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RADIAL PARTITIONED RELUCTANCE MACHINE AS GEAR PUMP – AN APPROXIMATION AMONG ELECTRICAL AND FLUID POWER

New operations often demand not only a further development of available technology with the target of increasing the efficiency but also in some case rethinking and breaking through of traditional structures are necessary, which were developed over the course of the technical development. The goal consists in adapting known active principles so, that they are suited for the technological process. The contribution describes the way to the arrangement of an electrical-hydrostatic gear pump of so far unrivaled integration degree. Originating in the particular physical connections common design guidelines are derived. From these criterions and limits for an integrated arrangement are derived. Furthermore fundamental problems arised from the integration are discussed like the consequences of the partition of the motor, the use of a common tooth form and the working out of a material with sufficient properties for both hands.

1. INTRODUCTION

Electrical engineering and hydraulic power: more oppositional two special fields can be hardly. Here invisible current flows through cables, there the prejudice exists of trickling lines and oil-polluted plants. Here closed-loop controlled drives move finest motions; there high pressures lift up loads with a power, from which electrical machine builders only can dream. The special fields are grown such various that also in the formula world an approximation experience difficulties. This way the pole pair number p is already assigned to the pressure, if in electrical engineering it is referred to the machine length l, so the hydraulic engineer refers to a tooth width with the same mind.

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However both fields are essential: because of their diverse physical principles are sequentially also the technical possibilities and with it the applications are different. A connection up to now is manufactured physically obvious by a shaft or by a gear unit. By this way it is reached an electrically driven pump or an integrated pump in a common housing. Both arrangements are state of the art, if fixed plants are regarded. The components keep functional decoupled in both cases as far as possible. A higher integration degree can only be achieved, if essential structural components of motor and pump are identically. Whatever all contrasts a conflation is enormously attractive, at further view also commonalities can be found. Thusly a likeness can be deduced already optically from Fig. 1. The rotors of both consists of gear wheels, in case of the gear pump they lift the fluid and seal up the pressure side to the sucking side, in case of the reluctance machine they effect a different conductance of the magnetic circuit, which is used for the generation of the rotary motion. Therewith the integration can be carried on up till the dissolution of the working unit by the electrical machine.

2. LINKED DIMENSIONING GUIDE LINES

For reluctance machines the predefinition of the fundamental dimensions, the rotor diameter $D$ and the ideal length $l$, are the starting point of the design. Afterwards the dimensioning of the inner components of the magnetic circuit happens. The fundamental form of the design equation is for the power and the torque.

$$M = \frac{P}{\omega} = \frac{C \cdot D^2 \cdot l}{2\pi}$$

(1)

The utilization factor $C$ of the machine is a measure for the electromagnetic stress respectively the utilization of the inner volume. The utilization factor is proportional to the product of the current coverage und the air gap induction. It is depending
among others on the size of the machine, its construction and the cooling. Typical
values for the utilization factor are in the range of 50 to 300 (..500) kNm/m³.

The air gap should be chosen always as small as possible. The air gap length is
chosen according to the rotor diameter and its relation to the active length. Values of
\( \delta = 0.25 \ldots 0.6 \text{ mm} \) are usually. At conventional reluctance machines relations are
assumed between the machine length \( l_i \) to the rotor diameter, the outer diameter to the
rotor diameter, the height of the rotor teeth and the relation of the yoke to the tooth
width.

For the gear pump it should be tried to describe the energy and material flow in
oversimplified form by a few basic equations. There the power \( P \) und the torque \( M \) are
to be calculated as follows by means of the pressure \( p \), the flow rate \( Q \) and the flow
volume \( V \), if the physical and hydraulic efficiency are neglected:

\[
P = p \cdot Q \quad \text{and} \quad M = \frac{V \cdot \Delta p}{2\pi}
\]

(2)

The module \( m \) is the scale factor of the tooth system referring to the pitch circle
diameter; therewith it is distinguished from the tooth division \( \tau \) which is usual in
electrical engineering by the ratio \( Pi \). It is linked with the tooth height by the factor 2.

