Development of a Program for Calculating Gearbox Synchronization

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Abstract

Synchronizers are used to reduce shift time and increase comfort for manual transmission gearboxes. The increased torque in modern trucks as well as overall vehicle refinements places higher demands on synchronizers. The short duration of the synchronization process, the tribological contact with many parameters including oil additives and surface properties as well as the requirement of many cycles before failure makes experimental testing hard and expensive to perform. To reduce the amount of testing needed, simulations can be performed. Few commercial programs exist for calculating the synchronization process and Scania used a self-made Excel spreadsheet. The objective of this master thesis was to develop a new MATLAB program for calculating synchronization of gearboxes, with focus on versatility, accuracy and user-friendliness, and verify it by physical testing.

The result is a program with support for more gearboxes than the old spreadsheets. Several new parameters were included to improve accuracy. The tribological contact is represented by using experimentally determined friction properties and material limits, and compare calculated values with the material limits. User-friendliness have been in focus during the entire development, and the program features an extensive manual.

Physical testing shows that the program shows good correlation when the synchronizer is not worn. The accuracy decreases as the wear increases, due to a sudden increase of the coefficient of friction at the end of the synchronization.
Acknowledgements

We would like to thank the staff at Scania, especially Johan Nordkvist and Daniel Jerneborg, for giving us the opportunity to perform this master thesis, and Per Arnelöf, Fredrik Brandt, Jörgen Forsberg, Per Holmlund, Johan Kingstedt, Jorma Koppala, Jonas Lundin, Peer Norberg, Peter Ocklind, Ortwin Schleuter, Håkan Settersjö, Dieter Slapak, Iréne Wahlqvist and Henrik Åström for their knowledge and supporting information, which has been invaluable for our work as well as the rest of Scania NT for the great atmosphere, which made it fun to go to work.

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Finally we would like to thank #censored oil company# for performing the tribological tests needed to determine the friction behaviour of the synchronizers in the oil wipe phase.

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Nomenclature
AMT Automated manual transmission.
BTL Break through load.
Clash Grating sound resulting from an unsuccessful synchronization.
CoF Coefficient of friction.
Direct gear Gear with 1:1 ratio of gears.
Final gear Gear engaged at the end of shift. Also rear axle ratio.
G Gearbox type without split and range.
GR Gearbox with range.
GRS Gearbox with range and split.
GUI Graphical user interface.
Initial gear Gear engaged at the start of shift.
Opticruise Scania’s name for their AMT transmission option.
Ramp speed Engines adaption speed.
RoG Ratio of gear.
RPM Revolutions per minute.
Shift mode How the gear change is performed.
$T_C$ Cone torque or synchronization torque.
$T_1$ Index torque.
1 Background and synchronizer theory

To better understand the background of this master thesis and existing problem with currently performed calculations and design decisions, a literature study was performed before all objectives and limitations were set.

1.1 Scania CV AB

"Scania CV AB’s ["Scania"] objective is to deliver optimized heavy trucks and buses, engines and services, provide the best total operating economy for their customers, and thereby be the leading company in our industry. [ . . . ] Scania operates in some 100 countries and has 34,000 employees. Of these, 2,400 work with research and development – mainly in Sweden, close to the company’s production units. Scania’s corporate purchasing department is supplemented by local procurement offices in Poland, the Czech Republic, the United States and China. Production takes place in Europe and Latin America. In addition, about 20,000 people work in Scania’s independent sales and service organization" ([01] Scania CV AB).

1.2 Project background

Since trucks and trucks’ vehicle-to-driver interfaces are more refined nowadays, gearbox refinements is also required for overall customer satisfaction. Long shift times and high shift effort are not acceptable for new trucks today. Truck powertrains must handle higher torque than car power trains, which makes components larger which increases rotational inertia. This increased inertia makes rotational speed changes harder to perform and places higher demands on involved parts [02][03].

Synchronizers are complex machine elements, and have caused problems such as high shift force and have had reliability issues for a long time. “The manual transmission synchronizer design has been a real challenge and is usually referred to as myth and black magic” ([04] Razzacki, 2004). “As the duration of these phenomena [Abnormal shift reaction force and clash] is very short and the gear-shifting mechanism is complex, it is impossible to figure out the mechanism of these phenomena through experimental approaches” ([05] Hoshino, 1998). Simulation of synchronizers is often performed with self-made software, only a few commercial software solutions exists [06]. Scania currently uses an self-made Excel spreadsheet for synchronization calculations created in 1996, which is not yet verified by physical tests. During this project a new, more user friendly MATLAB program will be developed based on the existing Excel spreadsheet and verified by physical testing.

1.3 Gearbox fundamentals

A gearbox is used to alternate the rotational speed and torque the engine delivers. Different gear ratios are used to achieve higher torque over a wide range of speeds and improve fuel consumption by changing engine RPM at common cruising speeds to the engines efficient RPM range. Both discrete and continuous gear ratios are available, with discrete being the most common [07].

The gearboxes currently used by Scania are of the types G, GR and GRS. G stands for gearbox, GR stands for Gearbox Range and GRS for Gearbox Range Split. There’s also an overdrive gearbox named GRSO, which means the gearbox have a gear with gear ratio < 1. In Scania’s gearboxes, the range and split synchronizers are actuated by air pressure, while the synchronizers in the main gearbox either are actuated by the force applied to the gear lever via a ratio, or by air pressure with the optional “Opticruise” module, which makes the gearbox an AMT gearbox. For a gear shift in a manual gearbox that involves split, main gearbox and range there are a multiple ways of executing the change. The split can change gear whenever the clutch is disengaged but the range can only change gear when the main gearbox is in neutral. Section 4.1.1 - Shift modes explains the different scenarios, both with and without Opticruise. Table 1 shows what positions the split, main gearbox and range is in depending on the current gear. Figure 1 shows a GRS gearbox with first gear selected. The purple ring shows the split, the black ring shows the main gearbox and the green ring shows the range, which is an epicyclic gear. The black and purple rings overlap since gear pair marked H 3 is used for both split and main gearbox. Crawl and reverse gear is also shown. The naming of the gear can differ, but for this master thesis 1-12 is used rather than 1L-6H. Truck gearboxes are massive compared to passenger car gearboxes, e.g. the GRS905 weighs over 300 kg [06].
Table 1. Available gears and gear positions.

<table>
<thead>
<tr>
<th>Main gearbox</th>
<th>LR (Low Range)</th>
<th>HR (High Range)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>LS (Low Split)</td>
<td>HS (High Split)</td>
</tr>
<tr>
<td>LR</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>8</td>
</tr>
<tr>
<td>HR</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>10</td>
</tr>
<tr>
<td>HS</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>12</td>
</tr>
</tbody>
</table>

1.4 Synchronizer objective

The objective of a synchronizer is to synchronize the rotational speed of the input shaft, clutch disc, and the gear to be engaged during gearshifts [04]. The effect is smoother and faster gear changes without clash, a term which will be explained later. Before synchronizers were used, the driver had to “double-clutch”, that is, the driver had to disengage the clutch, disengage the gear, match the engines RPM via a ratio to the output shafts RPM (which is driven by the wheels at approximately constant speed), disengage the clutch, engage the new gear and engage the clutch. This process requires more skill from the driver and does not fit well with the refined and comfortable vehicles existing today [03]. The synchronizer is a mechanical part of the gearbox, but can be controlled by air pressure or electronic actuators.

1.5 Synchronizer components and function

The synchronizer described is the type used by Scania, which is a modified Borg-Warner synchronizer. Due to the difference in required synchronization torque between different gear shifts, both single cone, double cone and triple cone designs are used. Figure 2 shows the seven parts used in a double cone synchronizer assembly, as well as four additional parts for a single cone synchronizer in the opposite side of the driver.

Figure 3 shows a cross section of a complete synchronizer, where the right side is a double cone design and the left side a single cone design. The driver is fixed to a shaft. The latch cone is attached to the driver, but can rotate a few degrees circumferentially around the shaft and move a small distance in the axial direction. A wire spring is located between the driver and latch cone. The shift sleeve, which moves axially outside the driver and is otherwise locked to the driver, energises the wire spring by moving the wire spring axially towards the latch cone. The latch cone, also called blocker ring, makes contact with the middle cone, which in turn makes contact with the inner cone. Note that the inner cone is attached to the latch cone and the driver, and that the middle cone is via a gear ratio connected to the input shaft and clutch disc. The contact surfaces of the outer ring, middle cone and inner cone are conical, and when the axial force from the wire spring is applied, a high normal force arises. If there is a relative rotational speed difference between the inner/outer cones and the middle cone, friction force arise which, since it’s acting around an axis, generates torque. This torque is called cone torque, which will reduce the relative speed difference between the input and output shaft to zero if an axial force is applied to the shift sleeve for a sufficient time. The process is described more thoroughly in section 1.5.2 - The different steps of synchronization.
Figure 2. Synchronizer components [F02].

Figure 3. Synchronizer cross section [F03].
As the name implies, the difference of single cone, double cone and triple cone synchronizers are the number of conical friction surfaces. More surfaces are required for generating the higher synchronization torque that some gear shifts require (while maintaining the same level of axial force and synchronization time), to dissipate the increased energy and to make sure that the cone torque is higher than the index torque, which could be a problem when using chamfers with small angles to index. The other alternatives would be to use materials with higher CoF and better thermal properties or increase the mean cone radius. With higher CoF, the cone angle would have to be increased to ensure self release and prevent ring seizure. The cone angle limit can be expressed as $\mu_C \leq \tan \varphi$. This condition must be satisfied at all times, and larger cone angle would reduce the cone torque [07]. Higher CoF also increases adhesive wear, which introduces particles and asperities that lead to two and three body abrasive wear, which reduces lifetime [08]. Increased mean cone radius would increase the synchronization torque, however gearbox dimensions and sliding speed would increase which is undesirable due to packaging constraints and wear rate.

### 1.5.1 Fundamental notions

Fundamental notions of synchronization include:

- **Cone torque** – The cone torque, also called synchronization torque, is the result of the friction force between the conical surfaces in the synchronizer, which reduces the relative speed difference between the initial and final gear. The cone torque must be higher than the index torque at all times during the synchronization to ensure blocking of axial shift sleeve motion.

- **Index torque** – Index torque arises from the shift sleeve’s chamfered teeth applying an axial force on the latch cones chamfered teeth. Index torque always opposes cone torque. As long as the cone torque is higher than the index torque, the latch cone is locked at an angular displacement of a few degrees behind the driver and shift sleeve in the rotational direction. This misalignment of splines locks the shift sleeve in axial direction until synchronization (of rotational speed) is complete and index torque becomes higher than cone torque, which causes the latch cone and driver/shift sleeve to align.

- **Shift impulse** – Shift impulse is the time integral of the force applied to the gear knob. The shift impulse influences the shift quality and shift feel. The shift impulse needs to be at an “reasonable” level [09].

- **Reflected inertia** – Since the gears mesh with each other, speed change of one component affects other components. The gear ratio alters the rotational inertia in the gearbox. Reflected inertia is the total inertia the synchronizer has to synchronize and is a function of mass, radial distance and gear ratio.

- **Drag** – Drag is friction losses in the system, mainly from bearings and oil churning. The drag torque always slow down the gear train which aids synchronization when the objective is to slow down the gear train to match the rotational speed with the new gear ratio. However the drag increases the synchronizer workload when the objective is to increase the speed of the gear train to match the new gear ratio [07].

### 1.5.2 The different steps of synchronization

The synchronization process can be defined as the following nine steps [10]. However, other definitions exist with both more [11] and less [05] steps, but the main idea is the same. The steps, shown in Figure 4, are:

1. **Disengagement** – disengage the current gear, which is not directly related to the synchronization process.
2. **Neutral** – the synchronization shift sleeve is in neutral position, and parts of the transmission and clutch decelerate due to drag in the gearbox.
3. **Neutral Detent** – the shift sleeve moves past neutral, and starts to energize the wire spring - blocker system, which initiates an oil wipe of the friction surface.
4. **Pre Synchronization** – the axial force transmitted via the wire spring generates cone torque which causes the latch cone to index and reduces the speed difference between the surfaces.
5. **Synchronizing** – Shift sleeve is stationary, speed difference is further reduced due to cone torque.
6. **Synchronization** – When the speed difference approaches zero, the cone torque approaches zero, and the index torque becomes larger than the cone torque. Shift sleeve moves forward.
7. **Blocking Release** – The axial shift sleeve motion forces the latch cone to move circumferentially and the shift sleeve and latch cone teeth align and the shift sleeve can pass. Note that the teeth always will be misaligned during synchronization and at first contact.
8. Engagement Tooth Contact – Shift sleeve continues to move axially and makes contact with engagement teeth. Depending on relative circumferential position, additional force may be required to align the teeth as in step 7, but in this case the teeth relative position is random, and they can be aligned before contact even is made. There are four different scenarios possible, shown in Figure 5, depending on shift sleeve axial speed, relative displacement and relative rotational speed [12].

9. Full Engagement – Shift sleeve locks the driver to the gear, and the shift is complete.

---

Figure 4. Nine steps of synchronization [F04].

Figure 5. The four possible engagement scenarios [F04].
The axial force during synchronization is schematically shown in Figure 6, with the different steps marked.

![Figure 6. Typical shift force at gear lever [N] vs time [ms] with numbered synchronization steps [F02].](image)

1.5.3 Problems during synchronization

Problems that can emerge during synchronization include:

**Clash** – Clash is a common malfunction during synchronization. Full clash is when the synchronizer ring isn’t energised at all, which could be caused by a chain of bad tolerances or poorly designed or possibly worn components that leads to higher index torque than cone torque. Partial clash is when the synchronizer rings have been energised, but there still exists a rotational speed difference. Possible causes are eccentric loading of the shift sleeve which causes the shift sleeve to tilt, damaged chamfers that increase index torque or high drag which slows the gears after synchronization is complete, but before full engagement. Cold clash can occur when the oil is cold and have higher viscosity, which can lead to the oil on the cone not being wiped away from the contact surfaces, and the CoF not reaching its intended value. The symptom of clash is a grating sound during shifting, which is caused by the shift sleeve and gear chamfers making contact while at different rotational speeds. Clash will damage the shift sleeve and gear chamfers [07].

**Hard shifting** – Hard shifting is when a higher force than normal is required to perform the gear change. Hard shifting ranges from slightly higher force than normal due to cold oil to blockout caused by latch cone seizure. In the worst cases the synchronizer rings can be welded to each other, making gear changes impossible [07].

**High shift impulse** – Unacceptably high shift impulse due to insufficient cone torque capacity leads to either long shift time or high shift effort [10].

**Double bump** – Double bump is a common problem of modern gearboxes. The symptom of double bump is that an increased force is required after synchronization is complete. As the name implies, there are two force peaks (“bumps”), but the whole process is so short it’s unlikely the driver will feel two distinct force peaks. The first force peak is to separate the cones, which due to thermal expansion will stick to each other. When the cones heat up, they expand and the axial force can push the outer cone higher on the inner cone. Static CoF also influences the first force peak. The second force peak is the force required for turning the gears, shaft(s) and clutch disc, and is influenced by the chamfer angle and friction properties as well as the relative spline position as shown in Figure 5, which will vary between gear shifts. Double bump is expected to happen, but when the force peaks becomes too high, shift quality and shift feel is diminished [13]. Figure 7 shows the axial force vs. travel with a visible double bump.
1.6 Physical principles of synchronizers

Table 2 shows the main variables used in the physical and mathematical relations.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Parameter</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>µ_B</td>
<td>Static CoF, latch cone – shift sleeve</td>
<td>-</td>
</tr>
<tr>
<td>µ_C</td>
<td>Dynamic CoF, cone</td>
<td>-</td>
</tr>
<tr>
<td>µ_S</td>
<td>Static CoF, wire spring – shift sleeve</td>
<td>-</td>
</tr>
<tr>
<td>A</td>
<td>Index distance</td>
<td>m</td>
</tr>
<tr>
<td>BTL</td>
<td>Break through load</td>
<td>N</td>
</tr>
<tr>
<td>F_AX</td>
<td>Axial force on shift sleeve</td>
<td>N</td>
</tr>
<tr>
<td>F_R</td>
<td>Wire spring force</td>
<td>N</td>
</tr>
<tr>
<td>RoG</td>
<td>Ratio of gear</td>
<td>-</td>
</tr>
<tr>
<td>I_N</td>
<td>Inertia of part</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>I_P</td>
<td>Reflected inertia</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>N_p</td>
<td>Number of plunges</td>
<td>-</td>
</tr>
<tr>
<td>R_B</td>
<td>Blocking chamfer pitch radius</td>
<td>m</td>
</tr>
<tr>
<td>R_C</td>
<td>Mean cone radius</td>
<td>m</td>
</tr>
<tr>
<td>T_C</td>
<td>Cone torque</td>
<td>Nm</td>
</tr>
<tr>
<td>T_D</td>
<td>Total drag torque</td>
<td>Nm</td>
</tr>
<tr>
<td>T_I</td>
<td>Index torque</td>
<td>Nm</td>
</tr>
<tr>
<td>t_s</td>
<td>Synchronization time</td>
<td>s</td>
</tr>
<tr>
<td>Z</td>
<td>Proximity distance</td>
<td>m</td>
</tr>
<tr>
<td>Ø</td>
<td>Cone angle</td>
<td>°</td>
</tr>
<tr>
<td>a_G</td>
<td>Gear acceleration</td>
<td>1/s²</td>
</tr>
<tr>
<td>B</td>
<td>Ramp angle (in shift sleeve)</td>
<td>°</td>
</tr>
<tr>
<td>Θ</td>
<td>Angle of chamfer at pitch radius</td>
<td>°</td>
</tr>
<tr>
<td>ω_F</td>
<td>Final angular velocity</td>
<td>1/s</td>
</tr>
<tr>
<td>ω_I</td>
<td>Initial angular velocity</td>
<td>1/s</td>
</tr>
</tbody>
</table>

In section 1.5.2 - The different steps of synchronization the synchronization process is described, however the synchronizer’s work starts in step three by energising the wire spring – blocker system. In order to set the latch cone into blocking position a wire spring or ball, depending on design, extend the axial force from the gear lever to the latch cone which initiates an oil wipe between the friction surfaces. When the break through load (BTL) is reached the shift sleeve travels the proximity distance (Z) forward to the latch cone and the teeth of the shift sleeve and the latch cone comes in contact. Figure 8 shows the proximity and index (a) distances.
The BTL should start to build as soon as the shift sleeve moves from the neutral position and should stay on until the teeth on the shift sleeve and latch cone comes into contact. BTL drop off prior to the contact point will interrupt the oil wiping and resulting in gear clash due to the low CoF, which leads to low cone torque. On the other hand, BTL continuing beyond the contact point will require higher force on the gear lever. BTL is a function of detent spring rate, wire spring bump or ball height, CoF between the wire spring and shift sleeve and the detent angle of the annulus groove in the shift sleeve [04]. The free body diagram for the shift sleeve/wire spring system is available in Figure 9.

