THE INFLUENCE OF WATER AND MINERAL OIL ON MECHANICAL LOSSES IN A HYDRAULIC MOTOR FOR OFFSHORE AND MARINE APPLICATIONS

Paweł Śliwiński
Gdansk University of Technology, Poland

ABSTRACT

In this paper, mechanical losses in a hydraulic motor supplied with water and mineral oil (two liquids having significantly different viscosity and lubricating properties) are described and compared. The experimental tests were conducted using a special design (prototype) of a hydraulic satellite motor. The design of the satellite motor is presented. This motor was developed to supply both with water and mineral oil and features a non-circular tooth working mechanism. The paper also characterizes sources of mechanical losses in this motor. On this basis, a mathematical model of these losses has been developed and presented. The results of calculation of mechanical losses according to the model are compared with the experimental results. Experimental studies have shown that the mechanical losses in the motor supplied with water are 2.8 times greater than those in the motor supplied with oil. The work demonstrates that the mechanical losses in both the motor supplied with water and the one supplied with oil are described well by the mathematical model. It has been found that for the loaded motor working at high speed, the simulation results differ from experimental ones by no more than 3% for oil and 4% for water.

Keywords: mechanical losses,satellite motor,water,oil

INTRODUCTION

The hydraulic motor in a hydraulic system is the executive element [5–9,35]. Its purpose is to convert hydraulic energy into mechanical energy. The energy carrier in the hydraulic system is liquid. The type of liquid is conditioned by the requirements for these systems. Mineral oil is a commonly used liquid in hydraulic systems. However, in some industrial sectors, the liquid must be non-flammable (mining, steel mills, etc.) or non-toxic (food industry). Then, non-flammable synthetic liquids, water, or water-based liquids (i.e. HFA-E emulsion) are used [26,33]. The viscosity, density, and lubrication properties of these liquids differ [15]. These parameters have a significant impact on the size of mechanical, volumetric, and pressure losses in hydraulic machines [30–32,36]. These losses have an impact on the energy conversion efficiency in these machines [1,16,21,23,29,34,36]. Furthermore, the design parameters of hydraulic components have an influence on energy conversion efficiency [2,3,10–12,18–20,32,38].

There is a growing trend worldwide towards research and development of components and hydraulic systems supplied with water [6,13,14,28,32,33,37]. Therefore, a hydraulic motor dedicated to oil systems should not be used in systems where the working medium is water [27]. In general, each hydraulic device is dedicated to a specific type of working liquid.

Studies of hydraulic systems where water is the working medium are especially important in marine technology. In offshore technology and marine applications, hydraulic power

circuits are used frequently and water is generally available as a working liquid [13]. Thus, in order to eliminate pollution of the environment and marine waters, it seems reasonable and justified to eliminate oil systems and replace them with water systems. Water hydraulics could be used especially in small vessels (yachts and small cruise ships) in regions of the world where there is no frost. To this end, it is necessary to develop and test innovative components for hydraulic water systems [25,26,28,29,37]. Thus, it was reasonable to develop a new hydraulic motor dedicated to supply with water. This motor, called the satellite motor and marked with the SM symbol, contains an innovative operating mechanism consisting of non-circular gears [25,27,29–32]. The construction of this motor is described in the next section. Such satellite motors can be widely used in the shipbuilding industry. For instance, they can be used as rudder drives, anchor hoists, and drives of various deck winches and davits. Motors with small working volumes are ideal drives for capstans and capstan winches on yachts (Fig. 1).

Fig. 1. Hydraulic motor driven capstan (top) and capstan winch (bottom) [39]

The development of the SM motor has made it possible to investigate the influence of types of working liquids, which differ in extreme viscosity and lubricity properties, on energetic losses that occur in this motor. This article is limited to the description of mechanical losses. Consequently, this article has the following objectives:
a) to indicate and describe the sources of mechanical losses in the satellite motor;
b) to describe the mathematical model of the mechanical losses;
c) to compare the mechanical losses in motors supplied with mineral oil and water;
d) to compare the results of experimental research with the mathematical model.

SATELLITE MOTOR

A prototype satellite motor marked with the symbol SM-0,75/25, with a theoretical displacement of 32.94 cm³/rev, was selected for the experimental tests. The design of this

motor is presented in Fig. 2. The working mechanism of the satellite motor is a specific gear mechanism in which the rotor revolves around the shaft axis and the revolving motion is made by satellites which are in gear with the curvature and the rotor (Fig. 3).
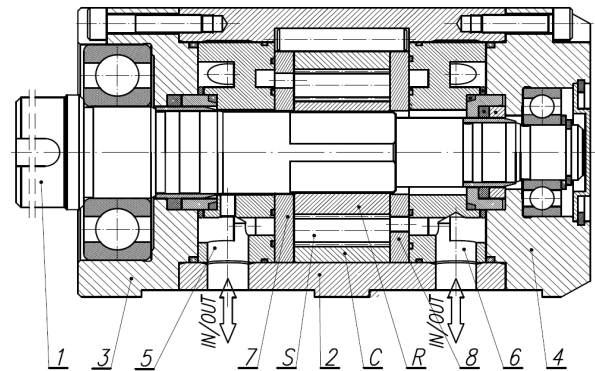

