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9 IFK

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9TH IFK | CONFERENCE PROCEEDINGS

9 IFK

MODERN FLUID POWER

CHALLENGES · RESPONSIBILITIES · MARKETS

24th - 26th MARCH 2014 | PROCEEDINGS

9TH INTERNATIONAL FLUID POWER CONFERENCE

Vol. 3

VOLUME 3 - CONFERENCE: WEDNESDAY, MARCH 26TH

9TH INTERNATIONAL FLUID POWER CONFERENCE (9TH IFK)

24th - 26th of March 2014
Aachen, Germany

Volume 1 - Symposium: Monday, March 24th

Volume 2 - Conference: Tuesday, March 25th
- Scientific Poster Session

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WELCOME TO THE 9th IFK!

On behalf of the Organizing Committee of the 9th IFK we are delighted that you have chosen to join us for this year's International Fluid Power Conference. Welcome to Aachen!

The IFK, one of the world's largest scientific conferences on fluid power, unites scientists with industry in an international forum to exchange knowledge in the area of hydraulic and pneumatic drives and control systems. The first conference was held in 1974 in Aachen. Since 1998 the Institute for Fluid Power Drives and Controls (IFAS) at RWTH Aachen University and the Institute of Fluid Power (IFD) at TU Dresden alternately organise the conference every two years.

This year we host 141 scientific contributions and speakers and attendees from 26 countries are registered. A special feature of the 9th IFK is the integrated Digital Fluid Power workshop (DFP) on Monday. In order to provide high quality of scientific contributions the authors of this year's scientific papers had the possibility to have their papers reviewed by a board of renowned scientists. Apart from the scientific programme an exhibition of 36 different companies provide the possibility to discover novel fluid power products.

The program begins on Monday morning with a symposium where researchers from mainly universities and other research facilities have the opportunity to present their research projects to a wide international community of scientists. In the evening of the first day all participants are invited to the opening event that marks the start of the exhibition.
The second day begins with the opening address followed by three plenary lectures. On Tuesday and Wednesday, there are seven groups of three parallel sessions of presentations covering a wide variety of application and technology oriented topics. In the evening of the second day, the conference banquet is held at the Coronation Hall of the Aachen town hall. The banquet will be followed by an after show party with cool beverages and snacks in the Aula Carolina.
The conference ends with two final lectures and the farewell address on Wednesday afternoon followed by the laboratory party at the IFAS.
During the conference, the cultural program offers a possibility to explore the antique surrounding of Aachen and the excursion following the conference on Thursday and Friday invites to learn more about industrial companies now and then.

Finally, we would like to express our thanks to all members of the program and organizing committee, scientific advisory board, plenary and keynote speakers, speakers, reviewers, chairmen and exhibitors for their time and commitment helping to conduct another successful and rewarding conference and we hope that you will enjoy the 9th IFK in Aachen.

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09:00 - 10:30 a.m.

GROUP J - SIMULATION

GROUP K - MOBILE APPLICATIONS

GROUP L - COMPONENTS

11:00 - 12:30 a.m.

GROUP M - RENEWABLE ENERGY

GROUP N - SYSTEMS

GROUP O - COMPONENTS

01:30 - 03:00 p.m.

GROUP P - COMPONENTS

GROUP Q - SIMULATION & VALIDATION

GROUP R - AUTOMOTIVE TECHNOLOGY

03:30 - 05:00 p.m.

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GROUP U - FLUIDS AND SYSTEMS

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Development of High Speed Electrical Drives for Mobile Machinery – Challenges and Potential Solutions

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Electrical drive technology for traction drives of mobile machinery is yet a niche application, due to low power density and high costs. Compared to the established hydraulic-mechanical drive technology, electrical automotives have some advantages, such as temporary emission-free operation and better partial-load efficiency. For applications in the automotive sector, power density increases significantly by increasing the speed of the electrical motor.

Goal of the project "High Speed Electrical Drives" is to show this potential of high speed electrical drives for mobile machinery and to prove their suitability. The high speed requires the development of appropriate electrical machines, control units and gears. Relevant research and development tasks regard acoustics, concepts for lubrication and cooling, calculation of losses and the fulfilment of function and durability.

Keywords: electrical drive, electrical motor, mobile machinery, high speed planetary gear, TEAM
Target audience: Mobile Machinery

1 Joint Research Project "TEAM" - Workgroup "High Speed Electrical Drives"

The goal of the joint research project TEAM – "Development of **T**echnologies for **E**nergy-s**a**ving Drives of **M**obile Machinery", funded by the German Federal Ministry of Education and Research, is to improve the energy efficiency of mobile machinery. The consortium consists of industrial partners and university institutes. The joint research project is divided into five workgroups, each dealing with different aspects to improve energy efficiency of mobile machinery – from technology to working process. An overview of the workgroups is shown in Table 1.