![Figure 8. Proximity and index distances [F05].](image)

![Figure 9. Free body diagram of shift sleeve - wire spring [F06].](image)
Taking sum of forces on the shift sleeve in x- and y-direction, equation (1),(2):

\[ F_A = N_s \cdot \sin \beta + f_s \cdot \cos \beta \]

\[ f_s = \mu_s \cdot N_s \]

\[ F_A = N_s \cdot (\sin \beta + \mu_s \cdot \cos \beta) \quad (1) \]

Taking sum of forces on the wire spring/ball:

\[ F_A = N_s \cdot \sin \beta + f_s \cdot \cos \beta = N_s \cdot \sin \beta + \mu_s \cdot N_s \cdot \cos \beta \]

\[ F_R + f_s \cdot \sin \beta = N_s \cdot \cos \beta \]

\[ F_R = N_s \cdot (\cos \beta - \mu_s \cdot \sin \beta) \quad (2) \]

Substituting \( N_s \) from equation (1) in equation (2) gives equation (3):

\[ F_R = F_A \cdot \left( \frac{\cos \beta - \mu_s \cdot \sin \beta}{\sin \beta + \mu_s \cdot \cos \beta} \right) \]

\[ F_A = F_R \cdot \left( \frac{\sin \beta + \mu_s \cdot \cos \beta}{\cos \beta - \mu_s \cdot \sin \beta} \right) = F_R \cdot \left( \frac{\mu_s + \tan \beta}{1 - \mu_s \cdot \tan \beta} \right) \]

\[ BTL = N_p \cdot F_A = N_p \cdot F_R \cdot \left( \frac{\mu_s + \tan \beta}{1 - \mu_s \cdot \tan \beta} \right) \quad (3) \]

In the fourth to fifth step of the synchronization process, the shift sleeve travels the proximity distance and the blocker teeth come in contact. The axial force generated by the gear lever or a piston is pushing the friction surfaces of the synchronizer together and generating the cone torque which reduces the relative speed between the surfaces. As the latch cone applies an axial force on to the cone, the oil is wiped off the surface and the CoF increases. The cone torque is a function of the axial force applied to the shift sleeve, cone angle, CoF and the mean cone radius \([04]\). Cone torque can be calculated from equation (4), which is derived from the free body diagram in Figure 10.

![Figure 10. Free body diagram of synchronizer ring [F06].](image)

\[ T_C = \frac{F_{AX} \cdot \mu_s \cdot R_C}{\sin \phi} \quad (4a) \]

For a multi-cone design with \( n \) number of cones:

\[ T_C = \sum_{i=1}^{n} F_{AX} \cdot \left( \frac{\mu_i \cdot R_C}{\sin \phi_i} \right) \quad (4b) \]
When the teeth on the shift sleeve pushes on the latch cone teeth an index torque arises, which opposes the cone torque and acts to rotate the latch cone circumferentially so the shift sleeve can pass through. The cone torque must be higher than the index torque until the synchronization is complete to avoid gear clash, as expressed in equation (5) [04].

\[ T_c \geq T_i \] (5)

When the two chamfers between the latch cone and shift sleeve is in contact a friction force is generated. The tangential component of the friction force and the distance from the centre results in the index torque. The index torque is a function of axial force applied to the shift sleeve, the teeth pointing angle, the pitch diameter of the blocking teeth and the CoF between the teeth pointing surface on the shift sleeve and latch cone. The index torque can be derived from the free body diagram of the shift sleeve and latch cone shown in Figure 11.

![Figure 11. Free body diagram of shift sleeve - latch cone [F06].](image)

\[ T_i = F_T \cdot R_B \]

Summation of the forces in y-direction on shift sleeve:

\[ F_{AX} = N_S \cdot (\sin \theta + \mu_B \cdot \cos \theta) \]

\[ N_S = \frac{F_{AX}}{(\sin \theta + \mu_B \cdot \cos \theta)} \]

\[ F_T = F_{AX} \cdot \left( \frac{\cos \theta - \mu_B \cdot \sin \theta}{\sin \theta + \mu_B \cdot \cos \theta} \right) \]

\[ T_i = F_{AX} \cdot R_B \left( \frac{\cos \theta - \mu_B \cdot \sin \theta}{\sin \theta + \mu_B \cdot \cos \theta} \right) = F_{AX} \cdot R_B \cdot \left( \frac{1 - \mu_B \cdot \tan \theta}{\mu_B \cdot \tan \theta} \right) \] (6)

All rotating parts in a gearbox (gears, clutch disc, shafts, bearing races etcetera) have rotational inertia which is a function of mass and radial distance. In order to synchronize the relative speed between the shafts the rotational energy either has to be reduced or increased depending on upshift or downshift in the gearbox. Since the gears are always in mesh, reflected inertia should be used for all calculations. All gears that are in mesh must be included and the total number of items depends on the layout of the transmission. Reflected inertia is the inertia of the parts multiplied by the gear ratio squared [07]. The reflected inertia of a system with n number of components is
When synchronizing, the relationship between cone torque, the reflected inertia and the acceleration of the gears can be described as \[ T_C = I_R \cdot \alpha_G \] (8)

However the gearbox also have a torque resulting from drag in the gearbox which arise from clutch drag \( T_d \), fluid churning \( T_o \) and system friction \( T_f \) as shown in [08][14][15].

\[ T_D = T_d + T_o + T_f \] (9)

The total torque affecting synchronization is \[ T_C = I_R \cdot \alpha_G + T_D \] (10)

The variable \( T_D \) has a negative effect in equation (10) when the shaft has to be slowed down because it is assisted by drag. The opposite is true when the axle needs to increase the speed and the drag resists synchronization. Time of synchronization is a function of how effectively the cone torque is developed, the maximum speed change of the components and drag torque. The rotational acceleration of the gears can be expressed as \[ \alpha_G = \frac{(\omega_f - \omega_i)}{t_s} \] (11)

and the synchronization time is \[ t_s = \frac{(\omega_f - \omega_i) I_R}{(T_C - T_D) \omega} \] (12)

where \( \omega_f \) is the final rotational speed, \( \omega_i \) initial rotational speed, \( I_R \) reflected inertia, \( T_C \) cone torque resulting from axial force and \( T_D \) torque resulting from drag.

It’s important to note that the variable \( T_D \) in equation (12) has a positive value when the axle is slowed down, since drag slows the gearbox down and less work has to be performed by the friction surfaces. The power developed in the friction surfaces is \[ P = T_C \cdot \omega \] (13)

which is a function of the torque and the angular velocity.

1.7 Materials on friction surface

Desired material properties for the friction surface of the synchronizer rings include good wear resistance, predictable friction behaviour, lower static CoF than dynamic CoF, good thermal conductivity and low temperature dependence regarding material hardness and thermal expansion. For heavy-duty applications like Scania’s truck synchronizers, two different materials are used today. The most commonly used is Molybdenum, in Scania’s case min 99.95% Mo, which is applied to the cone surface by a flame-spray process [17]. The other material is Carbon fibre, which is becoming more and more popular. Brass is used for less demanding applications, but not by Scania. Table 3 shows a comparison of material properties. For the CoF values, Texaco 1874 MTX oil is used [03][18][19].
Table 3. Material properties comparison.

<table>
<thead>
<tr>
<th>Property / Material</th>
<th>Molybdenum</th>
<th>Carbon fibre</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wear resistance</td>
<td>Lower</td>
<td>Higher</td>
</tr>
<tr>
<td>Variation in CoF</td>
<td>Higher</td>
<td>Lower</td>
</tr>
<tr>
<td>Static CoF</td>
<td>~0.1</td>
<td>~0.09</td>
</tr>
<tr>
<td>Dynamic CoF</td>
<td>~0.08</td>
<td>~0.11</td>
</tr>
<tr>
<td>Temperature dependence</td>
<td>Low</td>
<td>Low</td>
</tr>
</tbody>
</table>

Higher dynamic CoF generates higher cone torque, which leads to shorter shift times for a given shift impulse but develops higher power. Lower static CoF reduces the force needed to mesh at the end of the synchronization. High wear resistance and low variations in the CoF increases the consistency between gear shifts.
2 Master thesis definition

The thesis was scheduled to run for 23 weeks at Scania Technical Centre in Södertälje, Sweden. Master thesis examiner was Prof. Roland Larsson, Department of Applied Physics and Mechanical Engineering, division of Machine Elements at Luleå University of Technology. Master thesis supervisor was Daniel Jerneborg, senior engineer at Scania NTBM – Manual Gearbox Manoeuvring.

2.1 Master thesis objectives

The main objective was to develop an improved computer program for evaluation of design parameters of gearbox synchronizers, tailored for the gearboxes Scania currently uses or might use in a near future. The program should be written in MATLAB and focus on user-friendliness. In order to accomplish the task, the main objective was divided into six sub objectives to have smaller, more reachable objectives. If the sub objectives were fulfilled the main objective would also be reached. The six sub objectives were:

- Gain better understanding of gearbox synchronizers and try to find answers to Scania’s questions.
- Analyze currently used program and compare with analytic calculations.
- Gain better understanding of MATLAB and GUI design.
- Investigate different methods for data input and post-processing.
- Allow for future modification and improvements.
- If possible, verification by physical testing in lab environment.

2.2 Planning and limitations

This project focuses on gearbox synchronizers of the type mainly used by Scania. At the project start, the test rig for physical verification had not arrived at Scania. It was therefore not certain that physical testing would be performed. However, time for the testing was included in the time plan. The time plan consists of the following eight points:

- Introduction – Gain basic knowledge of Scania as a company and synchronizers in general.
- Literature study – Gain greater knowledge of synchronizers, focusing on the type used by Scania, as well as program and simulations already performed.
- Planning – Make the project time plan and set project objectives.
- Analysis of current program – thorough analysis of current programs theory, features and limitations.
- MATLAB – Gain greater knowledge of MATLAB.
- Program outline – Make a basic program outline containing features and methods that will be used.
- MATLAB programming – Develop the program in MATLAB.
- Physical testing – If possible, physical testing will be performed.

2.3 Questions from Scania

Project supervisor Daniel Jerneborg asked the following questions during the project start-up that needed investigation [06]:
- Does the absolute rotational speed matter, or only the relative rotational speed? E.g. is 1000 RPM to 500 RPM the same as 2000 RPM to 1500 RPM?
- How does synchronization time affect wear?

2.4 Results – Literature study and project definition

The literature study led to greatly increased knowledge about the synchronization process. It also resulted in a basic knowledge of the entire gearbox and it’s mechanics, and is a solid foundation for future work.

No evidence were found regarding absolute rotational speed directly affecting wear during synchronization. The synchronization time will affect the wear because of the required higher axial force to decrease the synchronization.
time, which leads to a higher surface pressure and higher power. According to the p-v diagram this could lead to higher wear rate [19]. The temperature differences and effect would also have to be investigated for a final answer.

2.5 Discussion – Literature study and project definition

Most papers found referred to Socin and Walters’ “Manual Transmission Synchronizer”, published in 1968. Newer publications mostly concern simulations or materials. While “simulation” sounds interesting for this project, it’s rarely any specific information about the simulations other than that they show good correlation with experimental data. Little is written about how the simulation is performed. “Materials” show that it’s important to make the program support different material behaviours, which is mostly related to the CoF. The goal was to support a variable CoF, something that is not available in the program previously used by Scania.

Regarding the questions asked by project supervisor Daniel Jerneborg, it’s likely that the temperature of the friction surface will be lower with increasing synchronization time due to higher thermal conduction into the bulk of the ring, and with higher absolute rotational speed due to better cooling from increased oil contact and higher heat transfer to the environment due to higher Reynolds number, which increases forced convection heat transfer. Disregarding drag, the synchronizer has to perform the same work. It has been shown that the wear of brass synchronizers increases significantly when the temperature increases, but that can probably be attributed to the fact that the hardness of brass is reduced by up to 50% going from 200 °C to 300 °C [18]. Molybdenum and carbon fibre retains its hardness in the relevant temperature interval [20][21].
3 Analysis and verification of currently used program

The currently used program for synchronization calculations consists of two spreadsheets in Excel, schematically shown in Appendix 1. One of the sheets are for inputs and one for outputs. In the input sheet the following parameters are defined:

- Gearbox type.
- Number of cones.
- Geometrical dimensions.
- CoF.
- Estimated fraction of area in contact.
- Forces and force gradients.
- Lever ratio.
- Number of gears per shift.
- Max engine RPM.
- Number of gear teeth.
- Rotational inertia of components.
- Drag of components.

For “Gearbox type”, “G”, “GS”, “GR” and “GRS” are valid inputs according to the spreadsheet, but only GRS has working calculations. If-cases like “if ‘GRS’ and ‘gear exists’ then ‘calculation’ else ‘0’” is often used. Estimated fraction of area in contact accounts for the surface macro roughness and angular misalignments, and is per experience estimated to 50% [06]. Max engine RPM is the RPM to initiate an upshift and finish a downshift at.

The output sheet consists of three parts, one part for static, time independent calculations, one part for time-discrete calculations and one part to recap the tabulated values of the time-discrete calculations. The static calculations determine:

- Cone area.
- Lowest possible CoF for successful synchronization.
- Maximum allowed power.
- Maximum cone- and index torque.
- Margin of safety for torque and power.
- Required force on lever for indexing a static gearbox.
- Reflected inertia.
- Reflected total drag for warm and cold gearbox.
- Surface pressure.
- Gear ratio.

The time-discrete calculations uses a time step of 0.005 s and presents the data in large tables. The calculations determine:

- Instantaneous cone torque.
- Instantaneous RPM difference.
- Instantaneous power dissipated in the friction surface.
- Total work up to current time step.

3.1 Limitations in currently used spreadsheets

The biggest limitation of the spreadsheets is that it isn’t easily editable and have no documentation of how the calculations is made. The program only supports GRS type gearboxes with 12 gears. Most of the static calculations support more than three gears in the main gearbox, while the time-discrete only supports the 12 gears of a GRS gearbox and triple shifts, e.g. 1-4 or 8-5. Triple shifts are used since that is the largest realistic RPM difference achievable with the current engines, and therefore most synchronization work is required. The spreadsheets only
supports one kind of shift sequence where the main gearbox moves into neutral, split and range changes gear and then main gearbox changes gear. 0.005 s is the only available time step. Only a constant CoF is supported, while experiments shows that the initial CoF is lower than the average CoF. The drag is static in the spreadsheet, while experiments shows that it depends on revolution speed, oil level and temperature, but temperature and oil level can be considered constant during a gear shift.

3.2 Corresponding formulas

The equations in this section were theoretically derived and correspond with the equations used in the Excel spreadsheet. Most of the equations presented in section 1.6 - Physical principles of synchronizers are also implemented in Excel and uses the instantaneous values for calculating time-discrete changes in e.g. cone torque. Table 4 shows the variables used and what the notations stands for. E.g. RoGIM and RoGFM stands for ratio of gear at initial main (the gear ratio of the gear in the main gearbox before the shift is initiated) and ratio of gear at final main (the gear ratio of the gear in the main gearbox after the shift).

Table 4. Variables used in the analysis.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Parameter</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>µCmin</td>
<td>Minimum CoF, cone</td>
<td>-</td>
</tr>
<tr>
<td>µsf</td>
<td>CoF, selector finger – sleeve</td>
<td>-</td>
</tr>
<tr>
<td>AC</td>
<td>Cone area</td>
<td>m²</td>
</tr>
<tr>
<td>AF</td>
<td>Fraction of area in contact</td>
<td>-</td>
</tr>
<tr>
<td>CW</td>
<td>Cone width</td>
<td>m</td>
</tr>
<tr>
<td>FCi</td>
<td>Force buildup rate</td>
<td>N/s</td>
</tr>
<tr>
<td>FCmax</td>
<td>Maximum cone force</td>
<td>N</td>
</tr>
<tr>
<td>FGl</td>
<td>Gear lever force</td>
<td>N</td>
</tr>
<tr>
<td>IA</td>
<td>Inertia, annulus</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>IC</td>
<td>Inertia, clutch</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>IS</td>
<td>Inertia, input shaft</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>ICs</td>
<td>Inertia, lay shaft</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>ISs</td>
<td>Inertia, main shaft</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>IR(High range)</td>
<td>Reflected inertia, high range</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>IR(Low range)</td>
<td>Reflected inertia, low range</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>IR(Main gearbox step 3 high split)</td>
<td>Reflected inertia, main gearbox gear 3 high split</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>IR(Main gearbox)</td>
<td>Reflected inertia, main gearbox</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>IR(Split)</td>
<td>Reflected inertia, split</td>
<td>Kg·m²</td>
</tr>
<tr>
<td>RL</td>
<td>Lever ratio</td>
<td>-</td>
</tr>
<tr>
<td>m</td>
<td>Oil viscosity-temperature slope</td>
<td>-</td>
</tr>
<tr>
<td>PC</td>
<td>Dissipated power in the cone</td>
<td>W</td>
</tr>
<tr>
<td>P Cmax</td>
<td>Maximum allowed cone power</td>
<td>W</td>
</tr>
<tr>
<td>P Cmax/area</td>
<td>Maximum allowed power/area in the cone</td>
<td>W/m²</td>
</tr>
<tr>
<td>RoG(Total)</td>
<td>Ratio of gear, total final</td>
<td>-</td>
</tr>
<tr>
<td>RoGFM</td>
<td>Ratio of gear, final main</td>
<td>-</td>
</tr>
<tr>
<td>RoGR</td>
<td>Ratio of gear, final range</td>
<td>-</td>
</tr>
<tr>
<td>RoGFS</td>
<td>Ratio of gear, final split</td>
<td>-</td>
</tr>
<tr>
<td>RoG(Total)</td>
<td>Ratio of gear, total initial</td>
<td>-</td>
</tr>
<tr>
<td>RoGIM</td>
<td>Ratio of gear, initial main</td>
<td>-</td>
</tr>
<tr>
<td>RoGIN</td>
<td>Ratio of gear, input gear</td>
<td>-</td>
</tr>
<tr>
<td>RoGIR</td>
<td>Ratio of gear, initial range</td>
<td>-</td>
</tr>
<tr>
<td>RoGIS</td>
<td>Ratio of gear, initial split</td>
<td>-</td>
</tr>
<tr>
<td>RoGL</td>
<td>Ratio of gear, low range</td>
<td>-</td>
</tr>
<tr>
<td>RoGS</td>
<td>Ratio of gear, split</td>
<td>-</td>
</tr>
<tr>
<td>RPM</td>
<td>Max engine RPM</td>
<td>1/s</td>
</tr>
<tr>
<td>RS</td>
<td>Radius shift sleeve</td>
<td>m</td>
</tr>
</tbody>
</table>
### The axial force is the lowest value of equation (14a) and (14b) at every time step.

\[ F_{AX} = F_{ci} \cdot \text{time} \]  \hspace{1cm} (14a)

\[ F_{AX} = F_{Cmax} \]  \hspace{1cm} (14b)

Maximal power the cone surface can withstand is

\[ P_{Cmax} = A_C \cdot P_{Cmax/area} \]  \hspace{1cm} (15)

where \( A_C \) is the cone area in contact and \( P_{Cmax/area} \) is the value of how much power the surface can withstand per area unit.