Fig. 2. General view and axial cross-section of SM-0,75/25 motor: 1 – shaft, 2 – casing, 3 – front casing, 4 – rear casing, 5 – inlet and outlet manifold, 7 and 8 – compensation (distribution) plates, C – curvature, S – satellite, R – rotor [25,27,30–32]
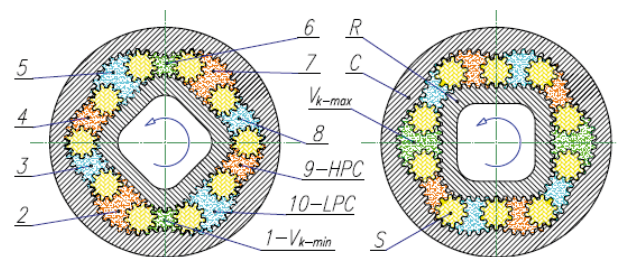

Fig. 3. The working mechanism of a satellite motor: C – curvature, S – satellite, R – rotor, 1–10 – working chambers, LPC and HPC – low-pressure and high-pressure working chamber, V_{k-min} and V_{k-max} – working chamber with minimum and maximum area (A_{min} and A_{max}) [25,27,30–32]

The curvature is toothed inside and consists of six humps. The rotor is toothed outside and consists of four humps. Gear wheels called satellites work with the curvature and the rotor. Spaces, called working chambers, are formed between the satellites, the curvature, and the rotor. The number of working chambers is equal to the number of satellites.

During rotation of the rotor, the working chambers:
a) change their volume from minimum V_{k-min} to maximum V_{k-max}, forming a high pressure chamber (HPC). The chamber is filled with water;
b) change their volume from maximum V_{k-max} to minimum V_{k-min}, forming a low pressure chamber (LPC). The chamber is emptied of liquid.
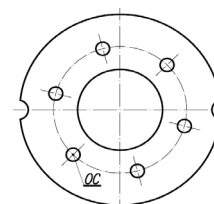

Fig. 4. Distribution plate: OC – inflow/outflow hole [25,27,30–32]

The working chambers in the satellite mechanism are closed by the compensation plates (Fig. 2 – elements 7 and 8, and Fig. 4), which also play the role of distribution plates.

The number of filling and emptying cycles of the working chambers per rotation of the shaft is the product of the number of humps of the rotor and the curvature. Thus, 24 cycles correspond to one shaft revolution.

Satellite motors have the smallest mass and overall dimensions as compared to other hydraulic motors. The power density of these motors exceeds 4 kW/kg.

SOURCES OF MECHANICAL LOSSES

Mechanical losses in a satellite motor result from [32]:

a) the rolling and sliding friction between the gear mechanism elements;
b) the compression of the liquid in the death volumes V_{k-min} of the satellite working mechanism (Fig. 3);
c) the compression of the liquid in the space between the teeth of the working mechanism elements;
d) the inertia of the working mechanism elements and the inertia of liquid in the working chambers;
e) the friction in the bearings and seals;
f) the viscous friction in the gaps of the working mechanism.

TORQUE OF LOSSES IN BEARINGS AND SEALS

The results of experimental studies of seals used in hydraulic machines show that the torque of friction in seals M_s mainly to the greatest extent depends on the pressure of the liquid contained in the seal chamber. The rotational speed of the shaft has a much smaller effect on M_s. In addition, M_s depends on the working fluid (lubricating properties) [28]. In a satellite motor, the rubbing speed of seals relative to the shaft neck exceeds 2 m/s. Additionally, the fluid pressure in the seal chamber is strictly dependent on the pressure in the motor low-pressure port (leaks from the working mechanism are discharged into the chamber of the shaft and further by a non-return valve to the outflow port). During the motor tests, the pressure in the outflow port reaches 2 MPa [28].

The results of seal tests have shown that the torque of friction in the motor seals (at a shaft speed range of up to 1500 rpm and a pressure p_2 of up to 2 MPa) does not exceed 1 Nm. However, for simplicity, the model assumes that the torque of friction in the seals is a function of the pressure in the outflow port of the motor and does not depend on the rotational speed.

Bearings in satellite machines have no contact with the working liquid. Hence, the torque of losses in bearings M_b does not depend on the liquid lubricant type. Furthermore, for simplicity, it is assumed that the friction in the bearings is independent from the speed. The motor working mechanism is hydrostatically balanced. Therefore, the load M of the motor has no influence on the torque of losses in the bearings.

Thus, the torque of losses in the bearings and seals can be described by Equation [32]:

$$M_{bs} = M_b + M_s = M_b + C_s \cdot p_2 \cong C_{bs} \cdot p_2 \qquad \textbf{(1)}$$

in which C_s $[\frac{Nm}{MPa}]$ and C_{bs} $[\frac{Nm}{MPa}]$ are constants.

TORQUE OF LOSSES DEPENDENT ON INERTIA OF SATELLITES AND INERTIA OF LIQUID IN WORKING CHAMBERS

During the operation of a satellite mechanism, the satellites are moving with a variable plane motion – there is a cyclic variation in speed of each satellite. Mechanical energy is supplied to the satellite during acceleration. Later, during deceleration, the energy is returned. The difference between the delivered energy and the returned energy is the energy consumed in the process of friction. Similarly, during the operation of a satellite mechanism, there is acceleration and deceleration of the liquid in the working chambers. Thus, the energy lost depends on the inertia of the liquid and the satellites. During the commutation change (e.g. at the time of opening of the inflow channel and cutting of the outflow channel by a satellite), inertial forces provide resistance to changes in the position of satellite teeth in the space between the rotor teeth and curvature. The inertia force of the satellite depends on its mass. Similarly, the inertia of the liquid depends on its mass in the working chamber.

