Theoretical and experimental analysis of the lubricating system of a high speed multiplier

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Abstract. Flywheel-based energy storage systems are used for energy storage in form of kinetic energy using a flywheel rotating at high speed. In order to achieve this high rotating speed a high speed multiplier could be used in order to increase the rotation speed of a conventional motor. This article presents a theoretical and experimental analysis of the lubricating system of a high speed multiplier used in a flywheel-based energy storage system. The necessary oil flow is theoretically computed using analytical formulas. The oil is used for lubricating the gears, the roller bearings and the sliding bearings. An experimental test rig is used to measure the oil flow. Finally the two results are compared.

1. Introduction

Flywheel-based energy storage systems are used for energy storage in form of kinetic energy using a flywheel rotating at high speed [1], [2]. This type of mechanical energy storage is used in Formula 1 cars in order to recover the kinetic energy of the cars during the braking process and to use it later in order to overtake the opponent. A small scale prototype of such a system was built in [3]. Different types of hybrid car systems based on flywheel energy storage were developed [4]-[7]. Flywheel energy storage is also used in space applications, as UPS in case of a power failure and as energy regulator in electrical grids [8], [9]. In order to accumulate maximum of energy at high speed, if the flywheel diameter is less than 50 cm it has to operate at speeds above 15000 rpm in a low friction environment like vacuum in order to minimize friction [10]. Composite materials are the first choice in order to withstand the high centrifugal forces [11]-[13]. Hydrodynamic bearings are also used in such systems [14]. As conventional motors turn at a rotation speed of 3000 to 5000 rpm, a high speed multiplier is interposed between the high speed flywheel and the electric motor. Different possibilities for oil jet lubrication for gears are presented in [15]. Also gear lubrication and different constructive factors have an influence on the efficiency of gear systems. The efficiency decreases with increasing speed [16]. The aim of the present paper is to analyze theoretically and experimentally the lubrication system of a high speed multiplier.

2. Experimental setup

The experimental test rig is presented in figure 1. The high speed multiplier is not put in rotation by a motor during the experimental tests. The first difference between the functioning system and the
stationary one is the oil viscosity. In the stationary system the oil viscosity is higher due to the lower oil temperature. Therefore the oil flow is lower. The second difference is that in order to observe the oil flow there are three roller bearings (8 in figure 3) missing on the top of the multiplier. Therefore there are three roller bearings on the test rig instead of six roller bearings in real functioning.

2. Lubricant supply system

![Experimental test rig](image1)

**Figure 1.** Experimental test rig.

The lubricant supply system of the high speed multiplier is presented in figure 2.

![Description of the lubricant supply system](image2)

**Figure 2.** Description of the lubricant supply system.
The operating pressure is 10 bars. The main limitation of the operating pressure is due to the hydraulic filter. The electric motor is operating at a rotation speed of 2800 rpm and has a nominal power of 375W. Due to the fact that the necessary computed oil flow is around 8 L/min, a hydraulic pump (7) with a displacement of 4.3 cm³/min and a flow up to 12 L/min is chosen in order to have also an oil flow reserve. The maximal working pressure of the hydraulic pump is 280 bars, far beyond the 10 bars needed working pressure.

![Figure 3](image-url)

**Figure 3.** The flywheel energy storage system: 1. High speed flywheel; 2. Entrance gear; 3. Satellites; 4. Output pinion; 5. High speed circumferential groove journal bearing; 6. Oil inlet; 7. Oil outlet; 8. Oil nozzle for roller bearing; 9. Lubrication system for roller element bearings; 10. Hydrostatic axial bearing; 11. Casing cover.
A jaw coupling (3) is used to compensate the small misalignment of the electric motor with the hydraulic pump. It also protects the hydraulic pump gears from high shocks when the electric motor is started. The supply pressure can be adjusted using the pressure relief valve (4). This pressure will be adjusted at 10 bars. Two manometers (5) are used, one before and one after the oil filter (9). If the pressure difference between the two pressures is more than 1.72 bars, the oil filter must be changed. The filtering element is a cellulose type one, having a maximum filtered particle size of 7 µm. It is important to keep a small particle size in the circuit in order to keep good functioning of the oil nozzles, gears (accuracy grade 4) and bearings. The thermometer (8) indicates the temperature of the oil, temperature which is extremely important in determining the oil viscosity. The on-off switch (2) is a motor protection circuit breaker which is used to manually start and stop the electrical motor, but also to protect the motor in case of overload.