No.	TEAM-workgroups
1	Determination of the energy efficiency of mobile machinery
2	Calculation of the interaction between machine and process
3	Optimised combustion engine for hybrid drives
4	High Speed Electrical Drives
5	Technology demonstrator "Green Wheel Loader"

Table 1: Topics of the joint research project TEAM /1/.

This publication focuses on the workgroup "High Speed Electrical Drives".

2 Motivation and Goal

Due to legal requirements to reduce CO_2 emissions and rising prices of fossil fuel, research on alternative drive concepts with improved energy efficiency advances in the construction and agricultural machinery sector. The automotive industry as a pioneer in developing innovative drives shows a trend towards electrical drives.

When compared to the established hydraulic-mechanical drive technology, electro-mechanical drives have some advantages, such as a better partial-load efficiency, less noise, reduced standby losses and an easier handling of control behavior. However, disadvantages are the currently high investment costs for classical electrical motors, on the one hand and for the modification to hybrid-electrical vehicle systems on the other hand /2/. Furthermore, in addition to high energy efficiency, a high power density is essential for electrical drives to be economical.

So far, caused by the above mentioned high costs and low power density, electrical drive technology for traction drives of mobile machinery is only used for applications where vehicles are operated in buildings (e.g. forklifts), or where large amounts of power are necessary (e.g. mining-vehicles) /3/. Electrical drives are used in forklifts, for emission-free operation inside buildings. For mining vehicles, electrical components offer a high availability and enable low-wear braking.

Concepts from the automotive sector show, by increasing the speed of electrical motors to over 20,000 min^{-1}, power density can be increased significantly (Chapter 3). Goal of the project "High Speed Electrical Drives" is to show the potential of high speed electrical drives for mobile machinery and to prove their suitability in laboratory tests and field tests. To transmit the high motor speed to the required wheel speed and to provide the required output torque, transmission gears are used. The high target speed of 20,000 min^{-1} requires the development of appropriate electrical machines, control units and gears.

The drives are to be used as wheel drives for a tractor and as a conveyor drive for a milling machine. A tractor from the Fendt Vario 700 series and a Wirtgen W 150 for the cold milling machine are chosen (Figure 1).

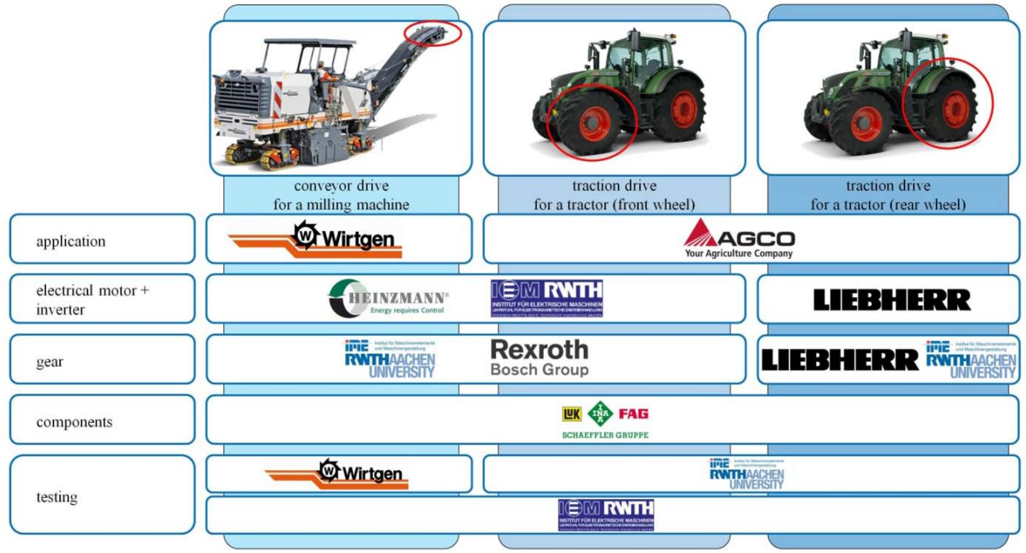


Figure 1: Project consortium "High Speed Electrical Drives" and vehicles.

The electrical motors and control units are developed by Heinzmann GmbH & Co. KG, in cooperation with the Institute of Electrical Machines RWTH Aachen University (conveyor drive and front wheel drive) and by Liebherr-Components Biberach GmbH (rear wheel drive). The development of the gearboxes is made by the Institute for Machine Elements and Machine Design of RWTH Aachen University, in cooperation with Bosch Rexroth AG (front wheel drive) and Liebherr-Components Biberach GmbH (rear wheel drive). For the design of the bearings and seals, Schaeffler Technologies GmbH & Co. KG is responsible. Wirtgen GmbH will test the conveyor drive in place. The wheel drives will be tested as modules on a test bench at the IME Test Center (Figure 1).

This paper only discusses the development of the front wheel drive for the tractor.