Starting from the equations of energy conservation, it can be proved that the torque of losses M_{id} can be described by the simplified equation [32]:

$$M_{id} = C_{id} \cdot m^4 \cdot H \cdot n^2 \qquad \textbf{(2)}$$

where C_{id} $[\frac{Nm}{rpm^2 mm^5}]$ is a constant that depends on the type of satellite material and the type of liquid.

TORQUE OF LOSSES CAUSED BY COMPRESSION OF LIQUID IN DEATH VOLUMES AND IN SPACES BETWEEN TEETH

The process of compression of the liquid in the death volumes of a working mechanism occurs when the mechanism rotates and there is no flow from the death volume to either the outflow channel or the inflow channel. The increase in pressure in the working chamber occurs as follows: from the supply pressure p_{i-1} in the chamber as it increases its volume to the pressure p_{i-max} in the maximum volume V_{k-max}. Then the pressure drops to p_{i-2} in the working chamber, reducing its volume. The pressure p_{i-max} depends on the height of the satellite axial gaps and the rotor axial gap and also on the rotational speed of the rotor. At low speeds, the pressure p_{i-max} does not reach large values. It is estimated that as a result, the flow of liquid from the maximum dead chamber through the gaps to the low pressure chamber p_{i-max} is close to p_{i-1}. And at high speeds, the liquid does not have time to flow through the gaps to the low-pressure chamber. As a result, depending on the type of liquid, its viscosity and compressibility, the

pressure p_{i-max} is much greater than p_{i-1}. In the dead chamber V_{k-min}, there is a minimum pressure value p_{i-min}, similar to p_{i-2}. Thus, the value of p_{i-max} will have an impact on the torque of mechanical losses in the motor.

The flow rate of liquid Q_{dc} from the dead chamber through the gaps can be expressed by the formulas [32]:

$$Q_{dc} = \Delta V_{dc} \cdot \omega \tag{3}$$

$$Q_{dc} = C_{dc-1} \cdot \left(\Delta p_{dc}\right)^{\alpha} \tag{4}$$

where:
- ΔV_{dc} is the change in volume of the dead chamber during rotation of the shaft, which is approximately equal to the volume of liquid flowing from the space;
- ω is the angular velocity of the motor shaft;
- Δp_{dc} is the pressure increase in a closed space between teeth;
- C_{dc-1} $\left[\left(\frac{m^3}{s \cdot MPa}\right)^{\alpha}\right]$ is a constant that depends on the geometrical dimensions of the gap through which liquid flows.

From the power balance:

$$N_{dc} = Q_{dc} \cdot \Delta p_{dc} = M_{dc} \cdot \omega \tag{5}$$

it follows that:

$$M_{dc} = \left(\Delta V_{dc}\right)^{\frac{\alpha+1}{\alpha}} \cdot \left(\frac{\omega}{C_{dc-1}}\right)^{1/\alpha} \tag{6}$$

The value of ΔV_{dc} is difficult to calculate. ΔV_{dc} can be described by the simplified equation:

$$\Delta V_{dc} = C_{dc-2} \cdot m^2 \cdot H \tag{7}$$

where C_{dc-2} is the coefficient.

It is estimated that the changes of ΔV_{dc} are minor. Hence, we can assume that the stream Q_{dc} flowing through the gap is laminar, for which $\alpha = 1$. Thus, Equation (6) takes the form:

$$M_{dc} = \left(C_{dc-2} \cdot m^2 \cdot H\right)^2 \cdot \frac{\omega}{C_{dc-1}} \tag{8}$$

The above formula can be generalized to the form:

$$M_{dc} = C_{dc} \cdot m^4 \cdot H^2 \cdot n \tag{9}$$

where C_{dc} $\left[\frac{Nm}{mm^6 \cdot rpm}\right]$ is a constant.

By analogy, the torque of losses M_{ts} caused by compression of the liquid in the death volumes of the working mechanism can be described by the formula [32]:

$$M_{ts} = C_{ts} \cdot m^4 \cdot H^2 \cdot n \tag{10}$$

where C_{ts} $\left[\frac{Nm}{mm^6 \cdot rpm}\right]$ is a constant.

TORQUE OF LOSSES CAUSED BY VISCOUS FRICTION IN GAPS

During the rotation of the motor shaft, the walls of gaps of the working mechanism move relative to each other. This causes drift of the liquid layers in the gaps. Viscous friction in the liquid in the gaps can be expressed as the torque of losses M_v. This torque is proportional to the relative speed of the gap walls and proportional to the viscosity of liquid. Thus [32]:

$$M_v = C_v \cdot \mu \cdot n \tag{11}$$

where C_v $\left[\frac{Nm}{mPas \cdot rpm}\right]$ is a constant.

THE REPLACEMENT COMPONENT M_{DSV} OF THE TORQUE OF LOSSES

Taking into account that:
- the M_{dc} and M_{ts} components are impossible to determine, and
- the M_v component reaches very small values [1],
it is proposed to introduce a replacement component M_{dsv} of the torque of losses, expressed by the formula [32]:

$$M_{dsv} = M_{dc} + M_{ts} + M_v = C_{dsv} \cdot m^4 \cdot H^2 \cdot n \tag{12}$$

where C_{dsv} $\left[\frac{Nm}{mm^6 \cdot rpm}\right]$ is a constant.