The power generated at the friction surface is

\[ P_c = (T_c \pm T_d) \cdot \Delta F_S \cdot \frac{\pi}{30} \]  \hspace{1cm} (16)

where the factor \( \pi/30 \) is to convert RPM to rad/s.

The surface pressure in the cone is

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>SP&lt;sub&gt;C&lt;/sub&gt;</td>
<td>Surface pressure in the cone</td>
<td>Pa</td>
</tr>
<tr>
<td>S&lt;sub&gt;S&lt;/sub&gt;</td>
<td>Step size</td>
<td>%</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
<td>K</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(C)&lt;/sub&gt;</td>
<td>Drag, clutch</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(High range)&lt;/sub&gt;</td>
<td>Drag, high range</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(IS)&lt;/sub&gt;</td>
<td>Drag, input shaft</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(Low range)&lt;/sub&gt;</td>
<td>Drag, low range</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(LS)&lt;/sub&gt;</td>
<td>Drag, lay shaft</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(Main gearbox gear 3 high split)&lt;/sub&gt;</td>
<td>Drag, main gearbox gear 3 high split</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(Main gearbox)&lt;/sub&gt;</td>
<td>Drag, main gearbox</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(MS)&lt;/sub&gt;</td>
<td>Drag, main shaft</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(Range)&lt;/sub&gt;</td>
<td>Drag, range</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(Selector finger)&lt;/sub&gt;</td>
<td>Drag, selector finger</td>
<td>Nm</td>
</tr>
<tr>
<td>T&lt;sub&gt;D(Split)&lt;/sub&gt;</td>
<td>Drag, split</td>
<td>Nm</td>
</tr>
<tr>
<td>Time</td>
<td>Current time</td>
<td>s</td>
</tr>
<tr>
<td>W</td>
<td>Work</td>
<td>J</td>
</tr>
<tr>
<td>W (oil)</td>
<td>Function of kinematic viscosity</td>
<td>-</td>
</tr>
<tr>
<td>Δ&lt;sub&gt;F_S&lt;/sub&gt;</td>
<td>RPM difference in friction surface</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;main&lt;/sub&gt;</td>
<td>Initial RPM difference, main gearbox</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;main-down&lt;/sub&gt;</td>
<td>Initial RPM difference, main gearbox downshift</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;main-down-range&lt;/sub&gt;</td>
<td>Initial RPM difference, main gearbox downshift range</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;main-up&lt;/sub&gt;</td>
<td>Initial RPM difference, main gearbox upshift</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;main-up-range&lt;/sub&gt;</td>
<td>Initial RPM difference, main gearbox upshift range</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;Range&lt;/sub&gt;</td>
<td>Initial RPM difference, range</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;Range-down&lt;/sub&gt;</td>
<td>Initial RPM difference, range downshift</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;Range-up&lt;/sub&gt;</td>
<td>Initial RPM difference, range upshift</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;Range-down&lt;/sub&gt;</td>
<td>Initial RPM difference, synchronizer surface range</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;Range-up&lt;/sub&gt;</td>
<td>Initial RPM difference, synchronizer surface range</td>
<td>1/s</td>
</tr>
<tr>
<td>ΔRPM&lt;sub&gt;Split&lt;/sub&gt;</td>
<td>Initial RPM difference, split</td>
<td>1/s</td>
</tr>
</tbody>
</table>
\[ SP_c = \frac{F_{AX}}{\sin \phi \cdot A_c} \]  

(17)

The limit for the CoF in the cone surface to avoid clash due to a larger index torque than cone torque before the relative speed between the parts being synchronized is zero is

\[ \mu_{C_{\text{min}}} = \frac{Rb \cdot (1 - \mu_b \cdot \tan \phi)}{\mu_c \cdot \tan \phi} \]  

(18)

Synchronization work performed by the conical surfaces is

\[ W = \int p_c \, dt \]  

(19)

The step size of the gearbox is

\[ S_S = \frac{ROG_{I(\text{total})}}{ROG_{F(\text{total})}} - 1 \]  

(20)

where ROG_{I(\text{total})} is the initial ratio of gear through the whole gearbox and ROG_{F(\text{total})} is the final ratio of gear through the whole gearbox.

The reflected inertia to the split, main gearbox or the range depending on what gear change is performed is described by

\[ I_{R(\text{split})} = I_c + I_{IS} \]  

(21)

\[ I_{R(\text{main gearbox step 3 high split})} = I_c + I_{IS} + I_{LS} \cdot ROG_{FM}^2 \]  

(22)

\[ I_{R(\text{main gearbox})} = (I_c + I_{IS}) \cdot (ROG_{S} \cdot ROG_{FM})^2 + I_{LS} \cdot ROG_{FM}^2 \]  

(23)

where the I’s are the inertia of the parts and the RoG’s the of gears.

### 3.3 Differing formulas

The calculated initial RPM difference for the gearbox does not correspond with the formulas used in the Excel spreadsheet. In the Excel spreadsheets, different equations describing the RPM difference in the main gearbox are used depending on if the range is involved or not, and then a constant is added or subtracted. The constant is hand calculated to simulate the shift mode where the main gearbox is in neutral when the split changes gear and both the input shaft and the lay shaft will alter its RPM. Because the Excel spreadsheets only supports one shift mode and a GRS905 gearbox with 3 step gear changes a new set of equations was derived to cover all relevant ways to change gear. The theory and an example of how the equations were derived are presented in section 4.1.2 - Initial RPM difference and the complete set of equations are presented in Appendix 2. Equations (24) to (30) are the formulas that were used in the Excel spreadsheets. The RPM difference in the split is

\[ \Delta RPM_{split} = RPM_M \cdot \left(\frac{ROG_{FS}}{ROG_{IS}} - 1\right) \]  

(24)

where the RPM_M is the maximal RPM aimed for in a downshift or the initial RPM in an upshift, ROG_{IS} the ratio of gear for the split in its initial position and ROG_{FS} the ratio of gear for the split in its final position.

The RPM difference in the range for upshifts is

\[ \Delta RPM_{range, up} = \frac{RPM_M}{ROG_{IS} \cdot ROG_{TM}} \]  

(25)
The RPM difference in the range for downshifts is

$$\Delta RPM_{\text{range\_down}} = \frac{RPM_M}{RoG_{FS}RoG_{FM}RoG_{FR}}$$  \hspace{1cm} (26)

The RPM difference in the main gearbox depending on if it’s an upshift or downshift and if the range is involved in the gear change is

$$\Delta RPM_{\text{main\_up}} = \frac{RPM_M}{RoG_{IS}RoG_{FM}} - \frac{RPM_M}{RoG_{IS}RoG_{IM}} + \text{constant}$$  \hspace{1cm} (27)

$$\Delta RPM_{\text{main\_up\_range}} = \frac{RPM_M}{RoG_{IS}RoG_{FM}} - \frac{RPM_M}{RoG_{IS}RoG_{IM}RoG_{IR}} + \text{constant}$$  \hspace{1cm} (28)

$$\Delta RPM_{\text{main\_down}} = \frac{RPM_M}{RoG_{FS}RoG_{FM}RoG_{FR}} - \frac{RPM_M}{RoG_{FS}RoG_{FM}RoG_{FR}RoG_{IR}} + \text{constant}$$  \hspace{1cm} (29)

$$\Delta RPM_{\text{main\_down\_range}} = \frac{RPM_M}{RoG_{FS}RoG_{FM}RoG_{FR}} - \frac{RPM_M}{RoG_{FS}RoG_{FM}RoG_{FR}RoG_{FM}} + \text{constant}$$  \hspace{1cm} (30)

The input and calculations for the drag and reflected inertia differs from the Excel spreadsheets, the theory and formulas that is used in the new program is described in detail in section 4.1 - Additional parameters. The drag that affects the split synchronizer during a gear change is

$$T_{D(Split)} = T_{D(C)} + T_{D(IS)}$$  \hspace{1cm} (31)

where $T_{D(C)}$ is the drag from the clutch and $T_{D(IS)}$ drag from the input shaft. The drag affecting the synchronizers in the main gearbox except for the case where the high split is in gear and the main gearbox is changing to third gear is

$$T_{D(\text{Main\_gearbox})} = \left( (T_{D(C)} + T_{D(IS)}) \cdot RoG_{FS} + T_{D(IS)} \right) \cdot RoG_{FM}$$  \hspace{1cm} (32)

The drag affecting the synchronizers in the main gearbox for the case where the high split is in gear and the main gearbox is changing to third gear is

$$T_{D(\text{Main\_gearbox}\_\text{gear 3 high split})} = T_{D(C)} + T_{D(IS)} + \frac{T_{D(IS)}}{RoG_{FM}}$$  \hspace{1cm} (33)

The drag that affects the synchronizer at the range depending on high range or low range are

$$T_{D(\text{low\_range})} = (RoG_{LR} - 1)^2 \cdot T_{D(MS)}$$  \hspace{1cm} (34)

$$T_{D(\text{high\_range})} = T_{D(MS)}$$  \hspace{1cm} (35)

The reflected inertia depending on high or low range are

$$I_{R(\text{Low\_range})} = I_{A} + I_{MS} \cdot (RoG_{LR} - 1)^2$$  \hspace{1cm} (36)

$$I_{R(\text{High\_range})} = I_{MS} + \frac{I_{A}}{(RoG_{LR}-1)^2}$$  \hspace{1cm} (37)

The calculated force required on the gear lever to index a static gearbox, equation (38a), differs from the derived in equation (38b). The difference is that the CoF is set to zero and the BTL is added in equation (38a), in equation (38b) the friction force is included which arises from the wire spring when it slides against the shift sleeve. In order to index the gearbox the BTL have to be exceeded for the shift sleeves chamfered surface to come in contact with the latch cones chamfered surface. The equation which gives the highest value of equation (38b) and (38c) should be used. If the CoF is hard to determine, CoF = 0 can be used since that’s the worst case scenario.
In the Excel spreadsheets the cone angle isn’t considered in the calculation of the active cone area according to

\[ A_c = A_F \cdot 2 \cdot \pi \cdot (R_{C1} \cdot C_W1 + R_{C2} \cdot C_W2 + R_{C3} \cdot C_W3) \]  

(39)

In the derived theoretical formula used in the new program the active cone area is

\[ A_c = A_F \cdot 2 \cdot \pi \cdot \left( R_{C1} \frac{C_{W1}}{\cos \phi_1} + R_{C2} \frac{C_{W2}}{\cos \phi_2} + R_{C3} \frac{C_{W3}}{\cos \phi_3} \right) \]  

(40)

The if-case for determining if the spreadsheet should calculate the RPM difference contains an incorrect parenthesis placement, which causes the spreadsheet to calculate for one time step longer than intended, which suggests a slight overshoot in the synchronization.

3.4 Example of an synchronization calculation

After all inputs are defined in the Excel spreadsheet the calculations are performed. All the static values is displayed in the output sheet, see section 3 - Analysis and verification of currently used program for the inputs and outputs. For the static calculations in the excel spreadsheet the following equations are used: (15), (17), (18), (20) to (37), (38a), (39).

The time discrete calculations are more complex. The first calculations determine the initial RPM difference, the reflected inertia and the drag according to equations (21) to (37). The drag is either assisting or counteracting the synchronizing torque which is a function of the axial force, equation (14a) and (14b), and the synchronizers’ design, equation (4b). In the first time step, only the drag affects the RPM difference according to equation (12), since the axial force is zero at the beginning of the synchronizing process. The power is also calculated in each time step accordingly to equation (13) and the work is the time integral of the power, equation (19). Once power and work is determined, the RPM difference for the next time step is calculated. These calculations continues iteratively until the RPM difference is zero. Figure 12 shows a flowchart over the calculations in the Excel spreadsheets, where \( K \) is a constant to convert rad/s to RPM.

![Flowchart over calculations in Excel spreadsheet.](image)

Figure 12. Flowchart over calculations in Excel spreadsheet.
3.5 Results – analysis of current spreadsheets

The analysis of the spreadsheets and the verification with theoretically derived formulas led to increased knowledge of how different components of the synchronizer pack and gearbox affect each other. The analysis also resulted in more ideas on how to improve the user-friendliness.

The calculations in the spreadsheets shows good correlation with the theoretically derived formulas except for:

- RPM difference of the main gearbox.
- Formulas for calculating the drag and reflected inertia for the range.
- The calculation of the active cone area.

3.6 Discussion – analysis of current spreadsheets

A major criticism regarding the current spreadsheets is that it is not easily editable and not user-friendly enough. These problems are partly the result of the abundant use of if-cases. Most of the cells derived from other cells starts with a long if-case verifying that there is a gear available for the current calculation. A better solution would have been to have two or more output sheets, one for each different type of gearbox.

It’s likely that the old spreadsheets never were finished, since there are obvious hints of integrating calculations for both “G”, “GS”, “GR” and “GRS” gearboxes with different number of gear combinations, even though the majority of the spreadsheets only supports the current 12-speed GRS gearbox. Another evidence showing that the spreadsheets was left unfinished is that support for single, double and triple shift seems to be planned, but only triple-shift is available. If the goal was to only use triple-shifts for a GRS gearbox, much of the cluttered if-cases could have been eliminated. Limitations in Excel forcing all calculated values for each time step and gear shift to be tabulated also degrades the user-friendliness; diagrams alone would be a better way to present the data.

The documentation is almost non-existent, which could be because the spreadsheets never were finished. Another explanation is that the spreadsheets could be made by the only person that would have used it at the time, and therefore documentation may have seemed unnecessary and even a waste of time. Since the spreadsheets hasn’t been replaced for 14 years, new users are using the spreadsheets now.

The calculations correspond well with the newly derived theoretical calculations, except for the initial RPM differences, reflected inertia and drag. Because of the criterion that the new program should be more flexible and cover several shifting scenarios and different gearboxes a new set of equations for the initial RPM difference, reflected inertia and drag was derived.

A time step of 0.005 s is probably acceptable, but the spreadsheets was created in 1996 and with the increased computational power of modern computers, a finer or adjustable time step would be preferable.

The major calculation limitations found in the spreadsheets was that a variable CoF was not supported and that the drag is not RPM dependant. A constant CoF excludes the effect of oil-wiping and the advantage Carbon fibre theoretically has over Molybdenum when rotating the latch cone due to lower static CoF than dynamic CoF. The duration and effect of the oil wiping is also excluded, which suggests that the synchronization will be finished faster.
4 New program

The objective was to make the new program more accurate, versatile and user friendly, both to use and to edit. In order to improve accuracy and versatility, several new parameters were implemented as described in section 4.1 - Additional parameters. The program was also made to support different shift scenarios and G, GR and GRS gearboxes with three to six gears in the main gearbox. All input values and output requests are easily changeable by the user via a GUI, which makes the program user friendly. The program is written to be editable by a person with basic MATLAB knowledge.

4.1 Additional parameters

To make the new program more flexible and more accurate several formulas and parameters used in the Excel spreadsheets had to be modified or redone.

4.1.1 Shift modes

Due to the fact that a shift in the GRS gearbox can be performed in more than one way, seven shift modes are possible to choose from. The shift modes only involves the split and main gearbox because the range only changes gear when the main is in neutral. This applies for all current gearbox models. The different shift modes that are possible to perform are:

1. “Manual (split when main in neutral)”, the program simulates a gear change where the main gearbox is set into neutral, then the split changes gear and after that the main changes gear. In this case both the input shaft and the lay shaft will change its rotational speed until the relative rotational speed is zero. How the input shaft and the lay shaft will alter their rotational speed is affected by the cone torque, drag from the selector finger, the rotational speed depending drag on both shafts and their reflected rotational inertia.
2. “Manual (split first)”, the program simulates a gear change where the split changes gear first and then the main changes gear. The main shaft is driven by the wheels (with approximately infinite inertia) and will not change its rotational speed unless the range is shifting gear. When the split changes gear the lay shaft is driven by the main shaft and will not change its rotational speed. When the main gearbox changes gear it will change both the lay shaft and the input shaft’s speed.
3. “Manual (split last)” same as above except for the main gearbox changes gear before the split.
4. “Manual (main when split in neutral)” the main gearbox changes gear when the split is in neutral, the lay shaft’s rotational speed is changed while the input shaft’s rotational speed is constant. When the split changes gear it will adapt to the lay shaft’s rotational speed.
5. “Opticruise 2-pedal (clutch closed)”, the clutch will be closed for every gear change except when only the split changes gear. In that case the clutch will open, the split will change gear, the engine match the RPM of the input shaft and then close the clutch. In all the other cases the split and range will change gear when main is in neutral and clutch closed. The engine will adapt its RPM to the calculated RPM [22].
6. “Opticruise 3-pedal”, the split and the range will only change gear when the main gearbox is in neutral. The engine will adapt its RPM to the calculated RPM after the complete change [22].
7. “Opticruise 2-pedal (clutch open)”, the clutch is open during the whole gear change. The main gearbox has to be in neutral for the split and range to change gear. This mode is the same as shift mode 1 [22].