A schematic of the high speed multiplier in the frame of a flywheel energy storage system is presented in figure 3. The high speed multiplier has a multiplication ratio of 11.75 which is realized in two steps, the first gear set having a multiplication ratio of 49/17 and the second a multiplication ratio of 53/13. Due to the high output rotation speed (18000 rpm) hydrodynamic bearings were used for the output gear train. The hydrodynamic bearings have also a good damping capacity. In order to reduce the dimensions of the high speed multiplier the power flow is split in 3 like in the case of a planetary gear set. Helical gears were used in order to have lower noise when functioning.

As lubricating additivated gear oil Mobilgear 600 XP150 was used, having following main properties:

- kinematic viscosity at 40 °C: 150 mm²/s
- kinematic viscosity at 100 °C: 14.7 mm²/s
- density at 15.6°C : 0.89 kg/L

The oil flow at the entrance gear set in the high speed multiplier is presented in figure 4. Only the first three nozzles from the first gear set are presented. The other three nozzles for the second gear set cannot be observed. Also the oil jets of the three roller bearings which are below could not be observed.

![Figure 4. Visualization of oil jets for the lubrication of gears (the multiplier is without the casing cover 11 from figure 3).](image-url)
The oil flow was measured using the scaled bucket. After three measurements we obtain an oil flow of around 7 L/min (mean value) at a functioning pressure of 10 bar. The measured oil temperature is 32°C.

3. Theoretical computations

The total oil flow is computed as following:

\[ Q_{\text{total}} = Q_{\text{hydrostatic}} + Q_{\text{hydrodynamic}} + Q_{\text{rotterb}} + Q_{\text{gearl}} \]  

(1)

where

- \( Q_{\text{hydrostatic}} \) – oil flow for the hydrostatic bearings
- \( Q_{\text{hydrodynamic}} \) – oil flow for the hydrodynamic bearings
- \( Q_{\text{rotterb}} \) – oil flow for the roller bearings
- \( Q_{\text{gearl}} \) – oil flow for the gear lubrication

3.1. Computation of the oil flow for the hydrostatic bearings

Due to the fact that the hydrostatic bearings are not externally loaded and the total gap is 0.5 mm, it is considered that the bearings are equally loaded due to the external lubrication pressure and the fluid film thickness is 0.25 mm for each bearing.

Using the conservation of flow (the flow that enters through the capillary restrictor is equal to the flow that leaves through the bearing gap) we obtain [17]:

\[ Q_{hs} = \frac{p_r \pi h^3}{6 \eta ln\left(\frac{r_a}{r_i}\right)} = \frac{p_s - p_r}{k_c \eta} \]  

(2)

where

\[ k_c = \frac{128 l}{\pi d_r^4} \]  

(3)

In order to minimize the total power of the hydrostatic bearings, following dimensions were computed: \( r_i=15 \text{ mm}, r_a=17 \text{ mm}, h=250 \ \mu\text{m}, p_s=10 \text{ bar}, l=4 \text{ mm}, d_r=0.5 \text{ mm}. \) With a restrictor length of 4mm and a diameter of 1mm a value of \( k_c=2.68 \times 10^{12} \text{ l/m}^3 \) is obtained.

The oil viscosity needs to be computed at the measured temperature of 32°C. For this it is considered that the dynamic viscosity varies with temperature using the Reynolds equation:

\[ \eta = \eta_1 \cdot e^{-\beta(T-T_1)} \]  

(4)

Knowing the dynamic viscosity of the oil Mobilgear 600 XP150 at 40°C \( \eta_{40}=0.133 \text{ Pa s} \) and at 100°C \( \eta_{100}=0.013 \text{ Pa s} \), the dynamic viscosity at 32°C is \( \eta_{32}=0.181 \text{ Pa s} \).