TORQUE OF LOSSES M_{LM} CAUSED BY MIXED FRICTION

In the satellite working mechanism, satellites can move freely in the backlash (satellites are not mounted on axles). Therefore, during the operation of the mechanism, there is a slip in the area of interaction of the teeth. Thus, a non-linear increase in torque M_{lm} is a result of the sliding friction and the rolling friction of teeth.

Dietrich [4] recommends adopting the coefficient of friction in the gear teeth, in accordance with ISO/DIS6336/IV, as:

$$f = 0{,}12 \cdot \left(\frac{R_a}{\rho}\right)^{0{,}25} \cdot \left(\frac{F}{b \cdot \mu \cdot v}\right)^{0{,}25} \tag{13}$$

where:
- F [N] is the circumferential force on the rolling circle:

$$F = 2 \cdot \frac{M}{D_p} \tag{14}$$

- b [mm] is the width of the gear wheel;
- D_p [m] is the gear pitch diameter;
- μ [mPas] is the dynamic viscosity of the liquid;
- v [m/s] is the total speed of the cooperating wheels;
- ρ [mm] is the replacement radius of the tooth profile at the pitch point;
- R_a [μm] is the mean surface roughness of the side gear of interacting teeth.

Hence, in a helical gear, the torque of losses M_{lM} is the function:

$$M_{lm} = C_{lm}'' \cdot R_a^{0,25} \cdot \left(\mu \cdot v \cdot \rho \cdot b \cdot D_p\right)^{-0,25} \cdot M^{1,25} \quad \textbf{(15)}$$

where C_{lm}'' is the coefficient. It should be noted that the slippage within the cooperation of the teeth in the helical gear unit is very small.

According to Niemann [17], the mean friction coefficient can be described by the equation:

$$f = 0,048 \cdot \left(\frac{F}{b \cdot v \cdot \rho}\right)^{0,2} \cdot \mu^{-0,05} \cdot R_a^{0,25} \cdot \left(\frac{F}{b}\right)^{-0,065} \quad \textbf{(16)}$$

In the above equation, there is no compatibility of the units of measurement. Thus, it is supposed that the above equation is purely hypothetical (or empirical).

In satellite machines:
- the radius ρ of curvature of the teeth is dependent upon the modulus m of the teeth;
- the width b of the teeth is the same as tantamount to the height H of the satellite mechanism;
- the pitch diameter D_p is a function of the teeth modulus m;
- the total speed v of cooperating wheels depends on the rotor (or shaft) rotational speed and the size of the mechanism (and the size of the teeth modulus m).

Taking into account the above-mentioned dependence, it is proposed to describe the torque of losses M_{lm} in a satellite mechanism by the empirical relationship [32]:

$$M_{lm} = C_{lm} \cdot \left(\frac{R_a}{\mu \cdot H \cdot m^3}\right)^{0,25} \cdot n^{-\beta} \cdot M^{\alpha} \quad \textbf{(17)}$$

where C_{lm} is the factor.

MATHEMATICAL MODEL OF TORQUE OF LOSSES

KNOWN METHODS OF DESCRIPTION OF MECHANICAL LOSSES IN HYDRAULIC MOTOR

In the literature, mechanical losses in a motor are described as:
a) a function of the pressure drop in the motor and the rotational speed of the shaft $M_l = f(\Delta p,n)$ [1]. In this case, it

is assumed that the pressure drops in the internal channels are a component of mechanical losses. A description of mechanical losses in the form of $M_l = f(\Delta p,n)$ is inaccurate. Nonetheless, it is clear that the pressure drop Δp in the motor is affected by the torque losses M_l;
b) a function of the motor load and speed $M_l = f(M,n)$ [16,21,22,24]. This model does not refer to a source of mechanical losses in the motor. It is a simplified linear model. The influence of liquid viscosity ν on the mechanical losses depends on the factor $\left(\frac{v}{v_n}\right)^{\alpha}$, where v_n is the reference viscosity. Similarly, the $\left(\frac{v}{v_n}\right)$ impact of the rotational speed n of the motor shaft on the mechanical losses refers to the so-called theoretical (nominal) speed n_t. Moreover, mechanical losses in an unloaded motor are expressed as a function of theoretical torque. This torque corresponds to the nominal pressure drop in the hydraulic system in which the motor is used. It has also been assumed that mechanical losses in the loaded motor are directly proportional to the motor load. Besides, this model does not describe mechanical losses during the motor start.

The results of the experimental research carried out by the author indicate that the torque corresponding to mechanical losses M_l in a satellite motor is a non-linear function of both the load M and the speed n. Therefore, it was desirable to develop a new mathematical model that would contain all the above-mentioned factors.

The parameters, independent of the motor, are the load M on the motor shaft and the flow rate Q of the liquid feeding the motor. Thus, it would be desirable to describe the torque of mechanical losses in the motor M_l as a function of M and Q. Describing the torque of losses M_l as a function of the load M is possible and correct. However, describing the torque of losses M_l in the motor as a function of the flow Q is extremely complicated. It is clear that it is the relative speed of the rubbing surface that influences the friction force value. It would be possible to describe $M_l = f(Q)$ only in the case of a motor without volumetric losses.

The new method described below is based on an analysis of sources of mechanical losses. The described mathematical model will provide a more accurate simulation of the characteristics of mechanical losses in a hydraulic motor.

