Development of a High-Performance Magnetic Gear

Peter Omand Rasmussen, Torben Ole Andersen, Frank T. Jørgensen, and Orla Nielsen

***Abstract*—This paper presents calculation and measurement results of a high-performance permanent-magnetic gear. The analyzed permanent-magnetic gear has a gear ratio of 5.5 and is able to deliver 27 N·m. The analysis has shown that special attention needs to be paid to the system where the gear is to be installed because of a low natural torsion spring constant. The analyzed gear was also constructed in practice in order to validate the analysis and predict the efficiency. The measured torque from the magnetic gear was only 16 N·m reduced by the large end-effects. A systematic analysis of the loss components in the magnetic gear is also performed in order to figure out why the efficiency for the actual construction was only 81%. A large magnetic loss component originated in the bearings, where an unplanned extra bearing was necessary due to mechanical problems. Without the losses of magnetic origin in the bearings and less end-effects caused by relatively short stack, an impressive efficiency estimated at 96% can be obtained. Comparison with classical mechanical gears has shown that the magnetic gear has a better efficiency and a comparable torque per volume density. Finally, it is concluded that the results in this paper may help to initiate a shift from mechanical gears to magnetic gears.**

***Index Terms*—End-effects, finite-element analysis (FEA), gearbox, high torque density, magnetic gear.**

I. Introduction

PERMANENT magnets have fascinated and inspired many people though the ages because a permanent magnet produces a flux and magnetic force. It is exactly these characteristics that have made permanent magnets useful in many applications like lifting mechanisms, loud speakers, and couplings, not to mention electrical machines, which have been going though a renaissance the last two decades through the use of permanent magnets. This renaissance is most often seen in smaller machines where both efficiency and torque density have been improved significantly by the use of permanent magnets.

An area where permanent magnets have not garnered much attention is gearing purposes, that is, if a magnetic coupling is not considered gearing. Magnetic couplings can basically be considered as a magnetic gear with a gear ratio of 1 : 1. Magnetic couplings have a very high torque per volume density in a range of 300–400 kN·m/m^3 with high-energy permanent magnets, compared to standard electrical machines with only about 10 kN·m/m^3.

The basic idea of magnetic gearing can be tracked down to the beginning of the 20th century. An interesting example is a U.S. Patent Application filled in 1913 [1] describing (electro)magnetic gears consisting of two rotational shafts with salient steel poles. The two shafts were magnetically connected with stationary (electro)magnetic poles. Even though the gearing topology in the patent seems quite effective, apparently nothing was done to utilize the idea in commercial applications, and the topologies in the patent may have been forgotten.

Another interesting publication concerning magnetic gears is from 1940, where a U.S. Patent by Faus was filed [2]. The patent describes a magnetic gear based on two disks with different diameters and different numbers of permanent magnets, i.e., a magnetic gearing topology quite similar to mechanical spur gears with different numbers of teeth on the two disks. A variation of the worm gear with magnets is also described in the patent. In 1940, only ferrite magnets were available, in which the force per unit volume is only about one-tenth of that which can be expected from modern NdFeB magnets which originated in the 1980s. The proposed gearing topology was also weak in relation to the utilization of the permanent magnets because only one magnet on each disk contributed to the transmission of force. Both of the above disadvantages may have limited usage and research in the magnetic gearing area and to date less than 50 publications are known. In [3]–[6], the magnetic spur gear is considered. In [3], a two-dimensional (2–D) analytical calculation approach is developed and compared to finite-element analysis (FEA) with very good agreement. If the results from the hypothetical gear in [3] are extrapolated into a gear having a small air gap (0.5 mm) the torque per volume of the two disks is around 20 kN·m/m^3 with sintered NdFeB. This is not too impressive and explains why magnetic "spur" gearing is not widely used. In [4]–[6], parametric analysis is carried out with the help of FEA. Some finite-element results are also compared to experiments where excellent agreement is seen.

