
for k = 11:12;
    omega_engine = v_car(k)/r_tire*(ratios(g(k))*i_diff);
    omega4_tmp = omega_engine*ratios(4)/ratios(g(k))-omega_engine;
    omega4 = [omega4;omega4_tmp];
end
t4 = [0.25; 0.47];
dwdt4 = omega4./t4;
omega5 = [];
for k = 13;
```

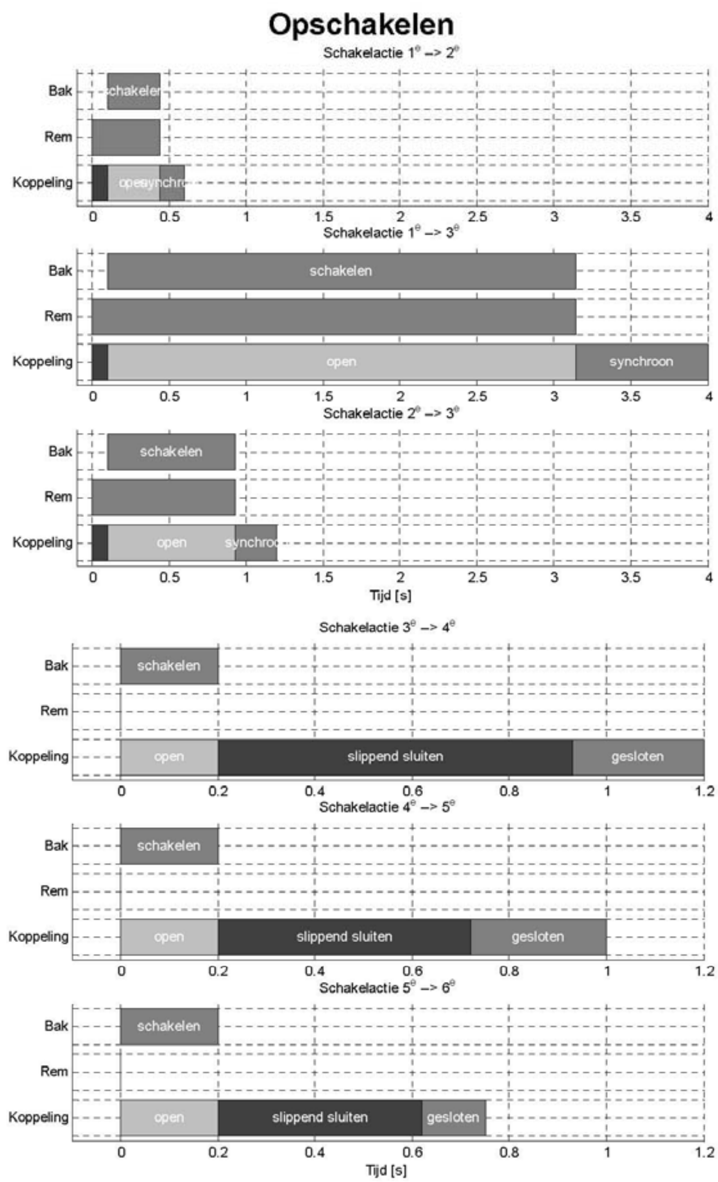
```
    omega_engine = v_car(k)/r_tire*(ratios(g(k))*i_diff);
    omega5_tmp = omega_engine*ratios(5)/ratios(g(k))-omega_engine;
    omega5 = [omega5;omega5_tmp];
end
t5 = 0.45;
dwdt5 = omega5/t5;

% Determining the most critical shift actions
[y,n] = max(dwdt1);
[y.n] = max(dwdt2);
[y.n] = max(dwdt3);
[y.n] = max(dwdt4);
[y.n] = max(dwdt5);