All the above shift modes are possible but some are less likely to occur in a real life situation. In the Opticruise models the gear change in the split, range and main gearbox are actuated by pneumatic pistons. In the manual version only the split and range are pneumatically actuated, the difference is that they are initiated by the driver. There are two different models of the Opticruise: the 2-pedal version and the 3-pedal version. In the 2-pedal version the clutch is also automated and controlled by a computer, while in the 3-pedal version the clutch is controlled by the driver but is only used when starting the vehicle or when coming to a complete stop. The engines ramp speed, how fast it changes its rotational speed, is computer controlled depending on if the gear change is set to be comfortable or fast. The pneumatic pistons in the gearbox have the same force build up independent of the gear change but differs depending on if it’s the range, split or main gearbox.
4.1.2 Initial RPM difference

In order to calculate the initial RPM difference at the synchronizers for different shift scenarios for G, GR and GRS gearboxes a large set of equations were derived. Below is an example on how the equations were derived for an upshift in a gearbox with a split, main and range with the following shift scenario: Split changes gear, main into neutral, range changes gear, main into gear. See Appendix 2 for a complete set of equations for different scenarios and types of gearboxes. Figure 13 shows the RPM of the different parts before and after the change for an GRS gearbox performing an upshift. RPM is the initial RPM when initiating the gear change, RoG stands for ratio of gear and the subscript after RoG stands for initial or final and the gearbox part (split, main and range). E.g. RoGIM is ratio of gear between the lay shaft and main shaft before the shift is initiated.

![Diagram of RPM changes](image)

**Figure 13. RPM at different places in a GRS gearbox.**

The main shaft is via the planetary gear driven by the wheels (with approximately infinite inertia) and the lay shaft is connected to the main shaft. When the split changes gear its synchronizer have to alter the rotational speed of the input shaft to synchronize the relative speed between the lay shaft and the input shaft

\[
\Delta RPM_{split} = \left[ \frac{RPM_M \cdot RoG_{FS}}{RoG_{IS}} \right] - RPM_M 
\]  

(41)

When the range changes gear, the main gearbox has to be in neutral (this applies for all range shifts), the range synchronizer will alter the rotational speed of the main shaft (which is disconnected from the lay shaft) as

\[
\Delta RPM_{range} = \frac{RPM_M}{RoG_{IS} \cdot RoG_{IM} \cdot RoG_{IR}} - \Delta RPM_{s, range} 
\]  

(42)

The relative speed the range synchronizer surfaces experiences is

\[
\Delta RPM_{s, range} = \frac{RPM_M}{RoG_{IS} \cdot RoG_{IM}} 
\]  

(43)
At the final stage of the shift, “main into gear“, the synchronizer on the main shaft have to alter the rotational speed on the lay shaft and the input shaft

\[
\Delta RPM_{\text{main}} = \left[ \frac{RPM_M \cdot \text{RoG}_{FR}}{\text{RoG}_{IN} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] - \left[ \frac{RPM_M}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}} \right]
\]  

(44)

The \( \Delta RPM_{\text{s_range}} \) with the “s_” subscript is used to display if it’s the relative RPM difference between the synchronizer surfaces or if it’s how much the shaft in front of the synchronizer is altering its RPM, \( \Delta RPM_{\text{range}} \), e.g. range synchronizer alters main shaft’s RPM.

The following equations are adapted GRS formulas to describe a GR gearbox, which lacks the split. \( \text{RoG}_{IN} \) is the gear ratio of the input gear pair. The \( \text{RoG}_{DM} \) (Ratio of Gears when going into the direct gear) is considered to be \( 1/\text{RoG}_{IN} \) to reduce the number of special cases needed. Figure 14 shows how the power is transferred in the real case to the left, and with the simplification to the right.

The RPM difference for the main gearbox depending on up- or downshift are

\[
\Delta RPM_{\text{main_up}} = \left[ \frac{RPM_M \cdot \text{RoG}_{FR}}{\text{RoG}_{IN} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] - \left[ \frac{RPM_M}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}} \right] = 
\]

\[
= \left[ \frac{RPM_M}{\text{RoG}_{IN}} \right] \cdot \left[ \frac{\text{RoG}_{FR}}{\text{RoG}_{IM} \cdot \text{RoG}_{IR}} - \frac{1}{\text{RoG}_{FM}} \right]
\]

\[
\Delta RPM_{\text{main_down}} = \left[ \frac{RPM_M \cdot \text{RoG}_{IR} \cdot \text{RoG}_{IM}}{\text{RoG}_{IN} \cdot \text{RoG}_{FM} \cdot \text{RoG}_{FR} \cdot \text{RoG}_{FM}} \right] - \left[ \frac{RPM_M}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}} \right] = 
\]

\[
= \left[ \frac{RPM_M}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}} \right] \cdot \left[ 1 - \frac{\text{RoG}_{IR} \cdot \text{RoG}_{IM}}{\text{RoG}_{FR} \cdot \text{RoG}_{FM}} \right]
\]

(45)

(46)

The RPM difference in the range depending on up- or downshift and if it’s what the synchronizer surfaces experience or how the shaft alter its speed are

\[
\Delta RPM_{\text{s_range_up}} = \frac{RPM_M}{\text{RoG}_{IN} \cdot \text{RoG}_{IM}}
\]  

(47)

\[
\Delta RPM_{\text{range_up}} = \left[ \frac{RPM_M}{\text{RoG}_{IN} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] - \left[ \frac{RPM_M}{\text{RoG}_{IN} \cdot \text{RoG}_{IM}} \right] =
\]

\[
\frac{
}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}}
\]
There is a shift scenario where the initial RPM difference in the gearbox isn’t possible to calculate by static calculation for all the involved parts; when the main gearbox is in neutral when split changes gear. The split and ranges initial RPM difference can always be calculated static. When the main gearbox is in neutral the lay shaft is disconnected to the main shaft and when the split changes gear the initial RPM difference in the split synchronizer will be distributed out on both the input shaft and the lay shaft. How much the shaft’s speed will change depends on the cone torque, RPM depending drag, the selector finger drag, the reflected inertia from the lay shaft and the reflected inertia from the input shaft.

In the preceding shift scenario, the initial RPM difference of the split synchronizer, the input shafts RPM and the lay shafts RPM is known and the values of the reflected inertia is static. A time-discrete calculation is needed to find how much the input shaft and the lay shaft changes RPM in order to calculate the initial RPM difference for the main gearbox.

The time-discrete calculation for the preceding shift mode is performed as described. The reflected inertia, the initial RPM difference for the input shaft, lay shaft and the relative speed between the synchronizer surfaces are first determined statically. After that a time-discrete calculation begins where one shaft is increasing its rotational speed and the other decreasing its rotational speed. The shafts are calculated separately and the values are compared in every time step to determine if the synchronization is complete. The factors that are affecting the lay shaft are: RPM depending drag and cone torque which is calculated in every time step and the reflected inertia. The factors that are affecting the input shaft are: RPM depending drag, selector finger drag and cone torque which is calculated in every time step and the reflected inertia.

4.1.3 Inertia

The components inertia was measured in the CAD models, and the program supports input for inertia as shown in Table 5.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Component</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iₐ</td>
<td>Annulus</td>
<td>Inertia of the annulus</td>
</tr>
<tr>
<td>Iₐ</td>
<td>Clutch</td>
<td>Inertia of the clutch</td>
</tr>
<tr>
<td>Iₐ₋ₐₐ</td>
<td>Crawl gear</td>
<td>Inertia of the crawl gear</td>
</tr>
<tr>
<td>Iₑ₋₁ₕ</td>
<td>Gear</td>
<td>For each gear in split and main gearbox. Including synchronizer parts attached to the gear</td>
</tr>
<tr>
<td>Iₐ</td>
<td>Input shaft (component)</td>
<td>Inertia of the input shaft (component)</td>
</tr>
<tr>
<td>Iₙ</td>
<td>Lay shaft (component)</td>
<td>Including gears attached to the lay shaft and oil pump</td>
</tr>
<tr>
<td>Iₘₛ</td>
<td>Main shaft (component)</td>
<td>Excluding gears on the main shaft, since they are attached by bearings. Including synchronizer parts attached to the main shaft</td>
</tr>
<tr>
<td>Iₚ₉₉</td>
<td>Planetary gear</td>
<td>Inertia of the planetary gear</td>
</tr>
<tr>
<td>Iₑ₋ₑ</td>
<td>Reverse gear</td>
<td>Inertia of the reverse gear</td>
</tr>
<tr>
<td>Iₑ₋ₑₑₑ</td>
<td>Reverse intermediate gear</td>
<td>Inertia of the reverse intermediate gear</td>
</tr>
<tr>
<td>Iᵣₛ</td>
<td>Reflected inertia at the input shaft</td>
<td></td>
</tr>
<tr>
<td>Iᵣₛ</td>
<td>Reflected inertia at the lay shaft</td>
<td></td>
</tr>
<tr>
<td>Iᵣₛ</td>
<td>Reflected inertia at the main shaft</td>
<td></td>
</tr>
</tbody>
</table>
The component inertia was added to the shafts and the range, so four inertias were used for the calculations. The inertia on the input shaft is

\[ I_{IS} = I_{IS} + I_c \]  \hspace{1cm} (51)

The inertia for the lay shaft is

\[ I_{LS} = I_{LS} + I_{g_c} \cdot \frac{1}{R_{oG_c}} + \left( I_{g_r} \cdot \frac{1}{R_{oG_r}} + I_{p,ri} \right) \cdot \frac{1}{R_{oG_{ri}}} + \sum_{i=1}^{n} I_{g_i} \cdot \frac{1}{R_{oG_i}} \]  \hspace{1cm} (52)

where RoG are the ratio of gear for the different gears which are connected to the lay shaft. The inertia for the main shaft is

\[ I_{MS} = I_{ms} \]  \hspace{1cm} (53)

The reflected inertia for the range is

\[ I_{Range} = I_a + n_{PG} \cdot I_{PG} \cdot (RoG_{A-PG})^2 + I_{MS} \cdot (RoG_{A-S})^2 \]  \hspace{1cm} (54)

### 4.1.4 Drag

The components drag in a GRS905 has been measured experimentally in an previous master thesis [15], and can be seen in Table 6. The values are obtained at direct gear (gear 12), 16.5 litres of oil and an oil viscosity and mechanical play equal to the viscosity and play at 90°C for the Scania factory fill oil.

<table>
<thead>
<tr>
<th>RPM / component</th>
<th>600</th>
<th>900</th>
<th>1200</th>
<th>1500</th>
<th>2000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total bearing losses [Nm]</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>Splash losses, no planetary [Nm]</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>Sync. Losses, no planetary [Nm]</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>Oil pump losses [Nm]</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>Input sealing</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>Planetary splash losses [Nm]</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>Fork losses, no planetary [Nm]</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>Planetary fork losses [Nm]</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
</tbody>
</table>

Figure 15 shows that the measured drag increases linearly with increasing rotational speed. Figure 16 shows that the measured drag increases approximately linearly with increasing oil quantity within the interval 12 to 20 litre, but can’t be assumed to be valid outside the interval. Figure 17 shows that the measured drag is approximately linear to the temperature in the given interval, but the interval is too small to determine cold start conditions. Figure 18 shows calculated values for the drag for different temperatures. The calculations are performed by the Analysis and Testing department at Scania Transmission Development, NTCS, using a self-made program [24].
Figure 15. Drag vs. rotational speed.

Figure 16. Drag vs. oil quantity.
Figure 17. Drag vs. temperature.

Figure 18. Calculated drag on direct gear

Figure 18 shows clearly that the drag is not linear to the temperature at lower temperatures, so the drag – viscosity relationship was investigated. The viscosity is interpolated between two reference values using the Ubbelohde-Walther method, shown in equations (55-58) [25].

\[
m = \frac{W_2 - W_1}{\log T_2 - \log T_1}
\]

\[
W_1 = \log(\log(v_1 + 0.8))
\]

\[
W_2 = \log(\log(v_2 + 0.8))
\]
The kinematic viscosity is

\[ \nu = \frac{\mu}{\rho} \]  \hspace{2cm} (58)

where \( \mu \) is the dynamic viscosity and \( \rho \) the oil's density. The density is assumed to drop 10% every 150°C increase (in the relevant interval). Figure 19 shows that the calculated drag is close to linear to the viscosity in the temperature interval.

Figure 19. Calculated drag vs. Viscosity.

The drag was modelled by distributing the component drag to the input shaft, lay shaft and main shaft. Some components affect more than one shaft, and was in that case applied to both since the objective is to calculate the synchronization which happens sequentially. Table 7 shows how the component drag is distributed to the different shafts, where \( RoG_M \) is the mean ratio of gears for all the main gearbox. The values in the table are only coefficients for the inputted drag values, which are determined by the total component drag divided on how many components there are and multiplied by the number of components that are influencing the specific shaft.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Component / shaft</th>
<th>Input shaft</th>
<th>Lay shaft</th>
<th>Main shaft</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T_{D\text{-bearing}} )</td>
<td>Total bearing losses</td>
<td>0.1</td>
<td>0.2 + 0.7 \cdot RoG_M</td>
<td>0.7</td>
</tr>
<tr>
<td>( T_{D\text{-splash}} )</td>
<td>Splash losses, planetary not incl.</td>
<td>0</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>( T_{D\text{-sync}} )</td>
<td>Sync. Losses, planetary not incl.</td>
<td>0.25</td>
<td>0</td>
<td>0.75</td>
</tr>
<tr>
<td>( T_{D\text{-pump}} )</td>
<td>Oil pump losses</td>
<td>0</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>( T_{D\text{-sealing}} )</td>
<td>Input sealing</td>
<td>1</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>( T_{D\text{-splash-p}} )</td>
<td>Planetary splash losses</td>
<td>0</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>( T_{D\text{-fork}} )</td>
<td>Fork losses, planetary not incl.</td>
<td>0</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>( T_{D\text{-fork-p}} )</td>
<td>Fork losses, planetary</td>
<td>0</td>
<td>0</td>
<td>1</td>
</tr>
</tbody>
</table>

The drag’s dependence to the engine RPM, oil quantity and oil viscosity are related to the total drag with three coefficients for the different dependencies. The coefficients are the following:

- Engine RPM (linear), the coefficient equals one for the different shafts and their RPM at direct gear with an engine RPM of 2000. Increase with higher RPM and decreases with lower RPM. The slope and y-intercept can be changed by the user via the GUI.
- Oil quantity (linear), the coefficient equals one for 16.5 litre and decreases with lower oil quantity and increases with higher oil quantity. The slope and y-intercept can be changed by the user via the GUI.
- Oil viscosity (non-linear), the coefficient equals one for 0.0155 Pa s and decreases with lower viscosity and increases with higher viscosity.

The coefficients for the oil quantity and the oil viscosity is calculated and assumed to be constant during the synchronization and the coefficient for the RPM is calculated in each time step in the time discrete calculations. Due to the fact that the coefficient for the oil quantity and the RPM is linear they can easily be modelled with an linear equation.

The drag’s viscosity (hence temperature) dependence is modelled with a third degree polynomial

\[ \text{visc}_\text{drag} = A \cdot x^3 - B \cdot x^2 + C \cdot x + D \]  \hspace{1cm} (59)

The polynomial is based on the values from Figure 19 and calculated in a least-square sense. The polynomial and the drag data is shown in Figure 20.

![Figure 20. Polynomial fitted to measured values for drag.](image)

The measured drag divided between the shafts is

\[ T_{\text{D,input shaft,base}} = T_{D,\text{bearing}} \cdot 0.1 + T_{D,\text{sync}} \cdot 0.25 + T_{D,\text{sealing}} \]  \hspace{1cm} (60)

\[ T_{D,\text{lay shaft,base}} = T_{D,\text{splash}} \cdot T_{D,\text{bearing}} \cdot (0.2 + 0.7 \cdot Ro \cdot G_M) + T_{D,\text{pump}} \]  \hspace{1cm} (61)

\[ T_{D,\text{main shaft,base}} = T_{D,\text{sync}} \cdot 0.75 + T_{D,\text{bearing}} \cdot 0.7 + T_{D,\text{fork}} + T_{D,\text{fork,\text{p}}} + T_{D,\text{splash,\text{p}}} \]  \hspace{1cm} (62)

The total drag for shaft n is

\[ T_{D,n} = T_{D,n,\text{base}} \cdot C_{\text{RPM}} \cdot C_{\text{quantity}} \cdot C_{\text{viscosity}} \]  \hspace{1cm} (63)

Due to the friction between the selector finger and the shift sleeve a torque arise from the axial force and the radial distance from the centre, it can be described by equation (64). The drag from the fork selector finger only affects the synchronizers that’s not connected to the main shaft since the range is in gear when main changes gear. In a manual gearbox it will assist the synchronizers work when the input shaft is supposed to lower its RPM to synchronize the relative sliding velocity or when the range is shifting to low range, however its increasing the workload for the synchronizer in the opposite cases. For the Opticruise 3-pedal it only affects the range shifts and in a similar way as the manual. In the Opticruise 2-pedal version it affects the gearbox in the same way as in manual in shifts where the clutch is open and like the 3-pedal version when the clutch is closed.

\[ T_{D(\text{selector finger})} = F_{AX} \cdot \mu_{sf} \cdot R_{S} \]  \hspace{1cm} (64)
4.1.5 Coefficient of Friction

The friction was modelled by two experimentally determined Stribeck curves, one for steel – molybdenum and one for steel – carbon fibre. The objective of the curve is to describe the oil wipe phase, which is hard to determine analytically. Since the Stribeck curve’s dimensionless lubrication parameter take load and sliding velocity into consideration, it can be used for the different gearshifts without modification. Since the dimensionless lubrication parameter also takes viscosity into consideration, it can be used for cold start analysis too.

Rotational speed can be converted to sliding velocity, axial force can be converted to load and oil temperature can be converted to viscosity, assuming the viscosity at 40°C and 100°C is known. This means the dimensionless lubrication parameter can be determined, and the resulting coefficient of friction can be interpolated from the experimentally determined values.

A pin-on-disc rig was determined to be a good way to perform the measurements [26][27].

4.2 Program outline

The program was divided into two parts, the GUI and the calculations. The file structure is shown in Figure 21, where the files main_gui.m and main_gui.fig and the .jpg images defines the GUI and the rest of the .m-files are the calculations. Files with other file extensions are saved data from the program, according to Table 8. The file clickableLegend.m is downloaded from MATLAB File Exchange [28] to replace the command plotbrowser. Both commands shows and hides plotted graphs, but plotbrowser had some unwanted effects on the GUI objects too.