Magnetic worm and skew gears have also been analyzed in the literature [7], [8]. In [7], a practical worm gear with $SMCO_5$ magnets is presented. The gear has a gearing ratio of 33 : 1 and the maximum output torque is 11.5 N·m. The volume of the gear was not stated, but from the figures and main numbers presented in the paper the active volume is estimated to roughly 0.005 m^3, which gives a torque per volume of 2.3 kN·m/m^3, i.e., a magnetic gear with a very low torque per volume density. In [8], the practical gear in [7] is modified to a skew gear. The torque from the magnetic skew gear was five times less than the worm gear, however, the authors somehow arrived at the wild conclusion that the gear topology presented could be used in automotive transmissions.

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The papers referred to above all deal with magnetic gears having a poor utilization of the permanent magnets and, thus, very low torque per volume density. However, two papers [9], [10] seem to present magnetic gears with a much higher utilization of the permanent magnets with gearing topology similar to an epicyclic gearbox and also with some modifications to the topologies in [1]. In both papers, the torque is transmitted though a stationary segmented steel part having many poles with low magnetic reluctance. The idea with this part is to commutate the magnetic field from the high-speed side with few permanent magnetic poles into the low-speed side with many poles. Other than the basic elements in the gearing topology in the two papers not much else seems to be equivalent. In [9], the high-speed-side permanent rotor is also used for an integrated axial permanent-magnet motor and the torque transmission is a reluctance torque (PM to steel) where the torque transmission in [10] is the more traditional permanent-magnet alignment torque (PM to PM). The gear proposed in [9] seems to have numerous mechanical problems caused by the axial layout and the use of reluctance torque, which required a very small air gap (40 μm). In the practical prototype there was mechanical friction and, thus, a limited speed. The torque density from the developed prototype with both motor and gearbox was quite impressive. In the paper 9 N·m/kg were stated which corresponds to 72 kN·m/m^3 if the construction is assumed to be solid with a mass density of 8000 kg/m^3. In [10], a relatively large air gap (1 mm) could be used caused by the permanent-magnet alignment torque. The disadvantage with the gearbox proposed in [10], which actually is similar to a topology in [11], was that the gearbox was only theoretically investigated with emphasis on predicting torque per volume. In the paper, the torque per volume was calculated to be more than 100 kN·m/m^3 for the active materials. All the cited papers state more or less that magnetic gearboxes have a high efficiency caused by the elimination of frictional losses seen in traditional mechanical gearboxes, but neither measurements nor calculations were performed to validate this statement. Thus, missing in the literature is a practical example of a high-torque/density permanent-magnetic gearbox with a measured torque capability equal to what was theoretically calculated. Efficiency measurement is also an important parameter required in order to evaluate this new promising technology versus classical mechanical gearboxes. This paper will focus on providing the above missing aspects.

This paper is divided into five further sections, where Section II deals with a basic description and FEA of the permanent-magnetic gear with a high torque per volume density. Next, in Section III, the development and construction of a practical gearbox is described. In Section IV, the developed magnetic gearbox is being tested and compared to the theoretically predicted torque transmission capability. After this, a comparison to classical commercial gearboxes is made in Section V and, finally, a conclusion is given in Section VI.

II. Basic Description and FEA

The considered magnetic gearbox in this paper is shown in Fig. 1.

The magnetic gearbox has the same pole and stator segment structure like the gearbox theoretically described in [10]. This means that the high-speed rotor has $p_h = 4$ pole pairs, the low-speed rotor has $p_l = 22$, and the number of stator segments is $n_s = 26$. According to [10], the gearing ratio $N_{\rm gear}$ is given by

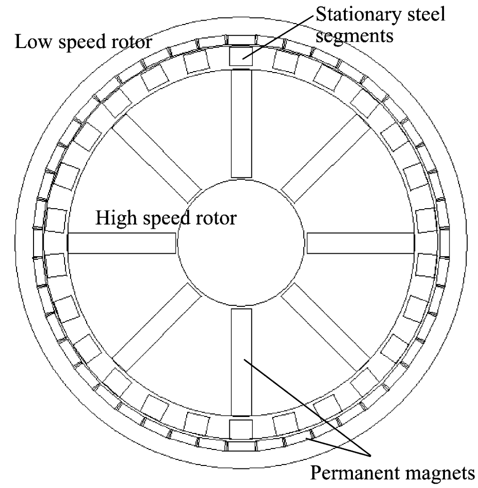

Fig. 1. Cross section of the considered magnetic gear.

$$N_{\rm gear} = \frac{P_h}{P_h - n_s} \quad (1)$$

which yields a gear ratio of $-5.5:1$.