3 State of the Art

Currently, electrification is widely discussed in the construction and agricultural machinery sector. First concepts have already been developed. Examples of the agricultural machinery sector are the electrification of a beet harvester /4/, a research project on electrified single wheel drives for agricultural machines (Rigitrac) /5/ and the development of an electrical drive for a sprayer vehicle (Bonfiglioli) /6/. An example of the construction machinery sector is the mobile crusher Baitrack BP 130/85 B (Baioni) /7/.

For all applications, classic concepts for the electrical motors with low (beet harvester: 3,000 min^{-1} /4/; Rigitrac: 1,200 min^{-1} /5/) and moderate (Bonfiglioli: 8,000 min^{-1} /6/) rotational speed were chosen. On the one hand, these concepts usually provide better efficiency compared to existing hydraulic-mechanical concepts. On the other hand, properties regarding costs and installation space are less favorable. High speed electrical drives with motor speeds over n_{max} = 8,000 min^{-1} have not been considered in the construction and agricultural machinery sector yet. New motor concepts with a maximum revolution speed of 20,000 min^{-1} compensate these weaknesses and achieve power densities up to 5 kW/kg. This means, high end electrical permanent synchronous motors today realise power densities which up to now were unique strength of hydrostatic motors (Figure 2) /8/.

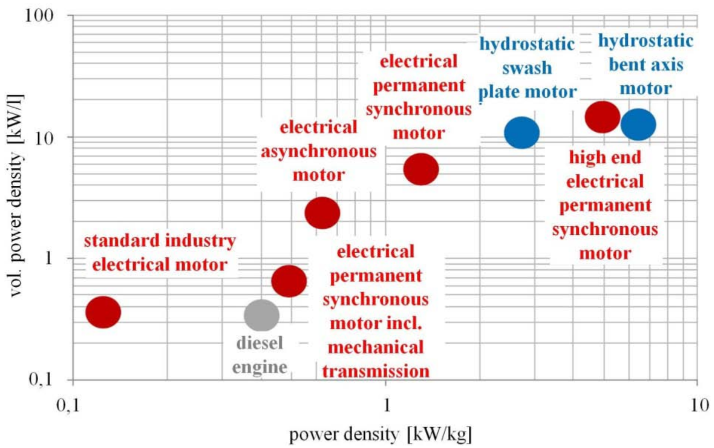


Figure 2: Comparison of power density for different motors /8/.

An example for the use of electrical high speed technology in the automotive sector is the electrical drive unit 2eDT100 of Getrag GmbH & Co. KG. This unit consists of a high speed electrical motor and a shiftable gear. For the motor, a permanent magnet synchronous machine with a power of P = 22 kW and a maximum speed of n_{max} = 22,500 min^{-1} was selected. The necessary transmission of the high motor speed to the required wheel speed of the vehicle is done by a shiftable helical gear with two gear stages. The first gear stage with a large gear ratio is used for start-up procedure as well as for driving at low speed and is thus particularly suitable for urban driving. At high speeds, the gear shifts automatically up to second gear stage. The weight of the drive unit is 48 kg, resulting in a power density of 0.46 kW/kg /9/. In comparison to a standard motor /10/, power density could be increased by the factor of five (Table 2).

An example for an even higher increase in power density is the electrical motor for a flywheel of rosseta Technik GmbH. For the motor, a permanent magnet synchronous machine with a power of P = 15 kW and a maximum speed of n_{max} = 35,000 min^{-1} was selected. The weight of the motor is 3 kg, resulting in a power density of 5 kW/kg. In comparison to a standard motor, the power density could be increased by the factor of 34 (Table 2).

However it has to be noted that the weight of the high speed motor does not include the housing, since it is integrated into the flywheel storage system /10/. Still, it is a remarkable increase in power density and shows the potential of electrical high speed technology.

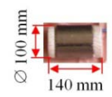
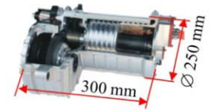
	Standard motor	High speed motor	High speed drive
Power	15 kW	15 kW	22 kW
Max. motor speed	3,000 min^{-1}	35,000 min^{-1}	22,500 min^{-1}
Motor diameter	300 mm	100 mm	250 mm
Motor length	480 mm	140 mm	300 mm
Motor volume	34 l	1 l	15 l
Total weight	140 kg	3 kg	48 kg
Power density	0.1 kW/kg	5 kW/kg	0.5 kW/kg
Picture			

Table 2: Comparison of standard motor, high speed motor of rosseta Technik GmbH /10/ and high speed drive of Getrag GmbH & Co. KG /9/.

4 Drive Concept

In this chapter, first the conception of the entire drive is discussed, followed by the description of the component development such as electrical motor (Chapter 4.1) and gear (Chapter 4.2), focusing on power losses. Finally, the individual losses of the components will be summarised to the overall efficiency of the drive unit (Chapter 4.3).