<table>
<thead>
<tr>
<th>File extension</th>
<th>Saved values</th>
</tr>
</thead>
<tbody>
<tr>
<td>.gbd</td>
<td>Gearbox data</td>
</tr>
<tr>
<td>.sync</td>
<td>Synchronizer data</td>
</tr>
<tr>
<td>.cav</td>
<td>Calculation data</td>
</tr>
<tr>
<td>.sins</td>
<td>Single synchronizer data</td>
</tr>
</tbody>
</table>

Table 8. File name extensions for saved data.
4.2.1 Graphical User Interface

The GUI consists of four windows for calculations of all synchronizers in a gearbox, shown in Figure 22 to Figure 25, and one additional window for calculating a single synchronizer, called Single sync., shown in Figure 26. Note that the name Single sync. refers to one synchronizer, not a synchronizer with one cone. In the Single sync. window, one to three cones are supported.

All five windows are stacked on top of each other, and shown or hidden by setting the property “Visible” to “on” or “off”, depending on the users action. In the Main window, the user chooses the gearbox type, number of gears in the main gearbox, to include crawl gear or not and what shift mode to use. The other windows updates according to the users choice, e.g. hiding input fields for split and range data for a type G gearbox.
Figure 22. GUI main window.

Figure 23. GUI for defining gearbox window for a GRS type gearbox with six main gears.
Figure 24. GUI for defining synchronizers for a GRS type gearbox with six main gears.

Figure 25. GUI for defining calculation.
The input parameters are divided into three windows, with some input fields shown or hidden depending on the values of the main window. In the *Gearbox* window, Figure 23, the data regarding the whole gearbox is requested. The following data is inputed:

- Number of teeth for both driver and driven.
- Inertia of components.
- Strut parameters.
- Oil data.
- Drag parameters.

In the *Synchronizers* window, Figure 24, synchronizer data is requested. The following data is inputed:

- Synchronizer material, Molybdenum or Carbon fibre.
- Number of cones.
- Cone radius, width and angle.
- Blocker angle and radius.
- Selector finger radius.

In the *Calculation* window, Figure 25, calculation variables are requested. The following data is inputed:

- Discrete time step and max time.
- Max engine RPM and engine ramp speed.
- Estimated cone area in contact due to angular misalignments, other tolerance and surface imperfections.
- Material limits.
- What gearshifts to include, to reduce number of plots.
- Force buildup and max force.
- Friction properties.

Figure 26. GUI for defining single synchronizer calculation.
In the Single sync. window, Figure 26, only a single synchronizer is used and the gearbox data is replaced by a few input fields for reflected inertia and drag. All input data can be saved and loaded via a standard interface native to the user’s computer platform, with one file type per input window. For common gearboxes, quick start buttons are available which loads the correct files automatically, based on filename. The output are divided into two parts, static output and time-discrete output. The static output is presented in the orange box in the Main window and Single sync. window and in the MATLAB command window. In the Main window’s orange box a popupmenu is available to switch between the different synchronizers. Only max values are shown in the orange box, all the values for each gear change and synchronizer are displayed in the command window. The time-discrete output is plotted into different figures. Subplots are used to reduce number of figures needed and increase user-friendliness. The function of the legend has been expanded beyond the standard MATLAB legend, and now allows for hiding and showing specific graphs by clicking their legend entries. For complete gearbox calculation, the user can choose to plot either:

- Sliding velocity [m/s], RPM difference or neither.
- Power/area [W/mm²], power [W] or neither.
- Work/area [J/mm²], work [J] or neither.
- P-v diagram.

For Single sync. calculation, double y-axis are used to display both total and per-area values.

**4.2.2 Time-discrete calculations in the program**

Below is a explanation of the workflow in the function file fun_calculate.m, which handles all major calculations or refers to other, smaller function files. Unless otherwise stated, the examples are for a GRS905 gearbox, 3 step upshifts. Not all variables are shown. fun_calculate_single.m works in a similar way but for the single synchronizer case.

1. Imports data from GUI
   - Imports calculation data, cone data and number of teeth from GUI and saves the data as variables. Cone data is saved in matrix or vector format, Row 1 low split, Row 2 high split, Row 3 1st gear, .... Row 9 low range, Row 10 high range.
   - Output: Cone data as cone_radius_mat, cone_width_mat, cone_angle_vec, cone_material_vec, blocker_angle_vec, blocker_radius_vec, the calculated cone_area_mat, force_split, force_main, force_range and misc. variables. cone_radius_mat and cone_material_vec with legends are shown below, with C1 being the outermost cone. The other vectors and matrices are defined in the same way.

   ```
   cone_radius_mat =
   (C1    C2    C3)
   (Low split)   85     0     0
   (High split)  85     0     0
   (1'st gear)   85    80    74
   (2'nd gear)   85    80    0
   (3'rd gear)   85     0     0
   (4'th gear)   0     0     0
   (5'th gear)   0     0     0
   (6'th gear)   0     0     0
   (Low range)   110    0     0
   (High range)  110    0     0
   cone_material_vec =
   (0 = molybdenum, 1 = carbon fiber)
   ```

2. Calculates Ratio of gears
   - Calculates ratio of gears and saves in three vectors (split, main, range) + misc. variables (crawl, reverse)
   - Output: Ratio of gears for split/main/range in vectors, + crawl and reverse as variables. The ratio of gears for the main gearbox in a GRS905 is shown.
rog_main =  
(G1 G2 G3)  
2.3000 1.4615 0.9412

3. Defines gear matrix
   - Function fun_gear_matrix.
   - Defines a matrix containing gear number, split position, main gearbox gear and range position depending on
     gearbox type and number of gears in main gearbox. Low split/range = -1, high split/range = 1.
   - Output: A gear matrix with two to four columns depending on gearbox type, and 3 to 24 rows depending
     on gearbox type and number of gears in the main gearbox. gear_matrix for GRS905 and a type G gearbox with
     6 gears is shown.

\[
gear_matrix = \begin{pmatrix}
  (GRS905) \\
  (Gear Split Main Range) \\
  1 & -1 & 1 & -1 \\
  2 & 1 & 1 & -1 \\
  3 & -1 & 2 & -1 \\
  4 & 1 & 2 & -1 \\
  5 & -1 & 3 & -1 \\
  6 & 1 & 3 & -1 \\
  7 & -1 & 1 & 1 \\
  8 & 1 & 1 & 1 \\
  9 & -1 & 2 & 1 \\
 10 & 1 & 2 & 1 \\
11 & -1 & 3 & 1 \\
12 & 1 & 3 & 1 \\
\end{pmatrix}
\]

\[
gear_matrix = \begin{pmatrix}
  (G with 6 gears) \\
  (Gear Main position) \\
  1 & 1 \\
  2 & 2 \\
  3 & 3 \\
  4 & 4 \\
  5 & 5 \\
  6 & 6 \\
\end{pmatrix}
\]

4. Defines vector of initial and final gears
   - Function fun_gear_vectors.
   - Defines two vectors, one for initial and one for final gear based on GUI inputs: include 1 step shifts, include
     2 step shifts, include 3 step shifts, upshifts, downshifts and max number of gears.
   - Output: Two gear vectors containing initial and final gear. The initial and final gear vectors for 3 step
     upshifts for a GRS905 is shown. In this example, 9 gearshifts are performed.

\[
\begin{align*}
  \text{initial_gear} & =  \\
  & =  \\
  & =  \\
  & =  \\
  & =  \\
  & =  \\
  & =  \\
  & =  \\
\end{align*}
\]

\[
\begin{align*}
  \text{final_gear} & =  \\
  & =  \\
  & =  \\
  & =  \\
  & =  \\
  & =  \\
\end{align*}
\]

5. Creates vectors with initial and final gear positions
   - Function fun_smr_pos.
   - Initial and final gear vectors and gear_matrix is used to determine initial and final position of split, main
     gearbox and range.
   - Output: Six vectors describing the position of the split, main gearbox and range before and after shift.
     Initial_split and final_main are shown. Note the relationship between initial_gear, initial_split and the
     gear_matrix as well as final_gear, final_main and gear_matrix.

\[
\begin{align*}
  \text{initial_split} & =  \\
  & =  \\
  & =  \\
  & =  \\
  & =  \\
\end{align*}
\]

\[
\begin{align*}
  \text{final_main} & =  \\
  & =  \\
  & =  \\
  & =  \\
  & =  \\
\end{align*}
\]
6. Determine RoG for initial and final gear positions
   - Function fun_rog.
   - Initial and final split, main and range positions and the ratio of gears is used to determine initial and final
     ratio of gears for split, main and range for every gearshift.
   - Output: Six vectors describing the ratio of gear for split, main gearbox and range before and after shift.
     Initial split and final main are shown. Note the relationship with initial_split as well as final_main and
     rog_main. Split and range will be set to 0 if not available in the current gearbox.

   \[
   \text{rog_initial_split} = \\
   \begin{pmatrix}
   1.3125 & 1.0625 & 1.3125 & 1.0625 & 1.3125 & 1.0625 & 1.3125 \\
   1.0625 & 1.3125 & 1.0625 & 1.3125 & 1.0625 & 1.3125 & 1.0625
   \end{pmatrix}
   \]

   \[
   \text{rog_final_main} = \\
   \begin{pmatrix}
   1.4615 & 0.9412 & 0.9412 & 2.3000 & 2.3000 & 1.4615 & 1.4615 \\
   0.9412 & 0.9412 & 2.3000 & 2.3000 & 1.4615 & 1.4615 & 1.4615
   \end{pmatrix}
   \]

7. Imports and calculates drag
   - Function fun_drag.
   - Imports drag data from GUI and calculates the drag at the input shaft, lay shaft and main shaft, as well as
     the friction from the selector finger, called td_peg. This drag takes oil data into consideration, as shown in
     section 4.1.4 - Drag. The effect of the rotational speed is however not included yet (since the rotational
     speed for every time step is not possible to calculate yet), so these drag values are for 2000 RPM at the input
     shaft for the direct gear (gear 12) of a GRS905 gearbox.
   - Output: Three variables containing drag for each shaft and three vectors containing the selector finger
     friction losses for each shaft at each time step, based on max simulation time and discrete time step inputs.

   \[
   \text{drag_lay_shaft} = \\
   \begin{pmatrix}
   \text{xx.} & \text{xxxx}
   \end{pmatrix}
   \]

   \[
   \text{drag_main_shaft} = \\
   \begin{pmatrix}
   \text{xx.} & \text{xxxx}
   \end{pmatrix}
   \]

   \[
   \text{td_peg_split} = \\
   \begin{pmatrix}
   0 & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx \\
   x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx \\
   x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx \\
   x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx \\
   [ \ldots ] & x. & xxxx & x. & xxxx & x. & xxxx & [ \ldots ]
   \end{pmatrix}
   \]

8. Imports and calculates inertia
   - Function fun_ref_inertia_lay.
   - Imports inertia per component and ratio of gears to calculate the reflected inertia at each synchronizer.
     \text{ref_inertia_split} and \text{ref_inertia_range} is two variables while \text{ref_inertia_main_vec} is a vector containing
     different inertia for each gearshift. Note that \text{ref_inertia_main_vec} effects the split synchronizer in some
     shift modes.
   - Output: Two variables, \text{ref_inertia_split} and \text{ref_inertia_range} as well as the vector \text{ref_inertia_main_vec}.

   \[
   \text{ref_inertia_main_vec} = \\
   \begin{pmatrix}
   x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx \\
   x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx & x. & xxxx
   \end{pmatrix}
   \]
ref_inertia_split =

X.xxxx

9. Initial RPM diff for GRS gearbox.
   • Functions fun_drpm_split_grs, fun_drpm_main_grs, fun_drpm_range_grs.
   • Calculates initial RPM differences in the gearbox based on gearshift, max RPM, shift mode, ratio of gears and inertia. Similar functions exists for G and GR gearboxes.
   • Output: Vectors containing initial RPM differences for the split (both axis to axis and surface to surface, as explained in section 4.1.2 - Initial RPM difference), main gearbox and range as well as initial RPM of the lay shaft and main shaft, which will be used to calculate the RPM-depending drag.

\[
\begin{align*}
\text{drpm_main} &= 1.0e+003 * \\
&\begin{bmatrix}
X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
X.xxxx & X.xxxx \end{bmatrix} \\
\text{rpm_lay_shaft} &= 1.0e+003 * \\
&\begin{bmatrix}
X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
X.xxxx & X.xxxx \end{bmatrix} \\
\text{drpm_split_surface} &= \begin{bmatrix}
X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\end{bmatrix}
\end{align*}
\]

10. Radians per second over time (for split/main/range)
   • Function \text{fun_rpm_time}.
   • Calculates the rotational speed difference in rad/s for each time step, based on initial RPM difference, cone data, force, drag, reflected inertia, CoF. Also recalculates drag with instantaneous rotational speed.
   • Output: Matrices containing rotational speed differences for each time step for each gear change and cone torque, to calculate power.

\[
\text{radpers_split} = \begin{bmatrix}
(t = 0) & t=0.005 & t=0.01 & t=0.015 & t=0.02 & t=0.025 \\
\text{(Gear 1 to 4)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\text{(Gear 2 to 5)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\text{(Gear 3 to 6)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\text{(Gear 4 to 7)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\text{(Gear 5 to 8)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\text{(Gear 6 to 9)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\text{(Gear 7 to 10)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\text{(Gear 8 to 11)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\text{(Gear 9 to 12)} & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx & X.xxxx \\
\end{bmatrix}
\]

11. Convert rad/s to RPM or meters per second (sliding velocity)
   • Trims the matrices by removing the columns with only zeros and converts radians per second to RPM or meters per second by element-per-element multiplication, depending on user input.
   • Output: Meters per second (sliding velocity) and, depending on the users inputs, RPM in the same format as in 10 - Radians per second over time (for split/main/range).

12. Calculate power, work and contact pressure
   • Uses the output from 10 - Radians per second over time (for split/main/range) to calculate power, power/area, work and work/area (depending on user input).
13. **Static output values**
   - Calculates parameters which are not time dependent and displays the values in the orange panel in the main window.
   - Output: Static worst-case values to the orange panel in the main window.

14. **Calculate and print the result for each gear change to command window**
   - Calculates parameters which are not time dependent and displays the values in the MATLAB *command window*.
   - Output: Static values for each gear change to the MATLAB *command window*.

15. **Initiates plots**
   - Prepares for the plotting by setting line color and style, legend and figure name. The five vectors named `skip_to_#_up/down` help extract the relevant data from the matrices, by e.g. plotting rows `i+skip_to_3_up` and forward for the figure containing 3 step upshifts. The vectors depends on what gearshifts are enabled, if only 3 step upshifts are enabled, the variable `skip_to_3_up = 0`, and if all shifts are selected the variable `skip_to_3_up = 42` (for GRS905). The legend text and figure name are saved in `legend_text_mat` and `figname_mat`, where the 1’st row is for the 1’st figure and so on. The vector `plots_enabled` contains six elements which are 0 (plot disabled) or 1 (plot enabled).
   - Output: Vectors containing figure name and legend text and vectors that helps to extract relevant data from the matrices.

16. **Plot velocity/power/work for G/GR/GRS**
   - Functions `fun_plot_g`, `fun_plot_gr`, `fun_plot_grs`.
   - Plots data into 1-18 plots with 1-3 subplots each, depending on gearbox and user input.
   - Output: Plots.

Figure 27 shows a flowchart over the calculations performed in the MATLAB program at each timestep.

![Flowchart](image.png)
4.2.3 Output

Notice that the RPM difference is calculated in the friction surface e.g. the range synchronizer surface is changing from the main shafts rpm to zero when downshifting. When the gear changes are similar, only difference is high or low range, only one of the case will be plotted.

The following outputs are available.

Detailed results (command window)

The following data is presented for all gearshifts and for split, main and range (when available):

- $drpm =$ Initial RPM difference.
- $W/mm^2 =$ Max power/area in the synchronizer surface.
- $J/mm^2 =$ Max work/area in the synchronizer surface.
- $Kgm^2_E =$ The reflected inertia at the engines side (for each synchronizer).
- $Kgm^2_W =$ The reflected inertia at the wheels side (for each synchronizer). Note: infinite when the shaft is driven by the wheels.
- $Ns =$ Shift impulse.
- $\%MP =$ Power/area safety margin.
- $\%MW =$ Work/area safety margin.
- $Ts =$ time to synchronize.
- $T_{above} =$ Duration of power/area over a user-specified limit.
- $Tc/Ti@BTL =$ Cone torque / index torque. $Tc/Ti$, when the force equals the BTL.

Max values (Orange panel)

In the orange panel in the Main window, the max values from the command window for each synchronizer are presented. The user can chose which synchronizer is shown with the popup menu.

Plotted data

The following data can be plotted:

- Sliding velocity [m/s], RPM difference or neither.
- Power/area [W/mm²], power [W] or neither.
- Work/area [J/mm²], work [J] or neither.
- P-v diagram.

In the p-v diagram, the material limits can be set by the user in the Calculation window, which needs to be determined experimentally.

4.3 Results – New program

The program’s GUI can be seen in Figure 22 to Figure 26, as well as in the manual. The main advantage over the Excel spreadsheet is the support for other gearboxes than GRS905. Other advantages include:

- 1-step, 2-step and 3-step shifts are available.
- The drag, which is now RPM dependant. Inputs for the drag has also been simplified by inputting temperature and viscosity at reference points.
- Simplified input of data.
- Simplified evaluation of outputs.

The documentation for the program consists of this report and a program manual, which can be found in Appendix 3. The code is commented to allow for future modification.
4.4 Discussion – New program

The material limits for the p-v diagram and safety margin calculations has to be determined experimentally. A guideline can be found in subcontractors data sheets, but measurements on a Scania synchronizer is needed. The estimated 50% of actual contact due to surface macro roughness and angular misalignments can be used if it’s considered when determining the material limits. This method is one way to account for a large part of the tribological contact between the friction surfaces.

4.4.1 Drag

Measured values for drag is only available for GRS905 gearbox, but is assumed to be valid for all combinations of GR and GRS gearboxes available. The design difference between GR and GRS are small, but it’s likely that GR has slightly lower drag than GRS. Values for 905 gearboxes is assumed to be valid for 875 and 925 type gearboxes too [29]. All parameters affecting drag is changeable if measurements are performed. No data is available for the purchased type G gearbox.