omega_down =
[max(omega1);max(omega2);max(omega3);max(omega4);max(omega5);0];
omega_max = [omega_up, omega_down]; % adapting for synchromesh_sti.m
```

Appendix F.

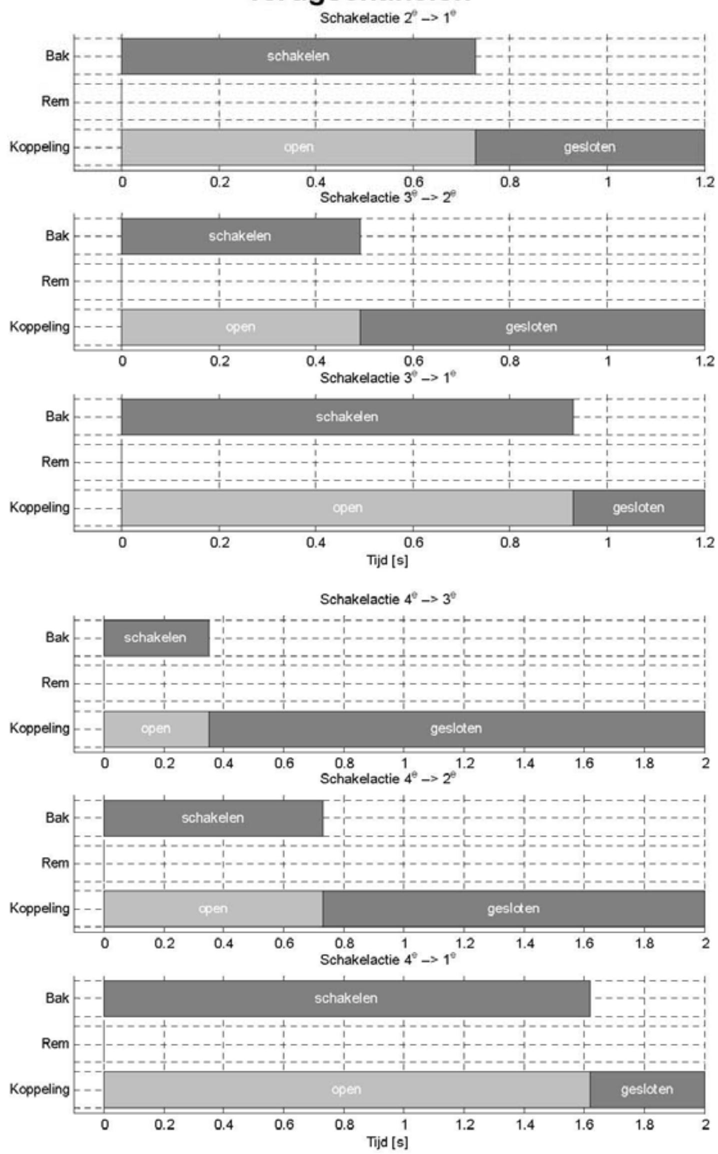
Check_ist.m

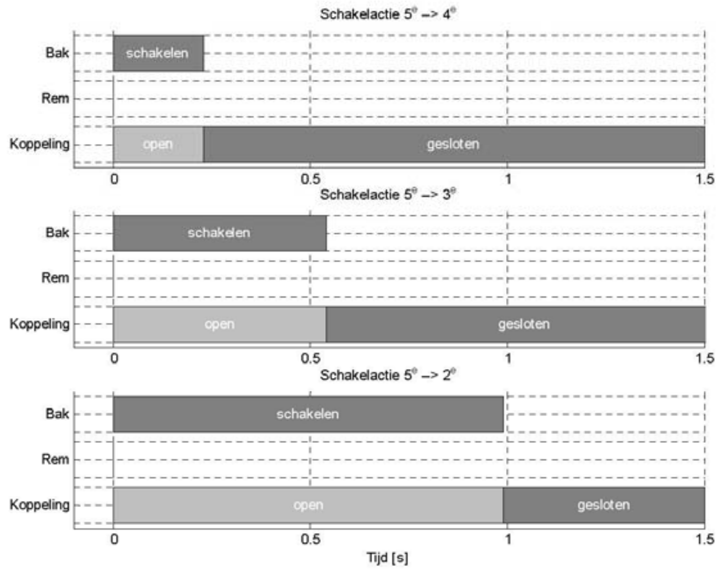
```
% Checking the results of synchromesh_sti.n

% Parameters - molybdenum
W_A = 0.53;      % [J/mm^2]  Specific frictional work
W_A_peak = 1.5; % [J/mm^2]  Specific frictional work at peak load
P_A = 0.84;      % [W/mm^2]  Specific frictional power
p_R = 6;         % [N/mm^2]  Contact pressure
v_perm = 7;      % [m/s]     Permissible friction speed