Now the dependence of the flow volume of the pump from the addendum circle di-
ameter \( D_h \) can be found. As a consequence the flow volume is a function of the
tooth number. With it the necessary torque can be stated as a function of the geometry
and the pressure to produce

\[
M = \frac{p}{2\cdot \pi} 2 \cdot \pi \cdot \frac{1}{4} \cdot D_h^2 \cdot \left(1 - \left(1 - \frac{2}{\tau}\right)^2\right)
\]

(3)

In case of the integration of the gear pump in the electrical motor with the
same components also the particular systems of equations must be transferred into one
another. Viewing the equations of the utilization of the electric motor and the neces-
sary torque of the gear pump it is apparent the same dependence on the geometry.
There is followed a simple relation for the required utilization of the electric motor in
dependence of the pressure as starting point of the dimensioning.

\[
C = p \cdot 2 \cdot \tau \cdot \frac{1}{4} \left(1 - \left(1 - \frac{2}{\tau}\right)^2\right) = \frac{p}{2 \cdot \tau} \cdot \frac{1}{4} \left(1 - \left(1 - \frac{2 \cdot \tau}{\pi \cdot D}\right)^2\right)
\]

(4)

Utilization factor \( C \) und pressure \( p \) are coupled by a unit-less factor which is only
determined by the geometry. Obvious a higher pressure can be managed by a higher
number of teeth at the same torque that means at a lower flow volume. A required
pressure calls for a certain utilization proportionally. Only the tooth number remains
as an additional degree of freedom. However this can not be decreased because of
a minimum module respectively a minimum tooth division in the range of 5 mm to 20 mm. That means that a higher pressure can be achieved only by an increasing diameter. At the predefinition of a utilization factor the realizable pressure can be pictured in dependence of the tooth number in Fig. 3. or as a ration of the tooth division and the diameter in Fig. 4. graphically.

Fig. 2: Achievable pressure as a function of the tooth number

![Fig. 2](image2.png)

Fig. 3: Achievable pressure as a function of the diameter with $C = 250 \text{ kNm/m}^3$.

Pressures of more then 25 bar are only achieved by a very high tooth number, which is unrealistically from the point of the fabrication. Furthermore the geometric proportions must be kept especially the ration of tooth height to the air gap length for preservation of the utilization factor.

The power of the integrated pump can be increased by the variation of the main dimensions linearly with the length or square with the diameter. However this increasing of power is linked with the increase of the flow rate but not with an increase of the pressure.
3. FEATURES OF THE TOOTH SYSTEM

The teeth form the actual function elements of the electric-hydraulic pump. For the pump they have to lift oil and to seal up at the points of contacts and prevent the oil from the flow back. For the electric motor the teeth in interoperation with the stator teeth are the place of the origin of the torque by the force on the joint faces in the magnetic field.

The choice of the tooth number and the pole number of the stator and rotor determines essentially the properties of the machine. Priority numbers of the pump are in the range between 9 and 15 teeth. Typical reluctance machines have at the ratio of stator and rotor teeth \( z_s/z_r \) a 6/4-arrangement with a phase number of \( m = 3 \) for example. For the choice of the teeth number of the rotor two possible variations are permissible in dependence of the phase number \( m \), pole pair number \( p \) and the number of teeth per pole \( z_p \)

\[
z_s = 2 \cdot p \cdot m \cdot z_p \quad \text{and} \quad z_r = 2 \cdot p \cdot m \left( z_p \pm \frac{1}{m} \right).
\]

(1)

Fig. 4. Possible tooth number combinations for the reluctance machine at \( m = 3 \), highlighted: chosen arrangement

The diagram in Fig. 4. shows the possible tooth number combinations up to \( z_p = 4 \). A further increase of the tooth number per rotor is only limited possible, since the number of variations becomes less rapidly. Furthermore the possibilities for the subsequent separation of the stator to two halves of the gear pump are restricted.

Always stator and rotor can be interchanged in their arrangement, though an external-rotor motor is not possible here. The most usual arrangement with \( z_p = 1 \) is unfavourable at this application. By the construction of a high-pole motor it is possible to minimize the tooth leakage flux. By the utilization of the large pole space the stator winding can be put up for generating the required magnetomotive force (mmf).
In the regions of the magnetic circuit the remagnetization happens with several frequencies which can be deduced from the switching frequency \( f_s = n \cdot z_p \) which is directly proportional to the speed and the tooth number of the rotor \( z_r \). Therewith the stator frequency is set after the choice of the tooth number. The frequency affects deeply to the reactances and the core losses.
For the adaption of the tooth form of the pump, the profile displacement is one constructive degree of freedom by the pump. At a gear wheel with profile displacement another part of the evolvent is used compared with a gear wheel without profile displacement.