TORQUE OF LOSSES IN UNLOADED MOTOR

In an unloaded motor, the torque of losses $M_{l\,(M=0)}$ is calculated from the relationship:

$$M_{l\,(M=0)} = \frac{q \cdot \Delta p_i}{2 \cdot \pi} \quad \textbf{(18)}$$

Furthermore, $M_{l\,(M=0)}$ is the sum (Fig. 5) [32]:

$$M_{l\,(M=0)} = M_{bs} + M_{id} + \underbrace{M_{dc} + M_{ts} + M_v}_{M_{dsv}} + \Delta M_{lm\,(M=0\,lub\,\Delta p=0)} \quad \textbf{(19)}$$

where:
- M_{bs} is the torque of losses in the bearings and seals;
- M_{id} is the torque of losses dependent on the inertia of the working mechanism elements and the inertia of liquid in the working chambers;
- M_{dc} is the torque of losses caused by compression of liquid in death volumes;
- M_{ts} is the torque of losses caused by compression of liquid in the spaces between the teeth;
- M_v is the torque of losses caused by viscous friction in the gaps of the working mechanism;
- $\Delta M_{lm\,(M=0)}$ is the torque of losses at low speed, that is, in the range of speed from close to 0 (n = 0$^+$) to a certain boundary speed n'. This component reaches a maximum value $\Delta M_{lm\,(n=0,\,M=0)}$ for a speed close to zero (n = 0$^+$).

It is proposed to simplify the problem and the $\Delta M_{lm\,(M=0)}$ component to describe a linear function of rotational speed n as:

$$\Delta M_{lm\,(M=0)} = \Delta M_{lm-0\,(M=0)} - \underbrace{C^1_{\Delta lm\,(M=0)}}_{c_{\Delta lm\,(M=0)}} \cdot \mu \cdot n \quad (20)$$

where $C_{lm\,(M=0)}$ is a constant calculated from the equation of the trend line of characteristics $\Delta M_{lm\,(M=0)} = f(n)$.
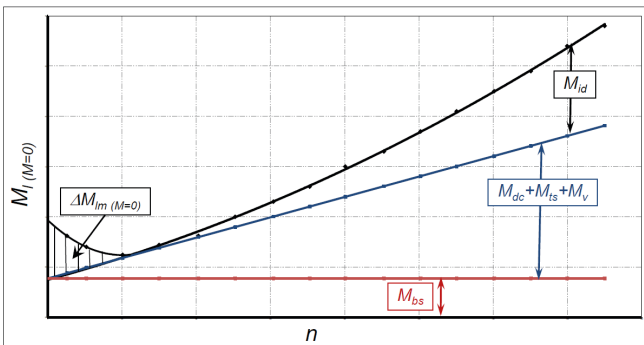

Fig. 5. The components of torque of losses in an unloaded motor [32]

After substituting (1), (2), (12), and (19) into (18), the following expression is obtained [32]:

$$M_{l\,(M=0)} = C_{id} \cdot m^4 \cdot H \cdot n^2 +$$
$$+ \left(C_{dsv} \cdot m^4 \cdot H^2 - \underbrace{C^1_{\Delta lm\,(M=0)} \cdot \mu}_{c_{\Delta lm\,(M=0)}} \right) \cdot n \quad (21)$$
$$+ C_{bs} \cdot p_2 + \Delta M_{lm\,(n=0,M=0)}$$

Hence, the equation describing the torque of losses in the unloaded motor is a square equation.

TORQUE OF LOSSES IN LOADED MOTOR

The torque of mechanical losses M_l in a loaded motor is the sum [32]:

$$M_l = \underbrace{M_{bs} + M_{id} + M_{dsv} + \Delta M_{lm\,(M=0)}}_{M_{l\,(M=0)}} \\ + \underbrace{\Delta M_{lm} + M_{lm}}_{M_{mf}} \quad (22)$$

where ΔM_{lm} is the torque of losses in a loaded motor occurring at low speeds (from n = 0 to a certain speed n = n") and reaching the maximum value $\Delta M_{lm(n=0)}$ for a speed n close to 0; that is, n = 0$^+$ (Fig. 6). ΔM_{lm} is a function of load M and motor speed n. That is, for n < n", M_{lm} should be described by the equation [32]:

$$M_{lm} = M_{lm(n=0)} - C_{lm\,(n")} \cdot \mu \cdot n \quad (23)$$

where M_{lm-0} is the torque of losses for n = 0 and $C_{lm(n")}$ is the factor.

ΔM_{lm} can be described by the simplified function [32]:

$$\Delta M_{lm} = \underbrace{C_{lm-0} \cdot M}_{\Delta M_{lm\,(n=0)}} - C_{\Delta lm} \cdot \mu \cdot n \quad (24)$$

where $C_{\Delta lm}$ and C_{lm-0} are coefficients.

In order to determine ΔM_{lm}, it should be assumed that for n < n", M_{lm} is a linear function (23). This causes an inconvenience in that:
- the speed limit n" should be found;
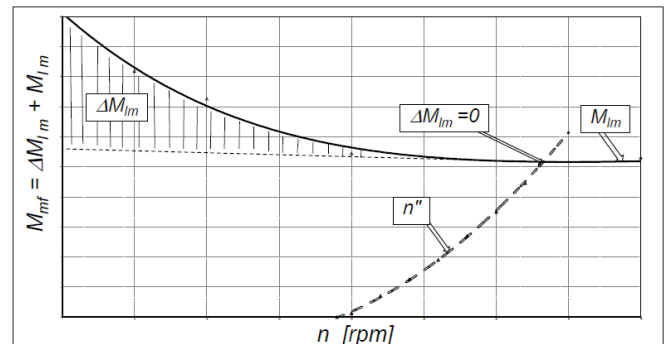- M_{lm} is described by Equation (16) for n > n" and by Equation (22) for n < n".