% Calculations
A_R = 1e6*(pi*d_N/2*sqrt((9.9e-3)^2+(d_N/2)^2));
W_perm = W_A*A_R;
W_perm_peak = W_A_peak*A_R;
P_perm = P_A*A_R;
F_perm = p_R*A_R;

% Comparison - results
if W_perm > max(max(abs(W_up)))
    disp('Upshift frictional work is within limits')
else
    disp('Upshift frictional work is too large')
end

if W_perm > max(max(abs(W_down)))
    disp('Downshift frictional work is within limits')
else
    disp('Downshift frictional work is too large')
end

if P_perm > max(max(abs(P_up))) % Momentarily transmitted power
    disp('Upshift specific friciton power is ok')
else
    disp('Upshift specific friction power is too large')
end

if P_perm > max(max(abs(P_down)))
    disp('Downshift specific friciton power is ok')
else
    disp('Downshift specific friction power is too large')
end

if F_perm > max(max(abs(F_up)))
    disp('Upshift contact pressure is ok')
else
    disp('Upshift contact pressure is too large')
end

if F_perm > max(max(abs(F_down)))
    disp('Downshift contact pressure is ok')
else
    disp('Downshift contact pressure is too large')
end
```

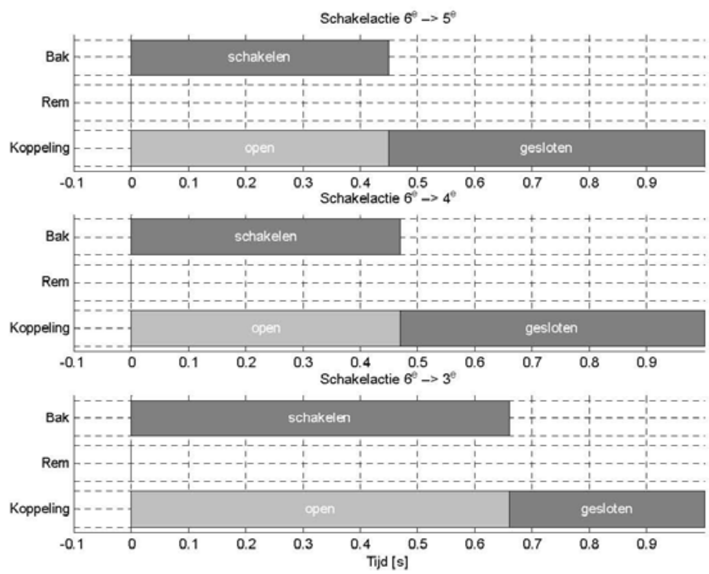
Appendix G.


Opschakelen
Schakelactie 1e –> 2e
Bak
Rem
Koppeling
schakelen
open
synchroon
Schakelactie 1e –> 3e
Schakelactie 2e –> 3e
Tijd [s]
Schakelactie 3e –> 4e
slippend sluiten
gesloten
Schakelactie 4e –> 5e
Schakelactie 5e –> 6e
0
0.5
1
1.5
2
2.5
3
3.5
4
0.2
0.4
0.6
0.8
1.2

Terugschakelen




Schakelactie 5^e –> 4^e
Bak
Rem
Koppeling
schakelen
open
gesloten
0
0.5
1
1.5
Schakelactie 5^e –> 3^e
Bak
Rem
Koppeling
schakelen
open
gesloten
0
0.5
1
1.5
Schakelactie 5^e –> 2^e
Bak
Rem
Koppeling
schakelen
open
gesloten
0
0.5
1
1.5
Tijd [s]


Schakelactie 6^e –> 5^e
Bak
Rem
Koppeling
schakelen
open
gesloten
-0.1
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
Schakelactie 6^e –> 4^e
Bak
Rem
Koppeling
schakelen
open
gesloten
-0.1
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
Schakelactie 6^e –> 3^e
Bak
Rem
Koppeling
schakelen
open
gesloten
-0.1
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
Tijd [s]

Appendix H.

Parameter	Geschatte Waarde	Gemeten Waarde	Eenheid	Symbool	Hoe te verkrijgen
Synchromesh					
afstand die schakelmof aflegt (van neutral tot synchro ring)	7	10	mm	s	Meten
aantal wrijvingsvlakken in synchromesh				j	kijken
naar 1 of 2	1	1		j1	kijken
naar 4 of 6	1	1		j2	kijken
naar 5	1	1		j3	kijken
halve kegelhoek synchroniser	6,5	11,42	graden	alpha	meten