The linear relationship between drag and oil quantity can’t be assumed to be valid outside the tested interval. When using less than 12 litres of oil in the gearbox, the system could turn into a dry sump system, meaning that the lay shaft isn’t submerged in oil. The breakpoint between wet and dry sump is not investigated. With an oil level above 20 litres, the lay shaft could become fully submerged and the main and input shaft partially submerged, which could lead to total system drag becomes nonlinear.

The drag temperature dependence seems to be close to linear to the viscosity according to the calculated drag, but the calculated values are uncertain in the lower temperature range of the diagram where the drag increases rapidly. A probable reason for this is that the oil heats up locally in the high-loss regions which lowers losses, and the simulation program doesn’t take that into consideration. The local heating makes it hard to measure the drag experimentally too. The local heating affects a vehicle mounted gearbox too, but the problem is where to measure the temperature. It is likely that the oil in the sump stays cold while the drag is reduced. However, the drag is high during cold starts, as any Swedish driver have experienced in their passenger car. In the higher temperature range of the diagram the calculated values corresponds well with the measured values.

4.4.2 Coefficient of Friction

Unfortunately the components needed to perform the measurements in the pin-on-disc test rig was delayed and therefore the tests got delayed, so no values could be implemented. The program is prepared for adding this functionality and only minor modifications have to be made to implement it.

4.4.3 GUI

To switch between different windows, the property “Visible” is set to “on” or “off”, rather than using the MATLAB function “uitab”. The reason is that uitab is undocumented in MATLAB 7.9.0 (R2009b). The MATLAB documentation states: “This function (uitab) is undocumented and will change in a future release” [30]. The risk of the program not working properly in a future version of MATLAB is not worth taking, when the method used accomplishes the same thing.

4.4.4 Single sync. calculations

The advantages of single sync calculations is that it is easier to achieve the same reflected inertia and drag as in the test rig internally named T24, and it can be easier to evaluate the results since the number of outputs are far less than when calculating the complete gearbox.
5 Experimental validation

The test rig used for verification is schematically shown in Figure 28. The test rig is built by Hoerbiger and is internally named T24. The electric motor rotates the flywheel to a user-controlled speed and disengages. An axial force is applied to the synchronizer, pressing the friction surfaces together like in a complete gearbox. When the flywheel stops, the electric motor engages again and the process is repeated. During the whole process, oil flows from a pipe above the synchronizer. The inertia can be changed by changing the flywheel, the oil flow and temperature can be controlled and both the axial force and rotational speed can be varied. The rig records the force, synchronization torque, rotational speed, temperature and shift sleeve position. From the measured values, the CoF can be calculated, however, in the beginning of the synchronization the values are unstable, and the CoF therefore cannot be considered correct. This is acknowledged by Hoerbiger, and the data evaluation software doesn’t include the early CoF values in the graphs.

![Figure 28. Schematic illustration of test rig T24.](image)

To be able to compare the measured values with the values calculated by the program, the following data is needed:

- Number of cones, cone radius(es) and cone angle(s).
- Axial force.
- CoF.
- Initial RPM to synchronize.
- Inertia to synchronize.
- Drag in the test rig.

The verifiable output is the synchronization time and the shape of the RPM curves. The test rig data was taken from Scania’s normal test procedure and not tailor made for the purpose of comparison.

5.1 Measured values for the drag in T24

The test rig reports all data needed for a comparison with the calculated values from the program, except the drag. The drag was calculated by measuring the time for the rig to decelerate from 1500-1000 RPM and 1000-500 RPM without applying an axial force to the synchronizer, and calculate the required torque for the entire deceleration, as in equation (12). The measurements were taken at different temperatures with an oil with similar properties as Scania factory fill oil. Table 9 shows the calculated drag in Nm at different temperatures and RPM, with 1250-750 RPM being the average (rollout time) of 1000-500 and 1500-1000.
Table 9. Calculated drag in T24 [Nm].

<table>
<thead>
<tr>
<th>RPM / temp.</th>
<th>1000-500</th>
<th>1500-1000</th>
<th>~1250-750</th>
</tr>
</thead>
<tbody>
<tr>
<td>55</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>60</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>70</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>80</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>85</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
<tr>
<td>90</td>
<td>XX</td>
<td>XX</td>
<td>XX</td>
</tr>
</tbody>
</table>

For the calculations in the *Single sync.* mode, the drag is assumed to be linear between an initial and a final value.

### 5.2 Adapting to measured values

Figure 29 shows an example of the rest rig output. Note the variable CoF and axial force, which differs from the simplified implementation in the program.

![SINGLE SHIFT DIAGRAM](image)

**Figure 29. Test rig T24 output, cycle 2200.**

Figure 30 shows a comparison between calculated and measured values, with the following data: $I_R = 0.42$ Kg·m², CoF = 0.095, max force = 2529 N, force buildup = 10241 N/s, initial RPM difference = 549 RPM, drag = $4 / 1.5$ (initial / final) Nm, standard single cone synchronizer. The values are taken from the test rig’s output, cycle 2200 [31].
Figure 30. First comparison of calculated and measured RPM.

Figure 30 shows that the synchronization time corresponds well with the measured values, but the ΔRPM at every time step is lower in the calculated values. Two main sources of error were identified, the CoF and the axial force. To improve accuracy, two more complex representations were tested. Figure 31 shows the same case but with a slightly variable CoF, which increases linearly between two measured values from the test rig as in Figure 29.

Figure 31. Second comparison of calculated and measured RPM.

The result improved slightly, but the CoF was not the main source of error. Figure 29 shows that the axial force is nonlinear, however, the shape of the force buildup curve differs greatly between different test cycles, as shown in Figure 32. Five random test cycles are shown, all taken from the same test sequence as the data in Figure 29 [31].
The same synchronizer was used for all five tests. Note that the force buildup rate and max force is different between the cycles, but a seemingly random shape of the curves exists. The curves are likely to differ even more between different synchronizers due to production tolerances. The biggest difference is the number of bumps in the curve, ranging from none to over five. The common feature between the curves, which is also observed in other similar situations (pneumatic and hydraulic systems), is the slow force buildup in the beginning, before the linear trend [32]. Force approximation by using different polynomials for the first 0.05-0.07 second and then a linear function is shown in Figure 33.

![Figure 32. Force buildup in test rig for 5 random cycles.](image1)

![Figure 33. Different ways to model the axial force.](image2)
Figure 33 shows that a 2nd degree polynomial represents the actual force buildup slightly better than a linear function. The two criterions used for the polynomial are:

\[
\begin{align*}
y_{poly}(t_e) &= y_{linear}(t_e) \\
y'_{poly}(t_e) &= y'_{linear}(t_e)
\end{align*}
\]  

(65)  

(66)

where \( t_e \) is the inputted force delay time.

Figure 34 shows the same case as in Figure 30 and Figure 31 with the calculated RPM difference based on the variable force included.

![Graph](image.png)

Figure 34. Third comparison of calculated and measured RPM.

5.3 Results – Experimental validation

The calculations have been adapted to measured values, and the maximum error had been decreased by almost 60% in this specific case. The maximum error is 3.1% in this case.

5.4 Discussion – Experimental validation

Table 9 shows that the drag in the test rig is small compared to the cone torque when using an axial force of similar magnitude as the axial force in the gearbox.

Figure 29 shows that the time scale in the test rig starts before the synchronization process starts. To be able to compare with calculated data, the test rig values will have to be trimmed so the synchronization starts at \( t = 0 \).
6 Results

The result of this master thesis is a new program for synchronization calculations. The advantages compared to the old program is:

- Support for G, GR and GRS gearboxes as well as support for three to six gears in the main gearbox.
- 1-step, 2-step and 3-step shifts are available.
- Verified by physical testing.
- It’s verified that the theory used at Scania corresponds with new publications.
- Improved accuracy, especially in the drag and force buildup.
- More user-friendly interface.
- Easier to edit and adapt for future needs.

Figure 35 shows a comparison of data from section 5.2 - *Adapting to measured values* as well as calculated values from the Excel spreadsheet.

![Comparison of MATLAB and Excel calculated values.](image)

To verify the output, five random test cycles were chosen from the same test sequence [31]. Table 10 shows a comparison of measured and calculated values. The force and RPM difference for each time step is shown in appendix 4 (cycle 2200 is shown in section 5.2 - *Adapting to measured values* instead).

<table>
<thead>
<tr>
<th>Table 10. Calculated and measured synchronization time comparison.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Constants</strong></td>
</tr>
<tr>
<td><strong>Drag, initial/final, [Nm]</strong></td>
</tr>
<tr>
<td><strong>Cone data</strong></td>
</tr>
<tr>
<td><strong>Variables</strong></td>
</tr>
<tr>
<td><strong>Cycle</strong></td>
</tr>
<tr>
<td>980</td>
</tr>
<tr>
<td>1960</td>
</tr>
<tr>
<td>2200</td>
</tr>
<tr>
<td>2780</td>
</tr>
<tr>
<td>3040</td>
</tr>
</tbody>
</table>

*The input to the test rig was wrong. Data for a triple cone synchronizer was used for a single cone synchronizer. The CoF was recalculated in MATLAB. This doesn’t change the results.*
7 Discussion and conclusions

The calculated values corresponds well with the measured values, and is an improvement over the Excel spreadsheets.

7.1 Sources of error

The sources of error is divided into two parts; CoF and the axial force which results in larger errors and assumption which results in minor calculation error

7.1.1 Coefficient of friction and axial force

The results corresponds well with the measured value from the test rig, but a few sources of error were identified. The biggest source of error found was the axial force, especially during the early stages of synchronization. Different methods were tested to improve the simulated force buildup:

- Linear function.
- Using MATLAB polyfit on measured values from the test rig.
- Adapting a simple function, like force = c·time^n, for the beginning of the synchronization.

The linear function used in the Excel spreadsheet gave the worst result. The advanced polynomial gave best results, but were considered too complicated since it very case-specific and impossible to predict and therefore dependant on physical testing. A 2nd degree polynomial for the first stage of the force buildup and then a linear function was chosen since it was a good trade-off between usability and accuracy. This method reduced the error by almost 60% compared to a linear force in some cases.

The CoF in the boundary lubrication regime is assumed to be constant. Measurements show that this is not entirely true, and tests were made on how to improve the accuracy of the CoF. However, the improvements were small and the input for the CoF became much more complex, and relied on measuring test data which can be time consuming.

The method for simulating the oil wiping phase, which is in the mixed lubrication regime, is described in section 4.1.5 - Coefficient of Friction. As stated in section 4.4.2 - Coefficient of Friction, the required measurements were delayed and the solution was not implemented. The error is likely small since it’s during a short duration and the force is low during that stage.

The CoF was expected to be the largest source of error in the Excel spreadsheet, but calculations and test rig output shows that the force were a bigger source of error.

7.1.2 Assumptions

The following assumptions were made:

The tribological contact modelled with the fraction of area in contact are assumed to take the following aspects into consideration; Surface roughness and angular tolerances which can lead to the surface in contact can be on varied positions on the cone and locally higher power/area.

- The oils density is assumed to decrease 10% per 150°C increase.
- Oil temperature and oil quantity are assumed to be constant during the synchronizing.
- The measured drag for GRS905 gearbox is used in all calculations, but is changeable via the GUI.

A constant CoF is used due to the small difference in the result and increase in complexity a variable induced.

The bearing dependent drag and the synchronizer dependent drag is divided on the number of parts with no respect taken to any other data like load and rotational speed.

All gear changes is assumed to happen directly with no time in neutral where it would be affected by drag, except for one case; OPC 2-pedal with clutch open where the split is not changing gear.

The drag in Single sync. mode is modeled to be linear.

The vehicle is also considered to have a constant speed during gear shifts.
7.2 Synchronization time and Shape of RPM vs. time curves

Two properties were compared with the rig data: the synchronization time and the shape of the RPM vs. time curves. The synchronization time is easier to compare but gives less detailed results. Even with the same synchronization time the RPM difference between the measured and calculated values at a certain time step can differ greatly between a good and bad simulation. To explain the difference, synchronization time can be replaced by mean cone torque. The axial force and CoF can be replaced by instantaneous cone torque. With a constant torque, the RPM decreases linearly. With a cone torque that starts off low and then increases rapidly, the RPM will decrease slowly in the beginning and faster at the end. This means the calculated RPM difference at each time step will be higher than if a constant torque was used. Figure 35 shows an example of this: the synchronization time for the green and black line are equal, but the black line are always closer to the blue, measured, line.

It is better to overestimate the instantaneous RPM than underestimating it, since a higher RPM leads to both higher sliding velocity and higher power, which increases wear.
8 Suggestions for future work

The variable CoF described in Section 4.1.5 - Coefficient of Friction can be implemented once the measurements are done.

To improve the accuracy of the drag, the drag have to be measured in colder temperature too, preferably on a component level. As the drag is modelled now, every component has the same temperature dependency as the complete gearbox. The drag temperature dependence is calculated via a tailor-made simulation program, which is suspected to report too high drag during cold conditions.

The “material limits” have to be investigated, preferably by using both the test rig T24 and more focused tribological tests like pin-on-disc and reciprocating ball tests.

Investigate the force build-up rate in the split, range and Opticruise main gearbox, as well as measure the force a driver applies on the gear lever during “normal” and “brutal” gear changes.
9 Bibliography

[26] Hardell, J. (2010, October 25). Mail regarding tribological testing. (D. Häggström, & M. Nordlander, Interviewers)
[27] (2010, December 15). Teleconferance with #censored oil company#. (D. Håggström, & M. Nordlander, Interviewers)

Excel spreadsheet

The currently used Excel spreadsheet for calculating synchronization consists of two sheets, one for inputs and the other for outputs. Figure 1 shows the input sheet, where the coloured area is the input fields, and the white area the documentation. Figure 2 shows the output sheet, where the leftmost area is output of static calculations, the white area is output of the time-discrete calculations (the tables are actually significantly larger than shown in the picture) and the area between static and time-discrete recaps the values of the time-discrete with tables and graphs. The figure’s is only made to get an overview of the old program and the size of the spreadsheet used for the time discrete calculations.

Figure 1. Excel input sheet.

Figure 2. Excel output sheet.
Initial RPM differences

In order to derive the formulas for the initial RPM differences two figures with the RPM at different places in the gearbox before and after the complete gear change was created, see Figure 1 for the upshifts and Figure 2 for the downshifts. This was done to clarify and simplify the derivation and understanding of the formulas. RPM\textsubscript{M} is the engine speed where a gear change is initiated in an upshift and the engine speed after the shift for a downshift. In one case a time-discrete calculation for the split is necessary in order to find the initial rpm difference for the main gearbox; the case where the main gearbox is in neutral when the split changes gear. That case is described in section 4.1.1 – Shift modes in the report.

\[
\begin{align*}
\text{Initial} & \\
\text{RPM}_M & \xrightarrow{\frac{\text{RPM}_M}{\text{RoG}_{IS}}} \xrightarrow{\frac{\text{RPM}_M}{\text{RoG}_{IS} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}}} \\
\text{Final} & \\
\frac{\text{RPM}_M \cdot \text{RoG}_{FS}}{\text{RoG}_{IS} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} & \xrightarrow{\frac{\text{RPM}_M \cdot \text{RoG}_{FR}}{\text{RoG}_{IS} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}}} \xrightarrow{\frac{\text{RPM}_M \cdot \text{RoG}_{FS}}{\text{RoG}_{IS} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}}} \\
\end{align*}
\]

Figure 1. RPM at different places in the gearbox before and after the gear change for an upshift.
Figure 2. RPM at different places in the gearbox before and after the gear change for a downshift.