The gearbox proposed herein is constructed in a more feasible manner for prototyping when compared to the one in [10]. The high-speed rotor is made as a spoke type instead of a surface-mounted type like the one in [10]. By doing so, standard rectangular magnets can be used and less consideration needs to be given to the centrifugal forces and glue for the magnets. Rectangular permanent magnets are also used for the low-speed rotor side instead of arc segments. In fact, the whole structure is developed by using many rectangular permanent magnets with the same geometry, whereby each spoke magnet on the low-speed rotor actually consist of eight magnets in the cross section.

With help of the free FEA program FEMM [12] the magnetic gear shown Fig. 1 is statically analyzed. The parameters for the gearbox are listed in Table I.

It has to be noted that the use of standard rectangular magnets results in a gearbox which is not optimized.

In Fig. 2, the flux lines and the induction are shown where the high- and low-speed rotors are placed in a position in relation to each other where the maximum stall torque is achieved. If the high-speed rotor is rotated 45° (1/8 revolution) while the low-speed rotor is kept fixed, then the gearbox is producing maximum negative stall torque. The variable torque is obtained by the relative position between the low- and high-speed rotor side like a classical magnetic coupling. Here, the torque has a quasi-sinusoidal relationship to the relative displacement between the two sides of the coupling. Worth noticing from Fig. 2 are the flux lines from the spoke magnets on the high-speed rotor side. Approximately 1/4 of the magnet is used for leakage near the shaft, which indicates both a poor utilization of the permanent magnet and also that large end-effects may exist in the construction.

TABLE I
PARAMETERS FOR THE ANALYZED MAGNETIC GEAR

Pole pairs on high speed rotor	4
Pole pairs on the low speed rotor	22
Number of stator segments	26
Outer radius of the low speed yoke [mm]	60
Inner radius of the low speed yoke [mm]	55.2
Outer radius of the stator segments [mm]	52
Inner radius of the stator segments [mm]	47
Outer radius of the high speed rotor [mm]	46.3
Radius of the high speed rotor shaft [mm]	17
Stack length [mm]	26
Standard rectangular permanent magnet size (HxWxL)	2.5x7x13
Remanence of the permanent magnets [T] (sintered NdFeB)	1.21
Coercivity of the permanent magnets [kA/m]	915
High and low speed rotor steel	Losil 400/50
Stator steel (Grain oriented)	M5-0.3 mm

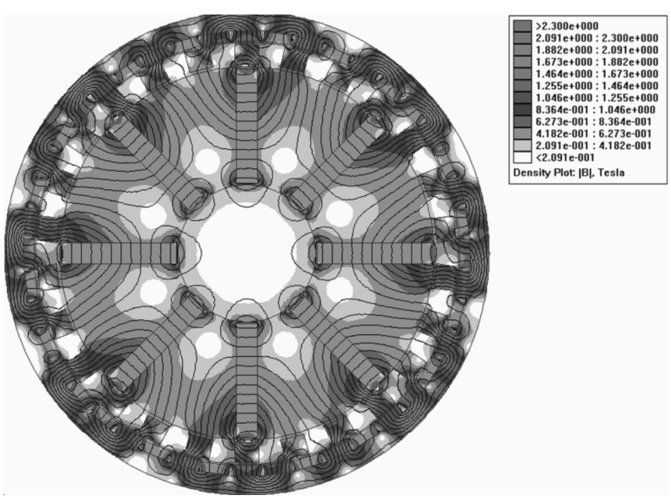

Fig. 2. Calculated flux lines and induction for magnetic gear.