opening angle tanden van de dogs	115		graden	beta	meten
effectieve diameter	5.70E-02	0,054	m	d	meten
clutch diameter	6.90E-02	0,07	m	d_C	Meten
nominale diameter	6.00E-02	0,055	m	d_N	Meten
wrijvingscoefficienten	0.1			mu	afschatten
dichtheid	7800	6250			
Versnellingsbak					
dikte tandwielen				d	meten
tandwiel 1 (zie schematische tekening)	0.0170 0.0085	17,3E-3 23,8E-3	m	d1	
tandwiel 2	0.0141 0.0028	1,4E-2	m	d2	
tandwiel 3	0.0141	1,53E-2	m	d3	
tandwiel 4	0.0141	1,48E-2	m	d4	
tandwiel 5	0.0141	1,46E-2	m	d5	
tandwiel 6	0.0141	1,55E-2	m	d6	
tandwiel 7	0.0141	1,48E-2	m	d7	
tandwiel 8	0.0141	1,52E-2	m	d8	
tandwiel 9	0.0141	1,54E-2	m	d9	
tandwiel 10	0.0141	1,41E-2	m	d10	
Lengte van prise as				L_IS	meten
Lengte van pignont as				L_OS	meten
traagheid ingaande as	0.0017			J_IS	afschatten en uitrekenen
Gereduceerde traagheden				J_red	afschatten en uitrekenen
naar 1	0.0743			J_red1	
naar 2	0.0235			J_red2	
naar 3	0.0103			J_red3	
naar 4	0.0058			J_red4	
naar 5	0.004			J_red5	
naar 6	0.002			J_red6	
Stijfheid veren op de schakelas			N	k	meten
Overbrengingsverhoudingen (automotive notatie)	3,45 1,94 1,29 0,97 0,8	ok		ratios	bekend
Hartafstand prise as en pignont as	65		mm	h	bekend
Toerental waarbij geschakeld wordt			rad/s	omega	van Christian
in 1	650		rad/s	omega1	
in 2	650		rad/s	omega2	
in 3	650		rad/s	omega3	
in 4	650		rad/s	omega4	
in 5	565		rad/s	omega5	
in 6	332		rad/s	omega6	
Koppel verliezen in de versnellingsbak	2		Nm	T_V	geschat lechner
Kieskracht	60		N	F_kies	geschat lechner
Actuatie gedeelte					
maximale omwentelingssnelheid schakelactuator	5000		omw/min	n	LuK kap 13 .pdf
massa schakelactuator	0.813		kg	m	LuK kap 13 .pdf
traagheid schakel motor	2.50E-05		kg*m^2	J_schalt	LuK kap 13 .pdf
maximale vermogen	132.5		W	P	LuK kap 13 .pdf
nominaal koppel	0.2531		Nm	M	berekend
totale overbrenging schakelactie	2500		rad/m	i_actuatie_totaal	patent schakelelasticiteit.pdf

Table 14: Dimension and parameter survey

Appendix I.

	heutiger Kupplungs-motor (DC)	heutiger Schaltmotor (DC)	neuer Motor (EC)
Leistungsdichte	101 W/kg 100%	163 W/kg 162%	267 W/kg 266%
Massenträgheit	$30{,}4 \cdot 10^{-6}$ kgm² 100%	$25{,}0 \cdot 10^{-6}$ kgm² 82%	$6{,}5 \cdot 10^{-6}$ kgm² 21%
mechanische Zeitkonstante	27,75 ms 100%	7,46 ms 27%	1,88 ms 6,8%
Gewicht	693 g 100%	813 g 117%	438 g 63%
Volumen	166 cm^2 100%	162 cm^2 98%	62 cm^2 37%

Bild 16: *Kennwerte neue Elektromotoren*