Basis for the gear and electrical motor design of the traction drive is a load spectrum of the vehicle under study. The requirements of the tractor, with a variety of different operating points, both, in the low and in the high speed range, represent a major challenge particularly for the electrical motor design (Chapter 4.1). In practical terms, the front wheel drive requires a maximum continuous output torque of 8,000 Nm at a vehicle speed of 4.2 km/h in the working range on one hand. On the other hand, a continuous torque of 1,000 Nm at a vehicle speed of 50 km/h is required for cruising (Figure 3).

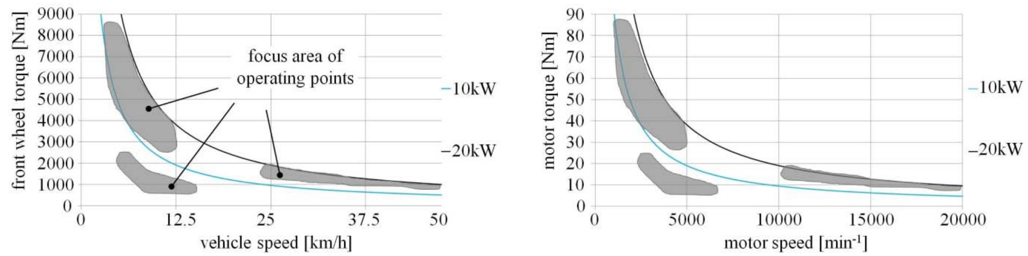


Figure 3: Load spectrum for the tractor's front wheel drive. Left: For front wheel. Right: For electrical motor.

The maximum required drive power for the front wheel drive of the tractor is P_{an}= 20 kW and for the rear wheel drive P_{an}= 41 kW. The maximum speed of the electrical motors are set to n_{max} = 20,000 min^{-1} for the front wheel drive, and to n_{max} = 14,000 min^{-1} for the rear wheel drive.

Due to the limited installation space for the drives and the high requested gear ratio of i > 95, planetary gears with three gear stages are used.

To find the best gear architecture for the drive concept, in a first step a variety of architecture combinations is generated by varying the operation mode of a planetary gear and by coupling these three planetary gears in different ways. Each planetary gear can be operated in four modes as shown in Figure 4.

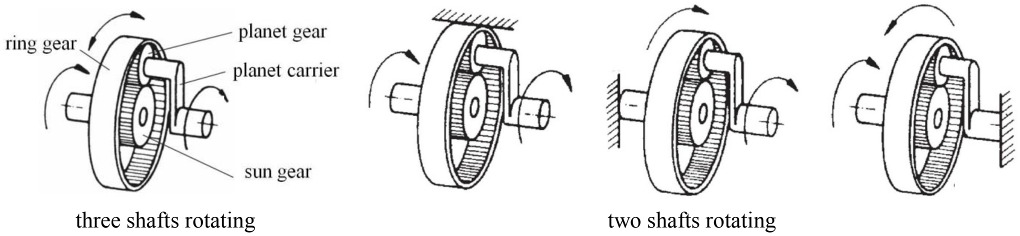


Figure 4: Operating modes of a planetary gear /11/.

Regardless of the operation mode, the three planetary gears can be coupled randomly as well. Possible coupling points are the sun gear, the ring gear and the planet carrier of each planetary gear. In the next step each architecture combination is analysed, using the torque scheme by RAMM /12/, and evaluated afterwards. With the torque scheme, the total gear ratio and the efficiency of all architectures can be determined and are used as decision-making criteria for the evaluation. In addition to these explicit criteria, an estimation regarding development risk is done and the opportunity to use existing gearboxes is taken into account. The evaluation leads to the reasonable solutions, shown in Table 3.

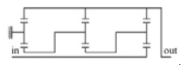
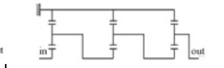
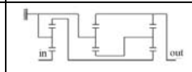
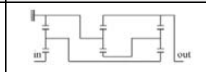
Criteria	Architecture a	Architecture b	Architecture c	Architecture d
				
Overall efficiency	96.2 %	96.4 %	96.4 %	96.2 %
Overall gear ratio	124	125	96	120
Development risk	high	high	low	medium
Use of existing gearbox possible	yes	no	yes	yes

Table 3: Comparison of gear architectures.

For the low speed gear stages, existing wheel hub gears shall be used. Architectures c and d from Table 3 are well suitable for the considered application, because they provide a convenient way of integrating high speed components into state-of-the-art wheel hub gears (Figure 5).

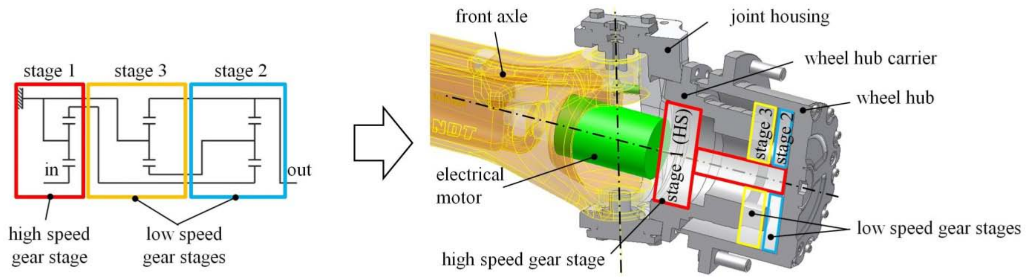


Figure 5: Concept for traction drive.