Fig. 6. Characteristic of M_{mf} component in loaded motor.
The component $\Delta M_{lm} = 0$ for n = n" [32]

The mathematical analysis of formula (16) indicates that $M_{lm\,(n<n")} \gg M_{lm\,(n>n")}$. Therefore, it is proposed to simplify the model and abandon the separate determination of ΔM_{lm}. Then [32]:

$$M_{mf} = \Delta M_{lm} + M_{lm} \approx M_{lm}$$
$$= C_{lm} \cdot \left(\frac{R_a}{\mu \cdot H \cdot m^3} \right)^{0,25} \cdot n^{-\beta} \cdot M^\alpha \quad (25)$$

The torque of losses M_l in a motor, expressed by the formula (20), in the expanded form is as follows [32]:

$$M_l = \underbrace{\Delta M_{lm-0\,(M=0)} - C_{\Delta lm\,(M=0)} \cdot n}_{\text{in the range of } n=(0^+;n')} + C_{bs} \cdot p_2$$
$$+ C_{id} \cdot m^4 \cdot H \cdot n^2 + C_{dsv} \cdot m^4 \cdot H^2 \cdot n$$
$$+ C_{lm} \cdot \left(\frac{R_a}{\mu \cdot H \cdot m^3}\right)^{0.25} \cdot n^{-\beta} \cdot M^{\alpha} \qquad (26)$$

THE TEST STAND AND MEASURING APPARATUS

The satellite motor was tested on a test stand with power recuperation. A diagram of the measurement system of this test stand is shown in Fig. 7.
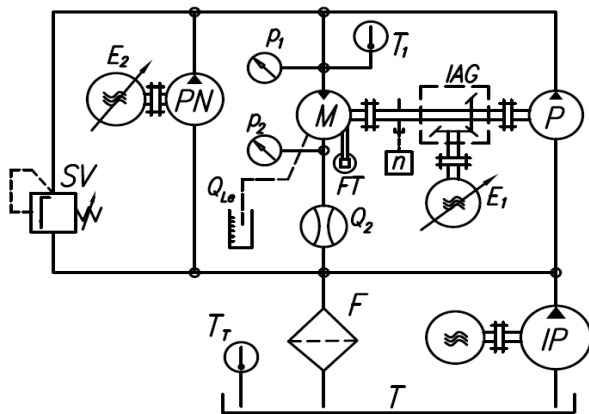

Fig. 7. Diagram of the measurement system of the test stand: P – pump, M – tested hydraulic motor, PN – pump for filling the leaks in P and M, IP – impeller pump (pre-supply pump), SV – safety valve, F – filter, T – reservoir, IAG – intersecting axis gear, E_1 and E_2 – electric motors with frequency converters, T_1 and T_T – temperature sensors, Q_2 – flowmeter, Q_{Le} – leakage measurement, FT – force transducer (torque measurement), n – inductive sensor (rotational speed measurement)

First, the satellite motor was tested using oil as the working medium. Next, the test was conducted using water. This motor was not disassembled between tests. During the tests of the motor supplied with both liquids, the same measurement apparatus was used. Only the tanks, pumps, valves, and tubes were changed.

During the test of the motor, the following parameters were measured:

- the pressure p_1 in the motor inlet port (strain gauge pressure transducers, range = 0–25 bars and 0–250 bars, accuracy = 0.3%);
- the pressure p_2 in the motor outflow port (strain gauge pressure transducer, range = 0–10 bar, accuracy = 0.3%);
- the torque M on the motor shaft [strain gauge force transducer mounted on the arm at 0.5 m (arm attached to the motor body), range = 0–100 N, class 0.1];
- the motor absorbency Q (the flow rate to the motor) (piston flow meter, range = 0–200 l/min, class 0.2);
- the rotational speed of the motor shaft n (inductive sensor, accuracy of measurement = ±1 rpm);
- the temperature T_1 of liquid in the inlet port of the motor (RTD temperature sensor, class A, max. measurement

error = 0.5 °C). The temperature of liquid was stabilized (for oil, $T_1 = 43 \pm 2$ °C; for water, $T_1 = 27 \pm 2$ °C).

The satellite motor was examined using:
- Total Azolla 46 oil ($\nu = 40$ mm²/s, $\rho = 873$ kg/m³, $\mu = 35$ mPas);
- pure tap water ($\nu = 0.853$ mm²/s, $\rho = 999$ kg/m³).

It should be added that in the satellite mechanism $R_a = 0.1$ μm.

RESULTS OF EXPERIMENT AND SIMULATION

TORQUE OF LOSSES IN UNLOADED MOTOR

The experimentally determined characteristics of mechanical losses $M_{l\,(M=0)}$ in an unloaded motor are shown in Fig. 8 and Fig. 9.
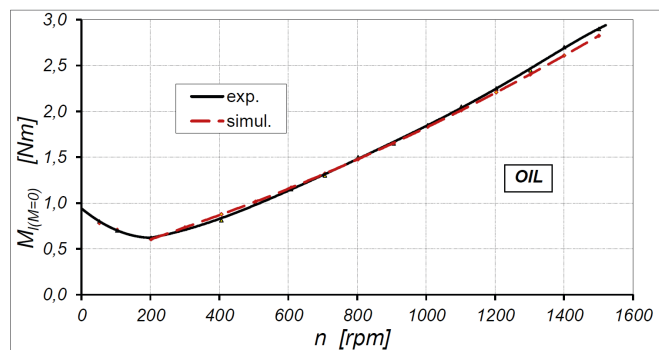