In Fig. 3, the calculated torque versus relative position displacement is shown. The curve is obtained by evaluating Maxwell's stress tensor in the air gap on the low-speed side where the low-speed rotor is incrementally rotated while the high-speed side is kept fixed.

From Fig. 3 it is possible to see that the maximum stall torque is 27 N·m, which, with the use of the diameter and stack length in Table I, gives a torque per volume density of 92 kN·m/m^3. If the high-speed rotor was made with surface-mounted permanent magnets with the same volume as the spoke magnets, the stall torque is then calculated with the help of finite-element method (FEM) to be 32 N·m.

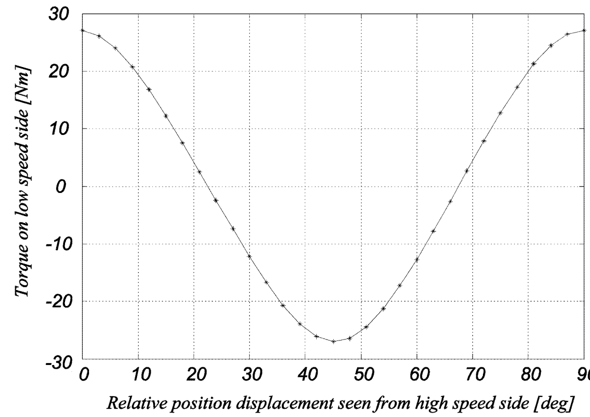

Fig. 3. Torque on low-speed side versus relative position displacement on the high-speed rotor side.

If a radial magnetic coupling were built with the same low-speed rotor and a matching inner rotor, the stall torque can be estimated to 117 N·m by the method given in [13]. This corresponds to a reduction of approximately four which is caused by the segments where approximately half of the available area is lost. The other half seems to be lost by the fact that not all segments and low-speed poles react with maximum torque and that there are double air gaps.

The curve in Fig. 3 can be interpreted as an internal torsion spring in the magnetic gear. In the relatively linear part from −20 to 20 N·m a displacement of 25° is seen, which transformed to the low-speed side will correspond to 0.08 rad. The torque difference of 40 N·m is, thus, giving a static torsional stiffness constant on 500 N·m/rad. For a rough comparison, a miniature double-disk flexible coupling with medium torsional stiffness and rated at 22 N·m has a static torsional stiffness constant of 2500 N·m/rad. Therefore, seen from the low-speed side the magnetic gear is about five times more flexible than a classical mechanical coupling and more concern needs to be given to vibrations and resonances in the system where the magnetic gear may be used. The torsion spring seen from the high-speed side is even worse because the static torsional stiffness constant has to be divided by the gearing ratio squared, which gives 16.5 N·m/rad.

III. DEVELOPMENT OF THE FIRST PROTOTYPE

The basic idea with the magnetic gear is to make a conservative design for what is believed to be the first prototype ever of this type of magnetic gearbox. The stack length is kept small in order to have fewer bearing problems with this double-sided construction and to limit the consumption of the permanent magnets. In Fig. 4, a photograph of the constructed magnetic gear is shown. The photograph shows the outer low-speed rotor together with the stator having the high-speed rotor inside.

From Fig. 4, it can be noticed that the stator segments are fixed by a nonmagnetic and dielectric nylon housing. The aluminum shaft of the low-speed rotor is mounted by adding eight holes to the outside of the low-speed rotor yoke. The high-speed rotor shaft is also made of aluminum. The magnetic field inside the shaft does not seem to vary much, so the eddy currents in the high-speed shaft should be inconsequential. Extra bolts and