Formulas for different upshift scenarios in an gearbox with split, main and range:

Split changes gear, main and range are not involved.

\[
\Delta RPM_{split} = \left[ \frac{RPM_M \cdot \frac{R_{oGFS}}{R_{oGIS}}}{R_{oGIS}} \right] - RPM_M
\]  \hspace{1cm} (1)

Main moves into neutral, split changes gear, main moves into same gear, range is not involved.

\[
\Delta RPM_{split} = \left[ \frac{RPM_M \cdot \frac{R_{oGFS}}{R_{oGIS}}}{R_{oGIS}} \right] - RPM_M
\]  \hspace{1cm} (1)

Split changes gear, main changes gear, range is not involved.

\[
\Delta RPM_{split} = \left[ \frac{RPM_M \cdot \frac{R_{oGFS}}{R_{oGIS}}}{R_{oGIS}} \right] - RPM_M
\]  \hspace{1cm} (1)

\[
\Delta RPM_{main} = \left[ \frac{RPM_M \cdot \frac{R_{oGFR}}{R_{oGIS} \cdot R_{oGIM} \cdot R_{oGIR}}}{R_{oGIS} \cdot R_{oGFM} \cdot R_{oGFR}} \right] - \left[ \frac{RPM_M}{R_{oGIS} \cdot R_{oGFM} \cdot R_{oGFR}} \right]
\]  \hspace{1cm} (2)

Main moves into neutral, split changes gear, main changes gear, range is not involved.

\[
\Delta RPM_{split} = \left[ \frac{RPM_M \cdot \frac{R_{oGFS}}{R_{oGIS}}}{R_{oGIS}} \right] - RPM_M
\]  \hspace{1cm} (1)
Appendix 2

Initial RPM differences

Split moves into neutral, main changes gear, split changes gear, range is not involved.

\[
\Delta RPM_{\text{main}} = \left[ \frac{RPM_M \cdot RoGF_R}{RoGIS \cdot RoGIM \cdot RoGIR} \right] - \left[ \frac{RPM_M}{RoGIS \cdot RoGFM} \right] \tag{2}
\]

\[
\Delta RPM_{\text{split}} = \left[ \frac{RPM_M \cdot RoGF_R \cdot RoGFM \cdot RoGS}{RoGIS \cdot RoGIM \cdot RoGIR} \right] - RPM_M \tag{3}
\]

Main changes gear, split changes gear, range is not involved.

\[
\Delta RPM_{\text{main}} = \left[ \frac{RPM_M \cdot RoGF_R}{RoGIS \cdot RoGIM \cdot RoGIR} \right] - \left[ \frac{RPM_M}{RoGIS \cdot RoGFM} \right] \tag{2}
\]

\[
\Delta RPM_{\text{split}} = \left[ \frac{RPM_M \cdot RoGF_R \cdot RoGFM \cdot RoGS}{RoGIS \cdot RoGIM \cdot RoGIR} \right] - \left[ \frac{RPM_M \cdot RoGF_R \cdot RoGFM \cdot RoGIS}{RoGIS \cdot RoGIM \cdot RoGIR} \right] = \left[ \frac{RPM_M \cdot RoGF_R \cdot RoGFM}{RoGIS \cdot RoGIM \cdot RoGIR} \right] \cdot \left[ RoGS - RoGIS \right] \tag{4}
\]

Split changes gear, main moves into neutral, range changes gear, main changes gear.

\[
\Delta RPM_{\text{split}} = \left[ \frac{RPM_M \cdot RoGF_S}{RoGIS} \right] - RPM_M \tag{1}
\]

\[
\Delta RPM_{S, range} = \frac{RPM_M}{RoGIS \cdot RoGIM} \tag{5}
\]

\[
\Delta RPM_{\text{range}} = \frac{RPM_M}{RoGIS \cdot RoGIM \cdot RoGIR} - \Delta RPM_{S, range} \tag{6}
\]

\[
\Delta RPM_{\text{main}} = \left[ \frac{RPM_M \cdot RoGF_R}{RoGIS \cdot RoGIM \cdot RoGIR} \right] - \left[ \frac{RPM_M}{RoGIS \cdot RoGFM} \right] \tag{2}
\]

Main moves into neutral, split changes gear, range changes gear, main changes gear.

\[
\Delta RPM_{\text{split}} = \left[ \frac{RPM_M \cdot RoGF_S}{RoGIS} \right] - RPM_M \tag{1}
\]

\[
\Delta RPM_{S, range} = \frac{RPM_M}{RoGIS \cdot RoGIM} \tag{5}
\]

\[
\Delta RPM_{\text{range}} = \frac{RPM_M}{RoGIS \cdot RoGIM \cdot RoGIR} - \Delta RPM_{S, range} \tag{6}
\]

Main moves into neutral, range changes gear, main changes gear, split changes gear.

\[
\Delta RPM_{S, range} = \frac{RPM_M}{RoGIS \cdot RoGIM} \tag{5}
\]

\[
\Delta RPM_{\text{range}} = \frac{RPM_M}{RoGIS \cdot RoGIM \cdot RoGIR} - \Delta RPM_{S, range} \tag{6}
\]
\[ \Delta \text{RPM}_{\text{main}} = \left[ \frac{\text{RPM}_M \cdot \text{RoGF}_R}{\text{RoG}_{IS} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] - \left[ \frac{\text{RPM}_M}{\text{RoG}_{IS} \cdot \text{RoG}_{FM}} \right] \]  

\( \text{(2)} \)

\[ \Delta \text{RPM}_{\text{split}} = \left[ \frac{\text{RPM}_M \cdot \text{RoGF}_R \cdot \text{RoGF}_{FM} \cdot \text{RoGF}_S}{\text{RoG}_{IS} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] - \left[ \frac{\text{RPM}_M \cdot \text{RoGF}_R \cdot \text{RoGF}_{FM} \cdot \text{RoG}_{IS}}{\text{RoG}_{IS} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] = \]

\[ = \left[ \frac{\text{RPM}_M \cdot \text{RoGF}_R \cdot \text{RoGF}_{FM}}{\text{RoG}_{IS} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] \cdot \left[ \text{RoG}_{FS} - \text{RoG}_{IS} \right] \]  

\( \text{(4)} \)

Formulas for different upshift scenarios in an gearbox without split.

Main changes gear, range not involved.

\[ \Delta \text{RPM}_{\text{main}} = \left[ \frac{\text{RPM}_M \cdot \text{RoGF}_R}{\text{RoG}_{IN} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] - \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}} \right] = \]

\[ = \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN}} \right] \cdot \left[ \frac{\text{RoGF}_R}{\text{RoG}_{IM} \cdot \text{RoG}_{IR}} - \frac{1}{\text{RoG}_{FM}} \right] \]  

\( \text{(7)} \)

Main moves into neutral, range changes gear, main changes gear.

\[ \Delta \text{RPM}_{\text{range}} = \frac{\text{RPM}_M}{\text{RoG}_{IN} \cdot \text{RoG}_{IM}} \]  

\( \text{(8)} \)

\[ \Delta \text{RPM}_{\text{range}} = \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] - \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}} \right] = \]

\[ = \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN}} \right] \cdot \left[ \frac{1}{\text{RoG}_{IR}} - 1 \right] \]  

\( \text{(9)} \)

\[ \Delta \text{RPM}_{\text{main}} = \left[ \frac{\text{RPM}_M \cdot \text{RoGF}_R}{\text{RoG}_{IN} \cdot \text{RoG}_{IM} \cdot \text{RoG}_{IR}} \right] - \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}} \right] = \]

\[ = \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN}} \right] \cdot \left[ \frac{\text{RoGF}_R}{\text{RoG}_{IM} \cdot \text{RoG}_{IR}} - \frac{1}{\text{RoG}_{FM}} \right] \]  

\( \text{(7)} \)

Formula for upshifts in an gearbox without split or range.

\[ \Delta \text{RPM}_{\text{main}} = \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN} \cdot \text{RoG}_{IM}} \right] - \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN} \cdot \text{RoG}_{FM}} \right] = \left[ \frac{\text{RPM}_M}{\text{RoG}_{IN}} \right] \cdot \left[ \frac{1}{\text{RoG}_{IM}} - \frac{1}{\text{RoG}_{FM}} \right] \]  

\( \text{(10)} \)
Formulas for different downshift scenarios in an gearbox with split, main and range:

Split changes gear, main and range is not involved.

\[
\Delta RPM_{\text{split}} = \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM} \cdot RoG_{FS}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} - \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM} \cdot RoG_{IS}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} = \\
= \left[ \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} \right] \cdot \left[ RoG_{FS} - RoG_{IS} \right] \quad (11)
\]

Main moves into neutral, split changes gear, main moves into same gear, range is not involved.

\[
\Delta RPM_{\text{split}} = \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM} \cdot RoG_{FS}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} - \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM} \cdot RoG_{IS}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} = \\
= \left[ \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} \right] \cdot \left[ RoG_{FS} - RoG_{IS} \right] \quad (11)
\]

Split changes gear, main changes gear, range is not involved.

\[
\Delta RPM_{\text{main}} = \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM} \cdot RoG_{FS}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} - \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM} \cdot RoG_{IS}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} = \\
= \left[ \frac{RPM_M}{RoG_{FS} \cdot RoG_{FM}} \right] \cdot \left[ 1 - \frac{RoG_{IR} \cdot RoG_{IM}}{RoG_{FR} \cdot RoG_{FM}} \right] \quad (14)
\]

Main moves into neutral, split changes gear, main changes gear, range is not involved.

\[
\Delta RPM_{\text{split}} = \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM} \cdot RoG_{FS}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} - \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM} \cdot RoG_{IS}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} = \\
= \left[ \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR}} \right] \cdot \left[ RoG_{FS} - RoG_{IS} \right] \quad (11)
\]

Split moves into neutral, main changes gear, split changes gear, range is not involved.

\[
\Delta RPM_{\text{main}} = \frac{RPM_M}{RoG_{FS} \cdot RoG_{FM}} - \frac{RPM_M \cdot RoG_{IR} \cdot RoG_{IM}}{RoG_{FS} \cdot RoG_{FM} \cdot RoG_{FR} \cdot RoG_{FM}} = \\
= \left[ \frac{RPM_M}{RoG_{FS} \cdot RoG_{FM}} \right] \cdot \left[ 1 - \frac{RoG_{IR} \cdot RoG_{IM}}{RoG_{FR} \cdot RoG_{FM}} \right] \quad (14)
\]
Appendix 2
Initial RPM differences

\[ \Delta RPM_{\text{split}} = RPM_M - \left[ \frac{RPM_M \cdot RoGIR \cdot RoGIM \cdot RoGIS}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] = \]

\[ = RPM_M \cdot \left[ 1 - \frac{RoGIR \cdot RoGIM}{RoGF_R \cdot RoGF_M} \right] \quad (15) \]

Main changes gear, split changes gear, range is not involved.

\[ \Delta RPM_{\text{main}} = \left[ \frac{RPM_M}{RoGF_S \cdot RoGF_M} \right] - \left[ \frac{RPM_M \cdot RoGIR \cdot RoGIM}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] = \]

\[ = \left[ \frac{RPM_M}{RoGF_S \cdot RoGF_M} \right] \cdot \left[ 1 - \frac{RoGIR \cdot RoGIM}{RoGF_R \cdot RoGF_M} \right] \quad (14) \]

\[ \Delta RPM_{\text{split}} = RPM_M - \left[ \frac{RPM_M \cdot RoGIS}{RoGF_S} \right] = RPM_M \cdot \left[ 1 - \frac{RoGIS}{RoGF_S} \right] \quad (16) \]

Split changes gear, main moves into neutral, range changes gear, main changes gear.

\[ \Delta RPM_{\text{split}} = \left[ \frac{RPM_M \cdot RoGIR \cdot RoGIM}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] - \left[ \frac{RPM_M \cdot RoGIR \cdot RoGIM \cdot RoGIS}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] = \]

\[ = \left[ \frac{RPM_M \cdot RoGIR \cdot RoGIM}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] \cdot \left[ RoGF_S - RoG_{IS} \right] \quad (11) \]

\[ \Delta RPM_{\text{range}} = - \frac{RPM_M \cdot RoGIR}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \quad (17) \]

\[ \Delta RPM_{\text{main}} = \left[ \frac{RPM_M}{RoGF_S \cdot RoGF_M} \right] - \left[ \frac{RPM_M \cdot RoGIR}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] = \]

\[ = \left[ \frac{RPM_M}{RoGF_S \cdot RoGF_M} \right] \cdot \left[ 1 - \frac{RoGIR \cdot RoGIM}{RoGF_R \cdot RoGF_M} \right] \quad (18) \]

Main moves into neutral, split changes gear, range changes gear, main changes gear.

\[ \Delta RPM_{\text{split}} = \left[ \frac{RPM_M \cdot RoGIR \cdot RoGIM \cdot RoGIS}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] - \left[ \frac{RPM_M \cdot RoGIR \cdot RoGIM \cdot RoGIS}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] = \]

\[ = \left[ \frac{RPM_M \cdot RoGIR \cdot RoGIM}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \right] \cdot \left[ RoGF_S - RoG_{IS} \right] \quad (11) \]

\[ \Delta RPM_{\text{range}} = - \frac{RPM_M \cdot RoGIR}{RoGF_S \cdot RoGF_M \cdot RoGF_R} \quad (17) \]
Appendix 2

Initial RPM differences

\[
\Delta RPM_{\text{range}} = \left[ \frac{RPM_M}{RoGFS \cdot RoGF} \right] - \left[ \frac{RPM_{M'} \cdot RoGIR}{RoGFS \cdot RoGF_{M'} \cdot RoGF_{FR}} \right] =
\]

\[
= \left[ \frac{RPM_M}{RoGFS \cdot RoGF} \right] \cdot \left[ 1 - \frac{RoGIR}{RoGF_{FR}} \right]
\]  

(18)

Main moves into neutral, range changes gear, main changes gear, split changes gear.

\[
\Delta RPM_{S \cdot \text{range}} = - \frac{RPM_{M'} \cdot RoGIR}{RoGFS \cdot RoGF_{M'} \cdot RoGF_{FR}}
\]  

(17)

\[
\Delta RPM_{\text{range}} = \left[ \frac{RPM_M}{RoGFS \cdot RoGF} \right] - \left[ \frac{RPM_{M'} \cdot RoGIR}{RoGFS \cdot RoGF_{M'} \cdot RoGF_{FR}} \right] =
\]

\[
= \left[ \frac{RPM_M}{RoGFS \cdot RoGF} \right] \cdot \left[ 1 - \frac{RoGIR}{RoGF_{FR}} \right]
\]  

(18)

\[
\Delta RPM_{\text{main}} = \left[ \frac{RPM_M}{RoGFS \cdot RoGF} \right] - \left[ \frac{RPM_{M'} \cdot RoGIR \cdot RoGIM}{RoGFS \cdot RoGF_{M'} \cdot RoGF_{FR} \cdot RoGF_{FM}} \right] =
\]

\[
= \left[ \frac{RPM_M}{RoGFS \cdot RoGF} \right] \cdot \left[ 1 - \frac{RoGIR \cdot RoGIM}{RoGF_{FR} \cdot RoGF_{FM}} \right]
\]  

(14)

\[
\Delta RPM_{\text{split}} = RPM_M - \left[ \frac{RPM_{M'} \cdot RoGIS}{RoGFS} \right] = RPM_M \cdot \left[ 1 - \frac{RoGIS}{RoGFS} \right]
\]  

(16)

Formulas for different downshift scenarios in an gearbox without split.

Main changes gear, range is not involved.

\[
\Delta RPM_{\text{main}} = \left[ \frac{RPM_M}{RoGIN' \cdot RoGF_{FM}} \right] - \left[ \frac{RPM_{M'} \cdot RoGIR \cdot RoGIM}{RoGIN' \cdot RoGF_{M'} \cdot RoGF_{FR} \cdot RoGF_{FM}} \right] =
\]

\[
= \left[ \frac{RPM_M}{RoGIN' \cdot RoGF_{FM}} \right] \cdot \left[ 1 - \frac{RoGIR \cdot RoGIM}{RoGF_{FR} \cdot RoGF_{FM}} \right]
\]  

(19)

Main moves into neutral, range changes gear, main changes gear.

\[
\Delta RPM_{S \cdot \text{range}} = - \frac{RPM_M}{RoGIN' \cdot RoGF_{FM}}
\]  

(20)

\[
\Delta RPM_{\text{range}} = \left[ \frac{RPM_M}{RoGIN' \cdot RoGF_{FM}} \right] - \left[ \frac{RPM_{M'} \cdot RoGIR}{RoGIN' \cdot RoGF_{M'} \cdot RoGF_{FR}} \right] =
\]

\[
= \left[ \frac{RPM_M}{RoGIN' \cdot RoGF_{FM}} \right] \cdot \left[ 1 - \frac{RoGIR}{RoGF_{FR}} \right]
\]  

(21)

\[
\Delta RPM_{\text{main}} = \left[ \frac{RPM_M}{RoGIN' \cdot RoGF_{FM}} \right] - \left[ \frac{RPM_{M'} \cdot RoGIR \cdot RoGIM}{RoGIN' \cdot RoGF_{M'} \cdot RoGF_{FR} \cdot RoGF_{FM}} \right] =
\]

\[
= \left[ \frac{RPM_M}{RoGIN' \cdot RoGF_{FM}} \right] \cdot \left[ 1 - \frac{RoGIR \cdot RoGIM}{RoGF_{FR} \cdot RoGF_{FM}} \right]
\]  

(19)
Appendix 2
Initial RPM differences

Formula for downshift in an gearbox without split or range.

\[
\Delta RPM_{main} = \left[ \frac{RPM_M}{RoG_{IN} \cdot RoG_{FM}} \right] - \left[ \frac{RPM_M \cdot RoG_{IM}}{RoG_{IN} \cdot RoG_{FM} \cdot RoG_{FM}} \right]
\]

\[
= \left[ \frac{RPM_M}{RoG_{IN} \cdot RoG_{FM}} \right] \cdot \left[ 1 - \frac{RoG_{IM}}{RoG_{FM}} \right]
\]
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1. **Program requirements and starting the program**

   The program is only tested on MATLAB 7.7.0 (R2008b) and 7.9.0 (R2009b), but should work with all recent versions of MATLAB, however, version 7.6.0 (2008a) or newer is recommended. A high resolution screen or multiple screens is also recommended. Future compatibility is assumed.

   To start the program, first start MATLAB and set the MATLAB work path to the folder containing the program (exact path might differ from user to user). Verify that the programs files are shown under the Current folder panel, and right-click on main_gui.m and select “Run file”. Note: If “Run file” doesn’t exist, main_gui.fig is probably selected instead of main_gui.m. Do not open main_gui.fig, then the program won’t work properly for that session. If the wrong file was opened, close the program (but not MATLAB) and run the correct file.

   When the program starts, the subfolders will be added to the MATLAB path so the program (and MATLAB) can access the files in the subfolders.

1.1. **Program delivery**

   The program will be delivered to Scania in three ways:

   1. The program will be uploaded to the gearbox_dev folder on the Scania intranet. The path is: Gearbox_Dev\NTBM\Artikel- och systeminfo\Synkar\Calculations synchronizers

   2. A zipped file, containing all program files in the original state will be uploaded in the same folder.

   3. A zipped and password protected file will be stored in an USB memory.
      The password is: scaniasynk
      This is only to make sure a user doesn’t edit the program files by accident.
2. Using the program

The program contains two modes, the Complete gearbox mode and the Single sync. mode. Blue input fields and italic text contains tooltips. To view the tooltip, hover the cursor over the object until the tooltip appears.

2.1. Complete gearbox mode.

When the program is started, the Main window appears. In this window, the user can define the gearbox to calculate or switch to Single sync. mode. Figure 1 shows the Main window with the features numbered to simplify the explanation.

![Main window with numbered features.](image)

The suggested working order for a new user is described in 1-7. 8-10 are optional features.

1. Quickload a standard gearbox to speed up the input process.
2. Chose what output to plot. It is recommended to only plot the relevant data to speed up calculation time and simplify evaluation. All static values are always displayed.
3. Verify that the Gearbox type, Gears in main gearbox and Include crawl gear is correct, and chose a Shift mode.
4. (Optional) If two different calculations will be compared, input a name for the first calculation. The name will appear in the figure title to help differentiate the two cases. The comparison is presented in section 3.4 - Comparing results.
5. Work your way through the Gearbox, Synchronizer and Calculation windows and change desired inputs.
6. Press Calculate to calculate. The program will show the Main window again so the user can review the output. Depending on user input, figures containing plots might appear.
7. Recap of maximum static output values per synchronizer. All values are available in MATLAB command window. The popupmenu is used to switch between different synchronizers. Only the worst-case values are shown here. The raw data can be found in the MATLAB command window.
8. Enables *Single sync.* mode, presented in section 2.2 - *Single sync mode.*
9. Buttons to clear MATLAB command window and close all figures (belonging to the current case).
10. The menu structure is shown in Figure 6.

Figure 2 shows the *Gearbox* window.