Fig. 8. Characteristics of $M_{l(M=0)} = f(n)$ in unloaded motor supplied with oil: results of experiment and simulation
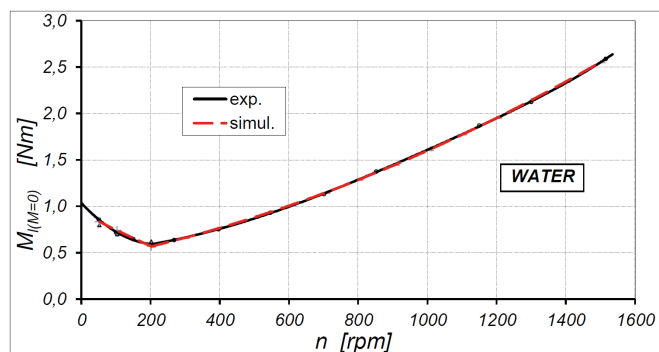

Fig. 9. Characteristics of $M_{l(M=0)} = f(n)$ in unloaded motor supplied with water: results of experiment and simulation

In the motor supplied with oil, $M_{bs-O} = 0.38$ Nm at $p_2 = 0.5$ MPa. However, in the motor supplied with water, $M_{bs-W} = 0.40$ Nm. Thus, $M_{bs-W} > M_{bs-O}$ due to the poorer lubricating properties of water. The values of the constant are given in Table 1.

The values of coefficients C_{bs}, C_{id-m}, and C_{dsv} can be calculated from the equation of the trend line of experimental characteristics (Fig. 8 and Fig. 9). These values are given in Table 1.

TORQUE OF LOSSES IN LOADED MOTOR

The torque of mechanical losses M_l in a loaded motor was calculated according to the formula:

$$M_l = \frac{q \cdot \Delta p_i}{2 \cdot \pi} - M \qquad (27)$$

The experimentally determined characteristics of mechanical losses M_l as a function of rotational speed n of the motor shaft and the load M are shown in Fig. 10 and Fig. 11. The characteristics of component M_{mf} are shown in Fig. 12 and Fig. 13. The coefficient C_{lm} was selected so as to obtain the best fit of a curve to the results of the experiment and is given in Table 1.
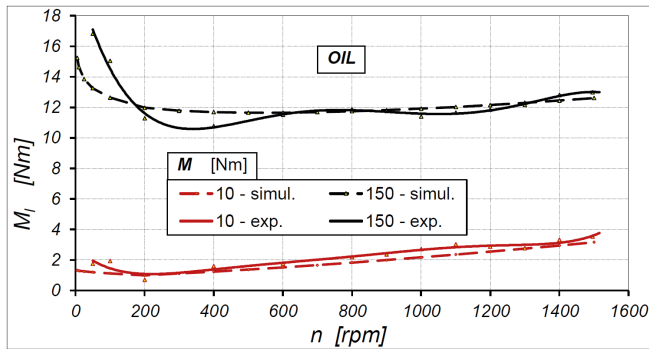

Fig. 10. Characteristics of $M_l = f(n)$ in motor supplied with oil: results of experiment and simulation
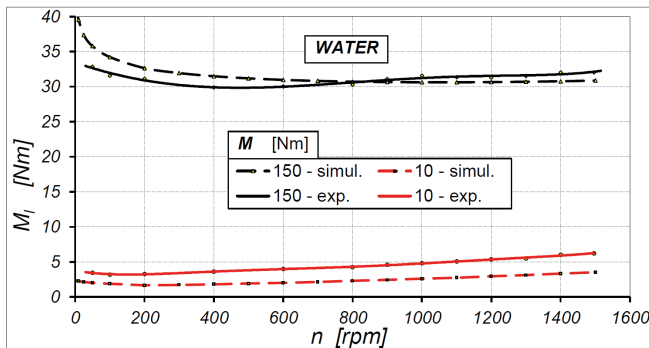

Fig. 11. Characteristics of $M_l = f(n)$ in motor supplied with water: results of experiment and simulation
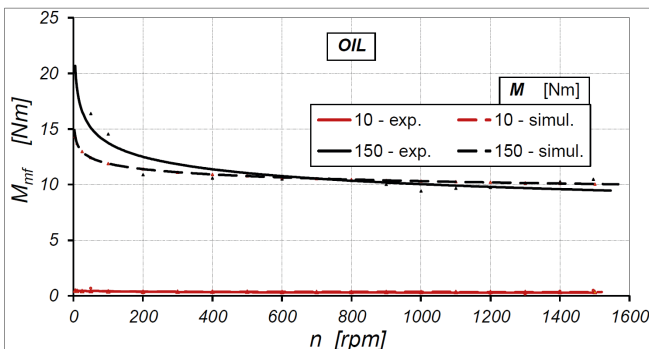

Fig. 12. Characteristics of $M_{mf} = f(n)$ in motor supplied with oil: results of experiment and simulation
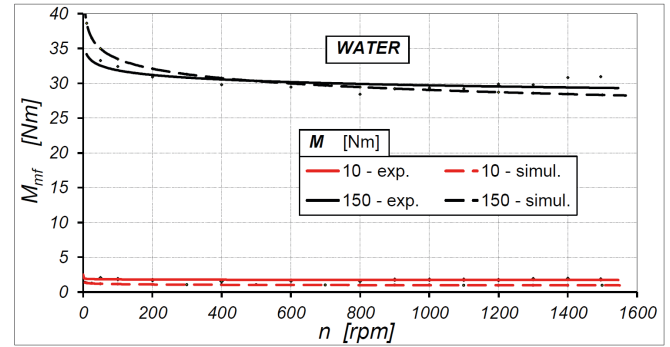