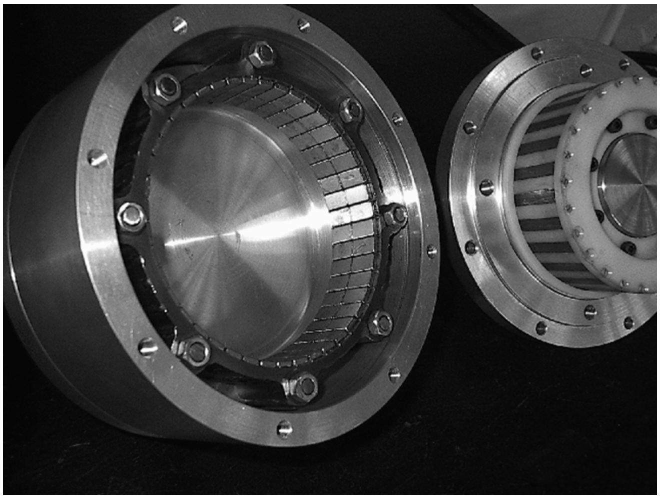

Fig. 4. Photograph of the constructed gear.

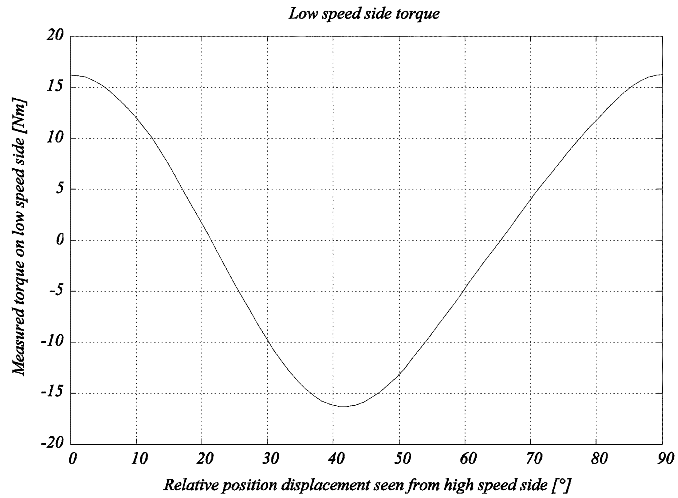

Fig. 5. Measured torque on low-speed side.

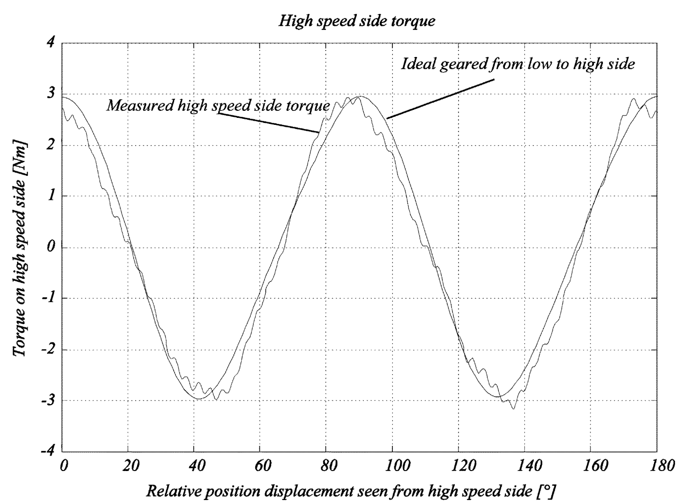

Fig. 6. Measured and ideal geared torque on high-speed side.

nuts are also used to secure the high-speed rotor stack. Further, it is worth noting in the photograph that the stack is made by doubling the amount of standard magnets.

The outer diameter of the whole construction is 160 mm and the total length is 90 mm, which yields a total torque per volume of only 15 kN·m/m^3. This number can, of course, be optimized due to the fact that both the layouts of the magnetic and the mechanical parts are conservative in this design. The stack length is very small which means that the end-shields play a dominant role in the total volume. The extra holes in the yoke on the low-speed side also play a significant role. If, for instance, the low-speed side of the gearbox was integrated into a wheel, the total volume would have been much lower.

IV. Test and Validation

In order to see the torque capability of the constructed magnetic gear, static torque measurement with the system described in [14] was done. The static torque-position curve was measured on both the high- and low-speed sides with the opposite side locked. The measured torque-position curve from the low-speed side can be seen in Fig. 5.