The wheel hub gear for the front wheel is provided by Bosch Rexroth AG and for the rear wheel by Liebherr-Components Biberach GmbH. The high speed gear stage is integrated into the carrier of the wheel hub, whereby it is physically separated from the low speed gear stages (Figure 5). This allows the lubrication and bearing concept being realised optimally and offers a poor development risk. In contrast to the injection lubricated high speed gears, the low speed gear stages are sump lubricated.

After a more detailed comparison of architectures c and d, which is discussed below, architecture c offers more advantages for high speed usage and is chosen to be realised (Table 4).

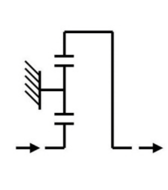
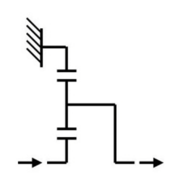
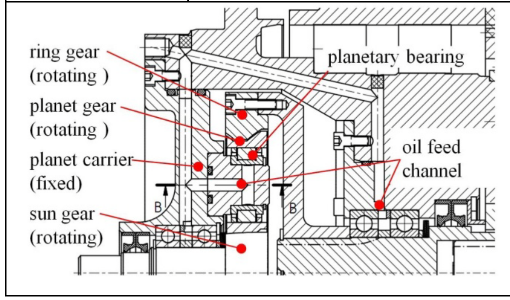
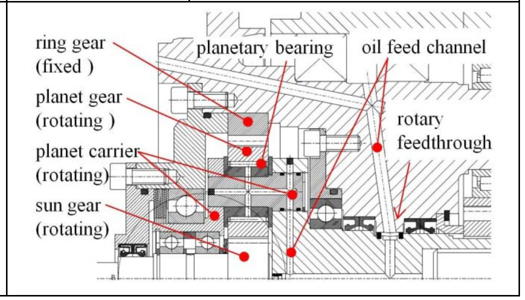
Output ring gear (high speed stage of structure c)		Output planet carrier (high speed stage of structure d)	
	- compact design difficult, because gear ratio is $i = i_{12}$;		+ compact design easier, because gear ratio $i = 1 - i_{12}$,
	+ easy lubrication system		- complex lubrication system
	+ no rotary feedthrough		- rotary feedthrough necessary
	+ no centrifugal load of planet bearing		- centrifugal load of planet bearing
			

Table 4: Comparison of variants for architecture of high speed gear stage.

Architectures c and d mainly differ in the operating mode of the high speed gear stage as shown in Table 4. Therefore only the high speed parts of the two gear architectures c and d are compared.

The operation mode of architecture d (input: sun gear; output: planet carrier; fixed ring gear) defines the gear ratio according to Equation (1) /11/.

$$i_d = 1 - i_{12} \tag{1}$$

For the operation mode of architecture c (input: sun gear; output: ring gear; fixed planet carrier) the gear ratio is defined to Equation (2) /11/.

$$i_c = i_{12} \tag{2}$$

The stationary gear ratio i_{12} is independent from the operation mode and defined according to Equation (3), where z_1 is the sun gear's number of teeth and z_2 the ring gear's number of teeth /11/.

$$i_{12} = \frac{z_2}{z_1}\,;\ (z_1 > 0,\ z_2 < 0) \tag{3}$$

Due to Equation (1), Equation (2) and Equation (3) the gear ratio i_c will always be smaller than gear ratio i_d. Therefore architecture d enables either a more compact design for a constant gear ratio or a bigger gear ratio for a constant installation space. Nevertheless, architecture c could be integrated into the available installation space offering the requested gear ratio.

4.1 Electrical Motor Efficiency

The electrical motor is as already mentioned designed and constructed by the Institute of Electrical Machines RWTH Aachen University. A particular challenge for the design of the electrical motor is the requirement of the front wheel drive, so that both a high achieved torque at low speed and a very high achieved maximal speed with less torque are necessary. The resulting problems are described in detail in this section. One of the problematic boundary condition is the limited space available for the front wheel drive, which requires a high power density of the electrical motor. Maximum diameter and maximum overall length inclusive housing and cooling jacket were predefined.

The basis for the design of the electrical motor is a choice of the most appropriate type of the electrical machine for the particular application. The results of first studies show that a permanent magnet synchronous machine (PMSM) with rare-earth magnets is most appropriate due to the high power density and high efficiency of that machine. Therefore, this machine is developed for the present project application. The approach for choosing an appropriate machine type for a given application is presented in /13/.