From equation 2 a recess pressure \( p_r=5.8 \times 10^3 \text{ Pa} \) is found. The flow for the hydrostatic bearing is then \( Q_{hs}=0.13 \text{ L/min} \). The total flow for the two hydrostatic bearings is \( Q_{\text{hydrostatic}}=2Q_{hs}=0.26 \text{ L/min} \).

3.2. Computation of the oil flow for the hydrodynamic bearings

The two hydrodynamic bearings are circumferential groove bearings. Due to the fact that there is no load on the bearings during the measurements, the shaft is considered to be concentric with the bushing and only the hydrostatic flow component will be computed [18]:

\[ Q_{\text{hydrodynamic}} = \frac{\pi p_s D^4 (B_1 + B_2) \Psi^3}{96 \eta B_1 B_2} \]  

(5)

The left bearing has following dimensions: \( B_1=B_2=4 \text{ mm}, D=15 \text{ mm}, \Psi=42 \ \mu\text{m}/15 \text{ mm}=0.0028, \eta_{32}=0.181 \text{ Pa s}, p_s=10 \text{ bar}. \) A value of \( Q_{\text{left}}=0.006 \text{ L/min} \) is obtained.

The right bearing has following dimensions: \( B_1=4 \text{ mm}, B_2=7 \text{ mm}, D=15 \text{ mm}, \Psi=72 \ \mu\text{m}/15\text{mm}=0.0048, \eta_{32}=0.181 \text{ Pa s}, p_s=10 \text{ bar}. \) A value of \( Q_{\text{right}}=0.024 \text{ L/min} \) is obtained.

The total flow for the two hydrodynamic bearings is \( Q_{\text{hydrodynamic}}=Q_{\text{left}}+Q_{\text{right}}=0.03 \text{ L/min} \).
3.3. Computation of the oil flow for the lubrication of the roller bearings

The roller bearings are lubricated using the lubrication nozzles presented in figure 3 (number 8). The flow through one lubricating nozzle is calculated using the relation:

\[ Q_{hs} = \frac{P_s}{K_c \eta} \]  

(6)

The necessary oil flow for lubricating the roller bearings can be estimated from [19]. It should be between 0.001 and 0.003 L/min. From equation 6 a diameter of 0.18mm for a length of 4mm of the oil nozzle is found. Due to the technical difficulties in executing these diameters, nozzle diameters of 0.5 mm and a length of 2 mm were machined using a lathe machine.

Hence a flow of 0.25 L/min was computed for each roller bearing. Due to the fact that there are only three roller bearings on the test rig instead of six in real functioning, the total roller bearing flow is \( Q_{\text{rollerb}} = 0.75 \) L/min.

3.4. Computation of the oil flow for the lubrication of the gears

In order to compute the oil flow necessary for the gear lubrication the dissipated power in the gears is necessary [20]:

\[ P_{Vzt} = P_{vz} + P_{vzo} - P_a \cdot K_a \left[ \frac{0.1}{z_1 \cos \beta} + \frac{0.03}{v_t + 2} \right] \]  

(7)

For the first gear train we have \( K_a = 1.2, z_1 = 17, \beta = 10^\circ, v_t = 8.13 \) m/s and \( P_{Vzt} = 332.4 \) W is obtained. For the second gear train we have \( K_a = 1.2, z_1 = 25, \beta = 10^\circ, v_t = 25 \) m/s. We obtain \( P_{Vzt} = 331.9 \) W.

Due to the fact that the dissipated heat by the outer surface of the multiplier is hard to compute it is estimated (like for journal bearings) that all the heat is absorbed by the cooling oil.

The necessary oil flow for cooling the gear multiplier is [20]:

\[ Q_{\text{gearl}} = \frac{P_{Vzt}}{c_{\text{oil}} \cdot \rho_{\text{oil}} \cdot \Delta T_{\text{oil}}} \]  

(8)

By replacing \( c_{\text{oil}} = 1900 \) Ws/(kgK), \( \rho_{\text{oil}} = 900 \) kg/m\(^3\) and \( \Delta T_{\text{oil}} = 5 \) K [20] a flow of \( Q_{\text{gearl}} = 4.65 \) L/min is obtained.