A negative profile displacement causes widened tooth tips at smaller tooth bottom width. The profile displacement can be applied to the adjustment of the centre distance of a gear pair; there it is applied to the approximation to an electromagnetically more favourable structure with thicker teeth. The observance of the two-side seal must be ensured.

The optimal tooth width of reluctance machines is always a compromise between oppositional effects. If the teeth are very thin the slot cross section would increase and the leakage flux would reduce. Thick teeth increase the maximal inductivity and allow a large magnetic flux. However, if the teeth are too thick, there would not be sufficient free zone between them at the unaligned position. Fundamentally it is proposed choosing the width of the stator and rotor teeth even. Beyond the tooth width also the tooth form has influence to the torque course and the torque maximum. On the whole the evolvent form of a gear wheel does not disagree with the trapezoidal form of the teeth of some reluctance machines.

The tooth form of the rotor is determined by the evolvent; for the optimization of the tooth form of the stator a number of transient FEM-calculations serve. The following pictures show one distribution of the flux density and the course of the torque as a function of the rotation angle. The torque maximum has its highest value at a stator tooth width of $b_z = 2.5$ mm. It corresponds with the tooth width of the rotor. This parameter is relatively insensitively; therefore a little bigger tooth tip width is to be preferred.

![Fig. 7. Distribution of the flux density](image)
4. RESULTS

The integrated gear pump was constructed like presented in the following pictures by the optimization of both special fields and the operation with magnetically and mechanically suited materials. Figure 9 shows the pictorial schematic with the fluid flow and the magnetic fields of the separated stators. Figure 10 shows the realized drive before the assembling.

The realized pump has the in the following table included data. An increase of the performance is problem-free possible. With the reached pressure the pump belongs to the range of low pressure applications. Despite of the integration this prototype approaches still the volume and the mass of comparable motor-/pump-systems. However the replacement of some mechanical components is advantageous like the connection
shaft, one housing and several sealing elements. In this contribution several advantages were not discussed like the possibility of a common cooling concept for the integrated drive.

![Fig. 10. Housing with the stators and the rotors](image)

There was verified with the investigations and the so far achieved results at the construction of the prototype, that a so much total integration is possible. Because of the too big differences at the power density just an application in the low pressure range was reached.

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<tr>
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<tbody>
<tr>
<td><strong>power</strong> P</td>
<td>1.1 kW</td>
<td><strong>pressure</strong> p</td>
<td>18 bar</td>
</tr>
<tr>
<td><strong>speed</strong> n</td>
<td>1000/min</td>
<td><strong>flow rate</strong> Q</td>
<td>37 l/min</td>
</tr>
<tr>
<td><strong>frequency</strong> f</td>
<td>667 Hz</td>
<td><strong>active length</strong> l_a</td>
<td>20 mm</td>
</tr>
<tr>
<td><strong>maximum torque</strong> M_{max}</td>
<td>14.0 Nm</td>
<td><strong>pitch circle diameter</strong> D_0</td>
<td>110 mm</td>
</tr>
<tr>
<td><strong>middle torque</strong> M_{bem}</td>
<td>10.6 Nm</td>
<td><strong>module</strong> m</td>
<td>2.75 mm</td>
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<tr>
<td><strong>utilization factor</strong> C</td>
<td>273 kNm/m³</td>
<td><strong>tooth number rotor</strong> z_r</td>
<td>40</td>
</tr>
</tbody>
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PROMIĘŚCIĘ PODZIELONA MASZYNA RELUKTANCYJNA JAKO POMPA ZĘBATKOWA POŁĄCZENIE ELEKTROTECHNIKI I HYDRAULIKI

Najwyższy stopień integracji w technice może być osiągnięty poprzez użycie elementów spełniających wspólne funkcje. W omawianym przykładzie dotyczy to wspólnego użycia zębów pompy zębatkowej i zębów silnika reluktancyjnego. W artykule tym opisany jest sposób projektowania pompy elektryczno-hydraulicznej z nieosiągniętym do tej pory stopniem integracji. Omówione zostały podstawowe problemy, do których należą m.in. projektowanie wymiarów pompy, użycie wspólnego kształtu zęba, dobór odpowiednich materiałów i konsekwencje podziału silnika. W artykule opisany został również prototyp pompy zębatkowej o mocy 1,1 kW, który znajduje zastosowanie w aplikacjach niskociśnieniowych. Ponadto, omówione zostały zalety i granice integracji funkcjonalnej napędu elektrycznego i pompy.