From Fig. 5, it is quite obvious that the torque calculated from the 2-D FEM model was not archived. Only 60% of the 2-D FEM calculated stall torque is measured. The reason for that may be caused by large end-effects, which, for the spoke-type magnets more or less also could be interpreted from the 2-D FEM results if the stack is short, which actually is the case. Also, the salient stator segments may participate with large end-effects. Large end-effects with salient poles are commonly known for switched reluctance machines.

The measured torque-position curve on the high-speed rotor side is shown in Fig. 6 together with measured low-speed-side torque-position curve divided by the gearing ratio of 5.5. The two curves are quite similar except that the measured torque-position curve on the high-speed side has a smaller ripple component. These ripple components harmonic order is 26 times higher than the fundamental component seen in Fig. 6 and is caused by the salient stator segments.

Measurement of accurate efficiency is quite difficult when a large efficiency is expected, whereby it is attempted to find a simple but reasonably accurate method to generate some indications. It is quickly realized that the losses in the gearbox are mechanical and iron losses, which in classical electrical machines are measured by a no-load test. At load, the iron losses may change a little, which is caused by another flux distribution in the magnetic components, where the major magnetic loss components are expected to be in the stator segments, in the low-speed rotor yoke and in the permanent magnets. Even though there may be some inaccuracies the choice was made to use the simple no-load test with a dc motor acting as an actuator and torque transducer utilizing the machine constant and the knowledge of dc motors no-load losses.

During the initial dynamic test of the magnetic gear, it was realized that there were some mechanical bearing problems and an extra bearing was installed in the open end between the high-speed rotor and the nylon housing having the stator segments. With the extra bearing installed a new static torque characterization was performed. The new measured static torque-position

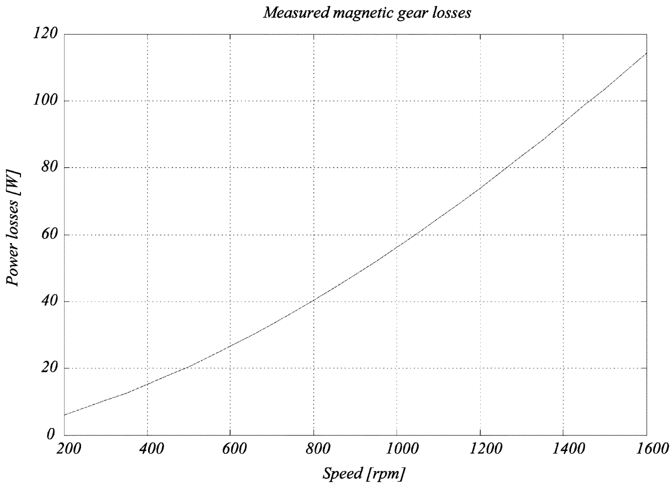

Fig. 7. Measured magnetic gear losses.

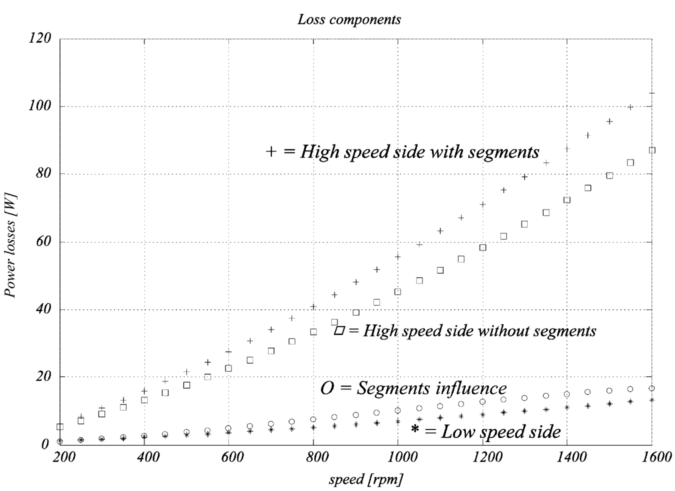

Fig. 8. Extraction of different loss components in the magnetic gear.

curves did not show any visible deviation from the curves in Figs. 5 and 6.