Fig. 13. Characteristics of $M_{mf} = f(n)$ in motor supplied with water: results of experiment and simulation

Tab. 1. Values of coefficients in the model of mechanical losses

	$\Delta M_{lm-0(M=0)}$	C_{bs}	C_{id}	C_{dsv}	C_{lm}
	[Nm]	$\left[\frac{Nm}{MPa}\right]$	$\left[\frac{Nm}{mm^5 \cdot rpm^2}\right]$	$\left[\frac{Nm}{mm^6 \cdot rpm}\right]$	[–]
Oil	0.46	0.76	47.1˙10-9	5.41˙10-6	0.235
Water	0.53	0.80	59.68˙10-9	3.68˙10-6	0.265
For both oil and water: α = 1.25; β = 0.062					

THE RATIO OF TORQUE OF LOSSES IN A MOTOR SUPPLIED WITH WATER AND WITH OIL

The test results have shown that the torque of losses M_{l-W} in the motor supplied with a liquid of low viscosity and poor lubricant properties (water) is greater than the torque of losses M_{l-O} in the motor supplied with oil. The ratio of these torques is:

$$k_{Ml} = \frac{M_{l-W}}{M_{l-O}} \qquad (28)$$

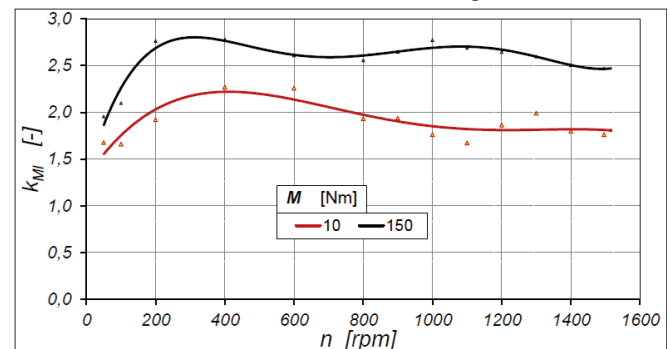The values of this ratio are shown in Fig. 14.

Fig. 14. The ratio of torque of losses in the motor supplied with water and with oil

So, the mechanical losses in the motor supplied with water are as much as 2.8 times greater than the mechanical losses in the motor supplied with oil.

INCREASE OF PRESSURE IN MOTOR WORKING CHAMBERS

The mechanical losses have an impact on the pressure increase Δp_i in the working chambers and on the pressure increase Δp in the motor. In effect,

$$\Delta p = p_1 - p_2 = \underbrace{\frac{2 \cdot \pi}{q} \cdot (M + M_l)}_{\Delta p_i} + \Delta p_{ich} \qquad \textbf{(29)}$$

This is of great practical importance. The use of water instead of oil causes an increase of the pressure drop in the motor. For example:

a) for M = 10 Nm and n = 1500 rpm in a motor supplied with
  – water: Δp_i = 3.1 MPa,
  – oil: Δp_i = 2.58 MPa.
  The difference is 0.52 MPa.

b) for M = 150 Nm and n = 1500 rpm in motor supplied with
  – water: Δp_i = 34.55 MPa,
  – oil: Δp_i = 31.1 MPa.
  The difference is 3.45 MPa.

Thus, in relation to the motor supplied with oil, the supply with water results in:

a) a 20% increase of Δp_i at a small load M of the motor;

b) an 11% increase of Δp_i at a large load M of the motor.

As a result of the higher pressure drop in the motor supplied with water, there is an increase in internal leakage and a reduction in motor speed (with Q = const.).

CONCLUSIONS

Experimental studies have shown that the greatest influence on the mechanical losses results from the friction between elements of the operating mechanism. In addition, the type of liquid has the biggest influence on the value of torque losses in this operating mechanism. Much smaller values of mechanical losses occur in the bearings and seals. The type of liquid has the least impact on the mechanical losses in these elements. The smallest component of the mechanical losses is the component depending on the viscosity of the liquid, that is, the mechanical losses caused by viscous friction in gaps.

The mathematical model of torque of mechanical losses has been developed based on the analysis of sources of these losses. It has been shown that on the basis of experimental results, it is possible to calculate the coefficients of the model. Based on the proposed mathematical model, the characteristics of the torque of losses have been calculated and then compared with the experimental results. It has been found that the simulation results differ from the results of the experiment:

a) at low speed (n = 50 rpm):
  – for M = 10 Nm: about 40% for both oil and water;
  – for M = 150 Nm: 21% for oil and 9% for water;

b) at high speed (n = 1500 rpm):
  – for M = 10 Nm: about 10% for oil and 43% for water;
  – for M = 150 Nm: 3% for oil and 4% for water.

Thus, the mathematical model describes the torque of losses in the motor supplied with mineral oil or water quite accurately. Therefore, the mechanical losses in the motor supplied with various liquids can be assessed by comparison of the model coefficients.

The model of mechanical losses presented in this publication is also suitable to describe mechanical losses in a satellite pump.

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CONTACT WITH THE AUTHOR

Paweł Śliwiński
e-mail: pawel.sliwinski@pg.edu.pl

Gdansk University of Technology,
Narutowicza 11/12, 80-233 Gdańsk,
Poland