1. The input field for the gear inertia for direct gear of a G or GR gearbox disappears, since the gear is assumed to be attached to the input shaft. That means, for the input field of the input shaft the requested inertia is input shaft + attached gear.
2. Inputs for measured drag coefficients. The data for GRS905 is assumed to be valid for GR and 875/925 gearboxes too, but is changeable by the user if more measurements are performed. The coefficients are explained in the report.
3. Checkboxes to override the drag, by either changing the viscosity or removing the drag completely. The friction from the selector finger / shift sleeve contact will NOT be affected by this.
5. Loads *.gbd* gearbox data.
Figure 3 shows the *Synchronizer* window.

![Synchronizer window](image)

<table>
<thead>
<tr>
<th>Gears</th>
<th>LS</th>
<th>HS</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>LR</th>
<th>HR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon fibre? (prior Molybdenum)</td>
<td>☑️</td>
<td>☑️</td>
<td>☑️</td>
<td>☑️</td>
<td>☑️</td>
<td>☑️</td>
<td>☑️</td>
<td>☑️</td>
<td>☑️</td>
<td>☑️</td>
</tr>
</tbody>
</table>

2. Saves or loads synchronizer data in `.sync` format.
Figure 4 shows the Calculation window.

1. Define force, force buildup and force delay, and plot the forces to verify.
2. Define friction properties. There is space to replace the cone CoF with controls for a changing CoF, as described in the report.
3. Material limits, that will be plotted in the p-v diagram and used to calculate safety margins.
4. Chose what gearshifts to include.
5. Saves or loads calculation data in .cav format.
2.2. Single sync mode

Figure 5 shows the *Single sync* mode.

1. Define friction properties. There is space to replace the cone CoF with controls for a changing CoF, as described in the report.
2. Define force, force buildup and force delay, and plot the force to verify.
3. Material limits, that will be plotted in the p-v diagram and used to calculate safety margins. Hover the fields with the cursor to see the tooltips indicating what field belong to the different properties.
4. Output of static calculations.
5. Saves or loads *Single sync* data in *.sins* format.
2.3. Menu structure

The menu structure is shown in Figure 6. Original values displays pictures of the original input values for standard gearboxes.

![Menu structure diagram]

Figure 6. Menu structure.

2.4. Accessing the help

Help texts are added for every function file in a standard MATLAB help format. To access the help, type `help function_name` (e.g. `help fun_plot_grs`) in the MATLAB command window, to get the output in the MATLAB command window. To get the output in the MATLAB documentation, type `doc function_name` instead. Note that the program need to be started before accessing the help, since the subfolders are added to the MATLAB path when the program starts.

Additional information about the files are written as comments in the code, which can be accessed by opening the function file in the MATLAB editor by double clicking the file in the MATLAB file structure.

This manual can be accessed from the program via Menu, Help, Manual. The master thesis report can be accessed via Menu, Help, Documentation.
3. Output

Notice that the RPM difference is calculated in the friction surface e.g. the range synchronizer surface is changing from the main shafts rpm to zero when downshifting. When the gear changes are similar, only difference is high or low range, only one of the case will be plotted.

The following outputs are available.

3.1. Detailed results (Command window)

Complete gearbox mode only. The following data will be presented for all gearshifts and for split, main and range (when available):

- \( drpm \) = Initial RPM difference.
- \( W/mm^2 \) = Max power/area in the synchronizer surface.
- \( J/mm^2 \) = Max work/area in the synchronizer surface.
- \( Kgm^2_E \) = The reflected inertia at the engines side (for each synchronizer).
- \( Kgm^2_W \) = The reflected inertia at the wheel’s side (for each synchronizer). Note: infinite when the shaft is driven by the wheels.
- \( Ns \) = Shift impulse.
- \( \%MP \) = Power/area safety margin.
- \( \%MW \) = Work/area safety margin.
- \( Ts \) = time to synchronize.
- \( T_{above} \) = Duration of power/area over a user-specified limit.
- \( Tc/Ti@BTL \) = Cone torque / index torque. \( Tc/Ti \) when the force equals the BTL.

3.2. Max values (Orange panel)

In the orange panel in the Main window, the max values from the command window is presented. The user can chose which synchronizer is shown with the popup menu.

3.3. Plotted data

The following data can be plotted:

- Sliding velocity [m/s], RPM difference or neither.
- Power/area [W/mm2], power [W] or neither.
- Work/area [J/mm2], work [J] or neither.
- P-v diagram.

In the p-v diagram, the material limits can be set by the user in the Calculation window, and needs to be determined experimentally. Figure 7 shows an example of plotted output values.
1. **MATLAB Data Cursor** is a tool to read data directly from a graph. To add a data cursor, press the button marked 1. in the figure and click on a graph. To move it, either drag it with the mouse or use the arrow keys. To add more than one data cursor, hold the Alt button and click on another graph (with data cursor enabled).

2. The data cursor shows the x and y values of the green line at the position of the black square.

3. **MATLAB Brush/Select Data** is a tool for exporting the graphs values to a workspace variable. A good use of this would be to be able to compare two or more graphs plotted in different axes. To save a variable, enable Brush/Select Data, mark the whole graph(s). Right-click on the selected data and select “Create Variable”. Name the variable and it appears in your MATLAB Workspace, where it can be modified or compared with other variables.

4. The functionality of the legend has been extended to be able to show and hide single graphs by clicking at their legend entry. In Figure 7, the graphs for 3-6 and 4-7 are disabled. Note that their legend entries are greyed to indicate that they graphs are hidden. To be able to use this functionality, all plot edit controls in MATLAB (Edit plot, Zoom In, Zoom Out, Pan, Rotate 3D, Data Cursor, Brush/Select Data) have to be disabled.
3.4. Comparing results

To be able to compare plots, a comparison mode was implemented. To use it, press the *Compare* button marked 4. in Figure 1. Another *Main window* opens and the user can perform two calculations simultaneously. To separate the two cases, it is possible to input a figure name. To be able to compare two graphs directly, it’s possible to use the *Brush/Select Data* tool as explained in section 3.3 - *Plotted data*. Table 1 shows the figure numbers for case 1 and case 2 with respect to gear changes and what data to plot.

<table>
<thead>
<tr>
<th>Gear changes</th>
<th>Velocity</th>
<th>Power</th>
<th>Work</th>
<th>P-v</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 step up</td>
<td>1, 25</td>
<td>7, 31</td>
<td>13, 37</td>
<td>19, 43</td>
</tr>
<tr>
<td>1 step down</td>
<td>2, 26</td>
<td>8, 32</td>
<td>14, 38</td>
<td>20, 44</td>
</tr>
<tr>
<td>2 step up</td>
<td>3, 27</td>
<td>9, 33</td>
<td>15, 39</td>
<td>21, 45</td>
</tr>
<tr>
<td>2 step down</td>
<td>4, 28</td>
<td>10, 34</td>
<td>16, 40</td>
<td>22, 46</td>
</tr>
<tr>
<td>3 step up</td>
<td>5, 29</td>
<td>11, 35</td>
<td>17, 41</td>
<td>23, 47</td>
</tr>
<tr>
<td>3 step down</td>
<td>6, 30</td>
<td>12, 36</td>
<td>18, 42</td>
<td>24, 48</td>
</tr>
</tbody>
</table>
4. Warning and error messages

The following warning messages are added to the program. Note that MATLAB also uses warning messages (possibly with different formatting), so more messages than those presented here might appear.

4.1. ERROR: Bad inputs, aborting

The following conditions must be satisfied to calculate, else a warning message will be issued and the program will abort the calculation. To add more conditions, see section 5.1 - Verify more inputs.

- Either upshift, downshift or both selected.
- At least one of the following selected: 1 step shifts, 2 step shifts, 3 step shifts.
- A time step between 0.01 and 0.0005 s.
- A max simulation time between 0.5 and 5 s.
- A max engine RPM greater than 0.
- Type G gearbox with 3 gears and only 3 step shifts are unavailable since there are no possible gearshifts to calculate.
- All number of teeth needs to be positive values.
- The oil temperature must be between 0 and 100 °C.
- The oil quantity must be between 12 and 20 litres.
- For Single sync. mode: A time step between 0.01 and 0.0005 s.

Note that these are the conditions to be able to run the program, but not for getting correct answers. If e.g. cone width is zero, the cone area cannot be calculated and therefore power/area cannot be determined (MATLAB will answer NaN, not a number or Inf, infinite).

4.2. ERROR: Failed to synchronize

This warning is issued if synchronization isn’t complete before the max simulation time is reached. Solutions are:

- Longer simulation time.
- Increase cone torque by increasing the force, friction properties or cone radius.
- Decrease the inertia.
- Change the number of teeth to reduce reflected inertia.

4.3. Warning: One or more values for the drag was overwritten (by 0)

This warning is issued if both the oil level and RPM dependence coefficients are set to zero, which will cause the drag to become impossible to calculate.

4.4. Warning: Drag approaches infinity and was overwritten by default values

*Single sync.* only. This warning message is issued if the drag approaches infinity, which could happen with certain input combinations. If the drag reaches 150% of its initial value, this value will override the calculated value and the warning will be issued.

4.5. MATLAB error messages.

A MATLAB error message is formatted red, and contains information about what caused the error and where in the program that is. Unfortunately, the use of try-catch cases which was needed to display custom error messages overwrites the information about what row caused the error. To determine what row caused the error, suppress the try-phrase and the catch block by comment (highlight, Ctrl + R) them. When done, uncomment (highlight, Ctrl + T) them again.
5. **Editing the program**

If something is missing in the program, here is a few hints on how to add them. Basic MATLAB knowledge is assumed. If additional information is needed, the MATLAB documentation and the GUI tutorials at http://blinkdagger.com/matlab/ should suffice.

5.1. **Verify more inputs (basic)**

The input verification (complete gearbox only) can be found in the function file `fun_verifyinput.m`. A list of values being verified and conditions that need to be satisfied are available in section 4.1 - ERROR: Bad inputs, aborting. This example will explain how to add a max limit of 3000 RPM for the maximum engine RPM (which is not in the program, however, maximum engine RPM needs to be greater than 0 so the variable `rpmm` exists).

1. Open `fun_verifyinput.m` in the MATLAB editor by double clicking the file in the MATLAB file structure. It’s recommended to open `main_gui.fig` in GUIDE by right click on `main_gui.fig` in the MATLAB file structure, and “Open in GUIDE”, and locate the input fields that needs verification before calculating by right clicking on the background panel and select “Send to Back” until the correct window is visible. In the example, the Calculation window is requested.
2. Double click on the requested input field, and the MATLAB Inspector appears. Find the Tag property and note the Tag.
3. All the values inputted in the GUI are available in the function file via the `handles` structure. To access the value, the input field Tag property is needed. The following command will retrieve the input value, convert it to `double` (“number” instead of character) and save it to a variable. `variable_name = str2double(get(handles.Tag,'String'));` Note that the `variable_name` and Tag needs to be changed, where Tag is the property from 2. In this example, `rpmm = str2double(get(handles.edit_input_rpmm,'String'));`
4. When the value is stored in a variable, make an if-case (or similar) and assign the variable `check_variable_name` the value 1 if the input is correct, and 0 if the input is outside the desired range. It is recommended to display a message indicating what variable didn’t pass the check. In this example:
   ```matlab
   if rpmm > 3000
       check_rpmm_upperlimit = 0;
       disp('Maximum RPM is to high')
   else
       check_rpmm_upperlimit = 1;
   end
   ```
5. In the bottom of the file, add the check-variable in the same way as the other check-variables. If all check-variables are 1, the variable `verified` will be 1 and the program will calculate. If one check-variable is 0, the variable `verified` will be 0 and the calculation aborted.

5.2. **Adding more quickstart buttons (basic)**

Quickstart buttons are included to speed up the workflow when using the program. At program creation, quickstart buttons for GR905, GRS905, GRSO905 and GRSO925 are included. A user can add another quickstart button by following these steps:

1. Create two saved files from the program in the folder `default_gearboxes`, one `gearbox definition (.gbd)` and one `synchronizer (.sync)`.
2. Right click on `main_gui.fig` in the MATLAB file structure, and “Open in GUIDE”. Right click on the background panel and select “Send to Back” until the main window is visible.
3. Duplicate one of the existing quickstart buttons by selecting it and pressing `Ctrl + D`. Double click on the new button and edit the Tag-property, which is the name MATLAB uses for the button. `pushbutton_quickload_custom` or something similar is recommended. It is also recommended to edit the String-property, which is the text displayed at the button, and reposition the button by dragging it or setting the Position-property.
4. Save the file and open `main_gui.m`, and locate
function pushbutton_quickload_custom_callback(hObject, eventdata, handles). Below, add the following text and edit to match the new files.

% COPY HERE
filename_gearbox = ('filename.gbd'); %Replace with the new filename
pathname_gearbox = ('default_gearboxes');
filename_synchronizers = ('filename.sync'); %Replace with the new filename
pathname_synchronizers = ('default_gearboxes');
set(handles.popupmenu_gearbox_type,'Value',3) %Edit the number, G=1, GR=2, GRS=3
set(handles.popupmenu_input_no_main_gears,'Value',1); %Edit the number, enter
"Number of gears in main gearbox - 2".
set(handles.checkbox_input_include_crawl,'Value',1); %Edit the number, 1=crawl enabled, 0 = crawl disabled

popupmenu_gearbox_type_Callback(hObject, eventdata, handles)

pushbutton_load_gearbox_Callback(hObject, eventdata, handles,
filename_gearbox, pathname_gearbox)
pushbutton_load_synchronizer_Callback(hObject, eventdata, handles,
filename_synchronizers, pathname_synchronizers)
pushbutton_main_Callback(hObject, eventdata, handles)
disp('filename loaded') %Replace with the new filename

% STOP COPYING

5. Save, and start the program as normal.

5.3. Adding input fields (basic)

If another input field is needed, the user have to add it.

1. Right click on main_gui.fig in the MATLAB file structure, and “Open in GUIDE”. Right click on the background panel and select “Send to Back” until the correct window is shown.
2. Duplicate one of the existing input fields by selecting it and pressing Ctrl + D. Double click on the new field and edit the Tag-property, which is the name MATLAB uses for the field. It is recommended to edit the String-property, which is the default value displayed in the field, and reposition the field by dragging it or setting the Position-property.
3. To enable saving and loading of the new value, open the main_gui.m in the MATLAB editor. Locate function pushbutton_save_synchronizer_Callback(hObject, eventdata, handles) if the edit field is added to the synchronizer window. If the added field is in the gearbox, calculation or single sync window, locate the appropriate function callback.
4. Just above [filename_synchronizers, pathname_synchronizers, filterindex], add
save_property = get(handles."name_of_new_input_field","Value");
where “property” should be replaced by what the input field is containing (e.g. number_of_cones) and “name_of_new_input_field” with the Tag-property given in step 2.
5. Locate function pushbutton_load_synchronizer_Callback(hObject, eventdata, handles) if the edit field is added to the synchronizer window. If the added field is in the gearbox, calculation or single sync window, locate the appropriate function callback.
6. Just above
else
    if save_no_main_gears
    add set(handles."name_of_new_input_field","String",save_property);
    where “name_of_new_input_field” and “property” should be replaced with the names used in step 4.
7. Start the program and test the buttons. Note that the first time every file is loaded, there will be an error message stating that the new variable doesn’t exist. Load – save – load the file to solve that error. Verify that it works next time the file is loaded.
5.4. Plotting additional parameters (advanced)

To reuse the plotting functions, the data needs to be in the same format as the rest of the data, that is, in a matrix with the gearshifts on different rows, and the different time steps in the columns, as shown in the example.

\[
\text{radpers\_split } =
\]  

\[
\begin{array}{cccccc}
\text{(t = 0) } & \text{t=0.005} & \text{t=0.01} & \text{t=0.015} & \text{t=0.02} & \text{t=0.025} \\
\text{(Gear 1 to 4)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\text{(Gear 2 to 5)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\text{(Gear 3 to 6)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\text{(Gear 4 to 7)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\text{(Gear 5 to 8)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\text{(Gear 6 to 9)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\text{(Gear 7 to 10)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\text{(Gear 8 to 11)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\text{(Gear 9 to 12)} & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX & X.XXXX \\
\end{array}
\]

When the data is calculated, call the function \text{fun\_plot\_grs} (or \text{fun\_plot\_g}, \text{fun\_plot\_gr}, depending on gearbox). The following input arguments are needed for normal plots, with the bold arguments needs change:

\text{skip\_to, max\_gear, xdata\_plot\_main, xdata\_plot\_range, xdata\_plot\_split, ydata\_plot\_main, ydata\_plot\_range, ydata\_plot\_split, xlabeltext, ylabeltext, fignr, legendtext, figname, bol\_wear, 0, 0, 0, 0}

5.5. Adding a new shift mode (advanced)

1. Only for GRS type gearbox, G or GR: jump to step 2. Locate \text{popupmenu\_shift\_mode} in the \text{main\_gui.fig}, and in the string property, add the shift case below the other cases.
2. Locate \text{function popupmenu\_gearbox\_type\_Callback(hObject, eventdata, handles)} in \text{main\_gui.m} in the MATLAB editor and add the shift mode in the if-case.
3. Update \text{fun\_drpm\_split\_grs}, \text{fun\_drpm\_main\_grs} and \text{fun\_drpm\_range\_grs} (or relevant function files) with the new shift mode. These functions contain the initial RPM difference for the synchronizer, and theoretical equations may have to be derived.
4. The reflected inertia and possibly other parameters might have to be updated to fit the new shift mode.
5. In the cells
\[
\text{%% Radians per second over time for split/main/range}
\]
add the shift mode in the same way as previous, similar, entries are added. Calculate easy cases to verify that the cone torque, the drag and the friction from the selector fingers act in the right direction.
Test rig output

Figure 1 shows the trend lines for the test rig input and output. The max axial force and initial RPM difference is determined by the test script. The vertical lines shows where the random test cycles used for verification is. The cycles where chosen without consulting the trend line.

Figure 1. Test rig trend lines for input and output.

Figure 2 to Figure 5 shows the axial force during the buildup phase and the RPM difference at every time step for four of the five cycles. Cycle 2200 is shown in the report, section 5.2 - Adapting to measured values, especially Figure 34.
Figure 2. Cycle 980.

Figure 3. Cycle 1960.
Figure 4. Cycle 2780.

Figure 5. Cycle 3040.