With the new bearing installed, the no-load losses for the gear were measured against speed (see Fig. 7).

At a nominal speed of 1500 r/min, the losses were predicted to be 104 W, which seems enormous due to the fact that only 1.9 kg is used in active materials, i.e., an indicated loss of more than 54 W/kg if the bearing losses were ignored.

Due to the huge amount of losses, the magnetic gear was more systematically analyzed in order to separate the loss components. The gear was separated into the two parts shown in Fig. 4 and the losses as a function of speed were measured on both sides. A test without the stator segments was also performed to identify the influences from these and "frictional" losses from the high-speed side. In Fig. 8, the results are shown for the three tests together with the difference between the two tests with and without stator segments.

From Fig. 8 is it quite clear that the high-speed side has extensive losses. When the segments are removed, the losses are 80 W at 1500 r/min, where the losses on the low side at the same speed are only 12 W. The high-speed side has three bearings while the low-speed side has only two, which are equal to two of the three

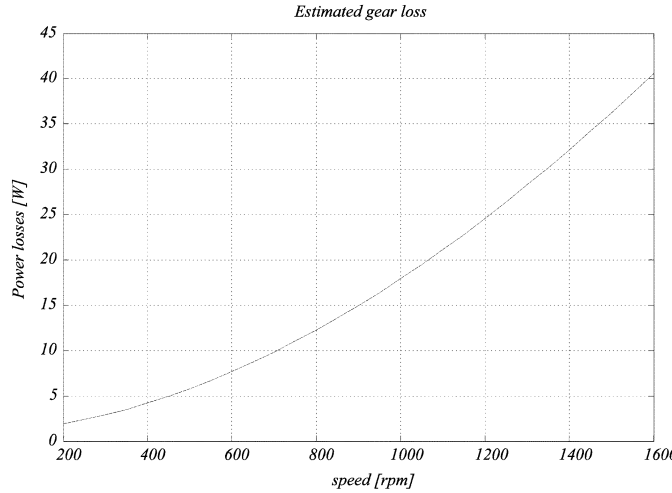

Fig. 9. Estimated magnetic gear losses without "magnetic" bearing losses.

bearings on the high-speed side. Therefore, if losses were purely mechanical bearing losses, the third bearing should have a mechanical loss of 68 W, which is impossible. The big difference seems to be caused by the end-fields from the spoke-magnet rotor which gives rise to extra friction and iron losses mainly in the extra installed bearing. The extra bearing was necessary in the construction because of mechanical problems. The influence of the stator segments also seems significantly higher than expected. The stator segments consist of 0.21 kg grain-oriented steel with 0.89 W/kg at 50 Hz and 1 T. Even though the fundamental frequency is 100 Hz at 1500 r/min, is it difficult to understand that the losses in the segments are 18 W. Some may be explained by the fact that the flux may cross the lamination plane and give higher eddy-current losses.

In order to have an idea of the magnetic gear losses without the unwanted "magnetic" bearing losses the measured losses on the high-speed side without the segments plus the low-speed side losses (real bearing losses measured on low side) can be subtracted from the total measured losses in Fig. 7. This gives the estimated result in Fig. 9.

The efficiency of the magnetic gear can be estimated as if it did not have the problems with the magnetic field in the bearings and was able to deliver the expected torque on 27 N·m without increasing the iron losses by a required lower leakage field. At 1500 r/min and with the help of Fig. 9, the estimated efficiency is calculated to be 96%.

From the experience gained developing the prototype, it is quite clear that more care must be taken if a new magnetic gear is to be constructed to render higher efficiency. A longer stack length is interesting in order to reduce the end-effects and also a surface-mounted high-speed rotor seems more preferable. If possible, the stator segments have to be stacked in the axial length with a stiffer material than nylon to secure them. More care has to given to the bearing layout in order to avoid couplings to the magnetic field. Bearing support in both ends for both the high- and low-speed rotor may solve some of the mechanical problems.