The main dimensions of an electrical machine have to be assessed. In general, the power of the electrical machine P_{mech} is proportional to the rotation speed n and the maximum continuous torque M of the machine $P_{mech} \sim n \cdot M$. The machine's volume is only proportional to the mechanical torque wherefore the mechanical power is proportional to the rotation speed and the volume of the machine $P_{mech} \sim n \cdot V$ /14/. Consequently, the power density can be improved by increasing the rotational speed (increase of power density) or by reducing the dimensions of the electrical machine (increase of torque density). For this reason it is useful to design a high speed electrical machine to reduce the torque demands and therefore the dimensions of the machine. After the definition of speed and torque, the design of the machine has to find the optimal machine dimensions for the required torque of the machine.

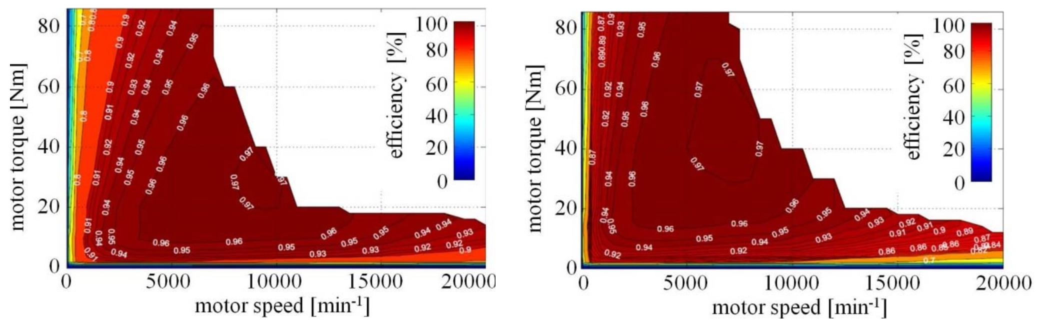
The dimensions of the electrical machine are significantly influenced by the number of pole pairs p. Basically, high torque motors are designed with a larger number of pole pairs, because this can reduce the outer diameter due to the inverse proportionality of the number of pole pairs to the yoke and teeth width of the electrical machine. A larger number of pole pairs ensures thinner yoke and teeth and therefore smaller outer machine dimensions.

In case of high speed applications a larger number of pole pairs can be crucial due to high iron losses. These losses can exceed the cooling capability of the machine and cause overheating of crucial machine parts such as the copper winding, so that the operational safety cannot be guaranteed. Iron losses depend on the flux density and strongly on the frequency of the rotating field. Highly utilised permanent magnet machines with rare earth magnets typically have high flux densities in the machines stator. At the same time the frequency of the rotating field is proportional to the number of pole pairs and the speed of the machine. Therefore, potentially high iron losses are possible for the machine in this project when operating at high speed and using a high number of pole pairs.

Furthermore, accurate iron loss estimation is especially important for the high speed PMSM. Classical iron loss models /15/ should only be applied for small flux densities and a limited frequency range. Therefore, an advanced method of iron loss estimation is used in this project which is especially developed for high frequencies and high flux densities /16/.

It should be mentioned that the number of pole pairs influences the end-winding length. The relationship between the end-winding length and the number of pole pairs is described in references /14/. A small number of pole pairs results in longer end windings and therefore increases the copper losses. Considering a bad cooling capability of the end windings, the small number of pole pairs can result an overheating of the end winding and isolation damage at high torque operating points.

The problems described above show that both, a small and a very large number of pole pairs are not suitable for the described application. This is because on the one hand the required volume should be as small as possible (a large number of pole pairs is advantageous) and on the other hand the machine should be operated at high speeds of 20,000 min^{-1} (a small number of pole pairs is advantageous to lower the frequency). In this project, two machine concepts with the number of pole pairs of $p = 2$ and $p = 3$ are compared concerning the resulting efficiency. This comparison can evaluate which concept is more efficient throughout the range of operating points. The efficiency maps of both concepts are shown in Figure 6.



$p = 2$ $p = 3$

Figure 6: Efficiency map of the electrical machines with number of pole pairs p = 2 and p = 3.

Figure 6 shows that the concept with the number of pole pairs $p = 2$ has a higher efficiency at higher speeds, while the concept with pole pairs $p = 3$ is advantageous at operating points with higher torque. The results of the calculation of the copper and iron losses and the efficiency are shown in Table 5.

	$p = 2$	$p = 3$
Operating point	85 Nm, 1,635 min^{-1}	
Copper losses	2.7 kW	1.2 kW
Iron losses	0.08 kW	0.1 kW
Efficiency	84 %	92 %
Operating point	9.8 Nm, 20,000 min^{-1}	
Copper losses	0.47 kW	0.67 kW
Iron losses	0.9 kW	2.1 kW
Efficiency	92 %	86 %

Table 5: Results of the calculation of the copper losses and iron losses and the efficiency.