Figure 5. Measurement of pressure behind a gear nozzle.
Using equation 6, an entrance temperature of 40°C (when the multiplier is functioning) and a dynamic viscosity of 0.133 Pas and knowing the fact that there are 6 restrictors the following can be found: \( Q_{\text{restrictor}} = \frac{Q_{\text{gear}}}{6} \), the restrictors diameter is \( d_r = 0.67 \) mm and their length is \( l = 3 \) mm. The restrictors should be manufactured at \( d_r = 0.7 \) mm. The six restrictors were manufactured with a bore diameter of 0.9 mm in order to compensate the pressure loss in the supply channels of the multiplier. In order to show the pressure difference, a manometer was put behind a nozzle which is different from the entrance nozzle (figure 5). The measured pressure was 5 bar.

Due to the fact that the high speed multiplier is not loaded, the oil temperature is lower (32°C). The oil flow for the gears at this temperature can be computed using equation 6, a dynamic viscosity of \( \eta_{32} = 0.181 \) Pa s and a diameter of 0.7 mm, neglecting the pressure drop in the multiplier oil circuit. The new oil flow is \( Q_{\text{gear}} = 3.93 \) L/min.

3.5. Computation of the total flow for the high speed multiplier

Using equation 1 we find that the total flow necessary to lubricate the high speed multiplier is \( Q_{\text{total}} = Q_{\text{hidrostatic}} + Q_{\text{hydrodynamic}} + Q_{\text{rollerb}} + Q_{\text{gear}} = 0.26 \) L/min + 0.03 L/min + 0.75 L/min + 3.93 L/min = 4.97 L/min.

4. Comparison between theoretical and experimental results

The total flow determined experimentally is \( Q_{\text{total, experimental}} = 7 \) L/min and the theoretical determined oil flow is \( Q_{\text{total, theoretical}} = 4.97 \) L/min. The relative error can be computed using following formula:

\[
\text{error} = \frac{Q_{\text{total, experimental}} - Q_{\text{total, theoretical}}}{Q_{\text{total, experimental}}} \cdot 100
\]

Hence an error of 29% is obtained.

5. Conclusions

After theoretical and experimental determination of the oil flow, following conclusions are obtained:

- the difference between the theoretically determined diameters of the oil nozzles and the manufactured diameters is the main source of the difference between the theoretical and experimentally determined oil flow;
- there is also a pressure drop in the oil channels of the multiplier, which determines a lower entrance pressure in the oil nozzles;
- the hydraulic pump having a maximum flow of 12 L/min was correctly dimensioned, giving some oil flow reserve if necessary;
- the lubrication system, which was designed from scratch, proves to be reliable;

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Appendix

Following notations were used in the article:

- \( B_1 \) – bearing width at the left side of the circumferential groove
- \( B_2 \) – bearing width at the right side of the circumferential groove
- \( C \) – radial clearance
- \( c_{oil} \) – oil specific heat
- \( D \) – diameter of the journal bearing
- \( d_r \) – diameter of the restrictor
- \( h \) – fluid film thickness
- \( K_a \) – application factor
- \( k_c \) – capillary constant of the hydrostatic bearing restrictor
- \( l \) – length of the restrictor
- \( p_r \) – recess pressure of the hydrostatic bearing
- \( p_s \) – supply pressure
- \( P_vz \) – dissipated power in the gears through friction under load
\( P_{vz0} \) – dissipated power in the gears under free motion
\( P_{vzt} \) – total dissipated power
\( Q_{gearl} \) – oil flow for the gear lubrication
\( Q_{hs} \) – oil flow for one hydrostatic bearing
\( Q_{hs} \) – oil flow for the hydrodynamic bearings
\( Q_{hs} \) – oil flow for the hydrostatic bearings, 2 \( Q_{hs} \)
\( Q_{rollerb} \) – oil flow for the roller bearings
\( r_o \) – outer radius of the hydrostatic bearing
\( r_i \) – recess radius of the hydrostatic bearing
\( \nu \) – circumferential speed
\( \beta \) – helix angle of helical gear
\( \Delta T_{oil} \) – oil temperature difference between inlet and outlet
\( \eta \) – dynamic oil viscosity
\( \rho_{oil} \) – oil density
\( \Psi \) – dimensionless clearance, 2\( CD \)

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