TABLE II
COMPARISION OF THE MAGNETIC GEAR WITH COMMERCIAL MECHANICAL GEARS. (a) 2-D FEM COMPUTATION. (b) MEASURED FROM THE CONSTRUCTED MAGNETIC GEAR. (c) ESTIMATED EFFICIENCY FOR A "PROBLEM-FREE" CONSTRUCTION

Manufacturer	EISELE	Stronglet	Zürrer	Rossi Motoriduttori	Magnetic gear active parts	Magnetic gear total assembly
Gear topology	Planet	Worm	Double-stage spur gear	Single-stage spur gear	Magnet with stator segments	Magnet with stator segments
Gearing ratio	5.35	5.25	5.65	5.23	5.5	5.5
Torque [Nm]	20	26	31	26.2	27[a] (16[b])	27[a] (16[b])
Efficiency [%]	85	90	90	96	96[c] (81[b])	96[c] (81[b])
Volume [cm^3]	581	-	-	4241	294	1810
Weight [kg]	-	4.5	8	21	1.9	7

V. COMPARISON WITH MECHANICAL GEARBOXES

The magnetic gear seems to have the following advantages when compared to classical mechanical gears:

- no mechanical fatigue;
- no lubrication;
- overload protected;
- no mechanical contact losses;
- no mechanical contact acoustic noise;
- potential for very high efficiency (only a little core loss and bearing loss);
- high torque per volume ratio (ten times standard motors).

Even though the general advantages above give the magnetic gear a plus, it is still interesting to compare the practical prototype with some standard gearboxes with similar major ratings in the form of torque and gearing ratio. In Table II four commercial mechanical gearboxes are compared to the prototype where the practical problems are more or less ignored. The comparison parameters are technical performance parameters in form of efficiency and volume (weight). Price also could have been an interesting parameter, but the magnetic gearing technology is too early in the development stage to give a good estimate.

From Table II, it is seen that the magnetic gear with only active parts is favorable when the efficiency and volume are compared to all the mechanical gears. Also, the total assembled magnetic gear has advantages even when the assembly is far from optimized. The efficiency is comparable to a single-stage spur gear and, yet, the assembly occupies less than half the volume. The estimated efficiency of the total assembled magnetic gear is much higher than the rest of the mechanical gears, but the volume seems higher. However, it is believed that an optimization may make comparable volumes. An aspect which is favorable for the magnetic gear is the compared torque rating. The stall torque is used as the comparison parameter, which, in practice, never will be the case because of safety. A safety factor of 70% will probably be a reasonable choice.

VI. CONCLUSION

A high torque per volume density magnetic gear was described and analyzed with the help of 2-D FEA. The calculations have shown that the magnetic gear with a gearing ratio of 5.5 and a stall torque of 27 N·m has a torque per volume density on 92 kN·m/m^3. In order to validate the 2-D FEA, a practical gear, which is believed to be the first of its kind, was constructed. Tests with the gear have shown that the stall torque is only 16 N·m. The reduction seems to be caused by large end-effects in the short practical construction. No-load tests were performed on the constructed magnetic gear in order to predict efficiency. For the actual construction the efficiency was only 81%. The low efficiency was mainly caused by losses of magnetic origin in an extra installed bearing, which was necessary due to mechanical problems. If the construction were improved with lower end-fields and no losses of magnetic origin in the bearings, the efficiency is expected to be 96%.

The magnetic gearbox was also compared to various classical mechanical gears. Here, it is seen that the magnetic gear has a significantly improved efficiency with a comparable or smaller volume than the classical gears.

In the future, more care must be given to the mechanical construction. Single-sided bearing supports may not be very advantageous if the gear box has a long stack. Further, it may be interesting to explore the gear with other gearing ratios and fully optimize its construction. However, indications of a possible shift from mechanical gears to magnetic gears seem even more realistic with the results of this paper.

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