Table 5 demonstrates the above described problem based on the example of two operating points from the load spectrum. The decisive factor here is which losses are more crucial for the thermal behaviour of the machine. Considering the thermal resistances and heat paths inside the electrical machine the conclusion is that high copper losses present a larger thermal problem than high iron losses due to the better capability to dissipate the losses from the iron surface of the machine. Based on these results a permanent-magnet synchronous machine with pole pairs $p = 3$ is preferred and designed for the applications of this project.

4.2 Gear Efficiency

The high motor speed requires specific technical solutions regarding gear teeth, lubrication, bearing technology and sealing technology to meet requirements regarding durability, acoustics, efficiency and thermal management. The following chapters focus on the efficiency of the components.

4.2.1 Gear Teeth

IME designs the gear teeth for the high speed gearbox in cooperation with Bosch Rexroth AG. The power losses due to teeth friction and paddling are calculated for each operating point with the simulation software AMESim. Teeth friction losses are load-dependent and compose of losses due to the sliding movement as well as losses due to the rolling movement of meshing teeth. Sliding losses are described by Coulomb's law as an integral over the path of contact. The friction coefficient is not constant /17/. Rolling losses depend on the lubricant film. The calculation of load-independent paddling losses bases on the theory of TEREKHOV /18/.

4.2.2 Bearing

The bearing design for the high speed gear is done by Schaeffler Technologies GmbH & Co. KG. After pre-selecting bearings for the high speed drive, a model of the drive is created in the analytical calculation software BEARINX. The software calculates bearing life according to ISO/TS 16281, considering typical influence factors such as axial and radial loads, tilting of the bearing rings, temperature, lubrication, internal clearance, internal bearing geometry and rib geometry. Bearing losses are determined, based on a friction calculation theory, using physical algorithms that have been validated by tests. These algorithms consider specific parameters such as paddling losses, losses in sliding contacts, losses in rolling contacts and losses in the load-free zone.

The goal is to create a robust and economical bearing concept for use in mobile machinery. Therefore, despite the high motor speed of n_{max} = 20,000 min^{-1}, high speed approved but expensive spindle bearings and bearings with ceramic balls are not used. To meet the requirement of the agricultural machinery sector for a maintenance-free design, grease lubricated bearings are not taken into consideration. The requested service life of 10 years for bearings cannot be provided without regreasing. Therefore, all bearings of electrical motor and gearboxes are oil lubricated.

For the motor shaft oil lubricated deep groove ball bearings with modified cage geometry are used. Due to the low friction, these are suitable for high speeds. The bearings withstand the poor radial loads caused by the rotor weight and the rotor imbalance. The sun gear (n_{max} = 20,000 min^{-1}) and ring gear (n_{max} = 4,380 min^{-1}) of the high speed gear are beared by angular contact ball bearings. This bearing type is suitable for high speeds as well and, in contrast to deep groove ball bearings, supports the significant axial forces of the helical gearing. For the planet gears (n_{max} = 11,200 min^{-1}) tapered roller bearings are used to support the high radial forces and the significant tilting torque of the planet gear wheels.

4.2.3 Seal

The drive concept has only one critical sealing location. It is located on the motor shaft and separates the oil lubricated area of the high speed gear from the electrical motor. For this purpose, a modified radial shaft seal is used, which is in contrast to the possible alternative labyrinth seal the economical solution. Due to the maximum motor speed of n_{max} = 20,000 min^{-1} and the design of the motor shaft, a peripheral speed of v_u = 36.7 m/s stresses the sealing lip. The limit for commercially available radial shaft seals is described as a maximum peripheral speed of v_u = 21 m/s. The modification of the sealing system includes an optimisation of the contact pressure force of the sealing lip as well as a tightening of the runout tolerances and surface quality of the shaft. The optimisation of the sealing lip's contact pressure force and the functional test of the sealing concept is provided in the context of the component tests. At the time of submitting the paper, these tests had not yet begun.

The values for the expected seal friction losses are obtained from the manufacturer. These data are only validated up to a peripheral speed of v_u = 25 m/s. For higher speeds, the values for losses are estimated by extrapolation.

4.2.4 Lubrication

The low speed gears are sump lubricated by ISO-VG 220 oil. For the high speed gear stages, ISO-VG 32 oil is an acceptable compromise on gear teeth strength and power losses caused by teeth friction. Additives improve temperature stability, foam characteristics and wear characteristics. Due to the high peripheral speed up to v_u = 31.2 m/s of the gear wheels, an injection lubrication is required /11/. To minimise paddling losses, the oil is extracted from the high speed gear. To keep the cooling system and lubrication system simple, the electrical motors are cooled with the oil of the high speed gear.

The injection lubrication generates additional power losses, which result from the pressure drop and the injected oil flow rate. These losses are low and therefore acceptable, due to a poor injection pressure of p_{ein} = 5 bar.

4.3 Efficiency of Drive Unit (Electrical Motor, Inverter and Gear)

Summing up all losses of the individual components, the overall efficiencies of the entire drive units are determined. Figure 7 shows the calculated overall full-load efficiency as well as the overall partial-load efficiency of the tractor's front wheel drive. The full-load graph was calculated for a driving power of P_{an} = 20 kW. The overall partial-load efficiency is based on a driving power of P_{an} = 6 kW.

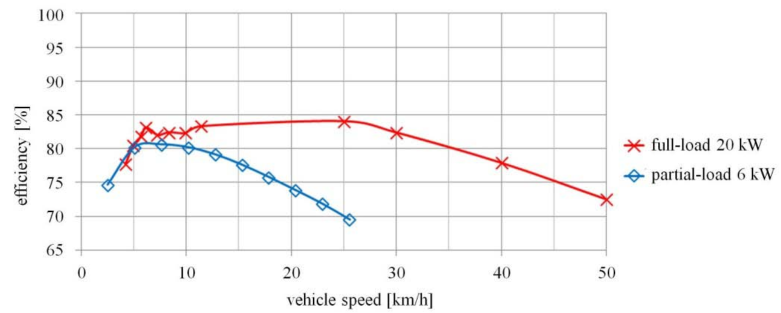


Figure 7: Calculated efficiency of the tractor's front wheel drive.

The efficiency calculation includes losses of the electrical motor (Chapter 4.1), the mechanical components (Chapter 4.2.1 to Chapter 4.2.3), the injection lubrication (Chapter 4.2.4) and the inverter. For the inverter, a constant efficiency of η = 97 % is assumed. Figure 8 presents the calculated power losses of the individual mechanical components of the high speed gear stage and of the electrical motor at full-load (P = 20 kW) and at partial-load (P = 6 kW).

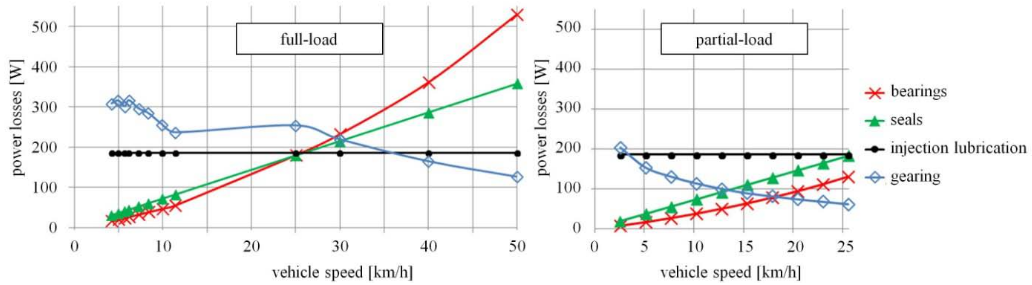


Figure 8: Calculated power losses of the high speed gear components for the tractor's front wheel drive.

The calculated efficiencies will be validated by laboratory tests at IEM (only electrical motors) and at the IME Test Center (entire drive unit).

5 Conclusion

Due to legal requirements to reduce CO_2 emissions and rising prices of fossil fuel, the project "TEAM" conducts research on energy efficiency improvements of mobile machinery.

The workgroup "High Speed Electrical Drives" develops and tests high speed electrical drives for a conveyor drive of a cold milling machine and for a traction drive of a tractor. This technology shall use the appreciated advantages of electrical drives. Power density, which is low for electrical drives when compared to hydraulic drives, shall be improved. Concepts from the automotive sector show, by increasing the speed of electrical motors to more than 20,000 min^{-1} that power density can be increased significantly.

New drive concepts for mobile machinery were developed and components were designed. With the calculated losses of the components, the expected overall efficiency of the drives was determined. Especially the partial-load efficiency is favourably. The high speed technology enabled the drives to be integrated into the limited space available due to a high volumetric power density. The concept can be applied to other mobile machinery applications.

In the upcoming project phase, the components will be tested. Then, the complete drive units, consisting of the electrical motors, the control units and the multi-stage gears, will be manufactured and tested on a test bench. The conveyor drive of the cold milling machine will be tested in the field as well.

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Nomenclature

Variable	*Description*	*Unit*
i	gear ratio	[-]
i_{12}	stationary gear ratio ($i_{12} < 0$)	[-]
i_c	gear ratio of concept c	[-]
i_d	gear ratio of concept d	[-]
M_{max}	maximum torque	[Nm]
n	rotation speed	[min^{-1}]
n_{max}	maximum rotation speed	[min^{-1}]
p	number of pole pairs	[-]
P	Power	[kW]
P_{an}	driving power	[kW]
p_{ein}	injection pressure	[bar]
P_{mech}	power of the electrical machine	[kW]
V	Volume	[l]
v_u	peripheral speed	[m/s]
z_1	number of teeth of sun gear	[-]
z_2	number of teeth of ring gear	[-]
η	Efficiency	[%]

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