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Thermal stability of lubricants in cycloidal reducers

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ABSTRACT

There is intensive rolling and sliding between the meshing elements of a cycloidal reducer, whereby a significant amount of mechanical energy is converted into heat and absorbed by the lubricant. In order to stabilize the temperature of the lubricant, it is necessary to achieve thermal equilibrium, i.e. the amount of heat dissipated should equal the amount of heat generated. A complex task of determining the equilibrium temperature of a lubricant generally involves theoretical analysis, numerical calculation, computational simulations and experimental testing.

The aim of this paper is to develop a model to predict the amount of heat dissipated from the outer surface of the housing as well as the equilibrium temperature of the lubricant. The model is based on the basic laws of thermodynamics, while the computational simulation is performed for an actual cycloidal reducer in the Matlab software package. The simulation results show changes in the equilibrium temperature in relation to: the coefficient of heat transfer from the lubricant to the inner wall of the housing, ambient air velocity, wall thickness and housing material.

KEYWORDS

Cycloidal reducer, Heat generation, Heat dissipation, Thermal equilibrium, Thermal stability of lubricants

1. INTRODUCTION

Compared to other speed reducers, cycloidal reducers cover a wide range of gear ratios, have high efficiency, compact construction, light weight, low noise and low vibration. Thermal stability, which plays a crucial role in maintaining the proper functioning and intended use of cycloidal reducers, mainly depends on the lubricant used and the housing design.Inside the housing, there is intensive rolling and sliding between the meshing elements, whereby a significant amount of mechanical energy is converted into heat and absorbed by the lubricant. An increase in the temperature of the outer surface of the housing and the lubricant in relation to the ambient temperature indicates excessive heating. In order to stabilize the temperature of the lubricant, it is necessary to achieve thermal equilibrium, i.e. the amount of heat dissipated should equal the amount of heat generated.

There is currently little research on thermal behaviour of cycloidal reducers. Zah et al. [1] used an equilibrium equation, i.e. they equalled the amount of heat generated and the amount of heat dissipated in order to determine the operating temperature of the housing. Kudryavcev's model was used to predict the amount of mechanical energy converted into heat [2]. Nowadays, after years of research, other models based on a larger number of contacts between the elements are available [3-6]. Mihailidis et al. [7-8] used thermal elastohydrodynamic lubrication theory to determine the contact area temperatures independently of power losses. Wei et al. [9] investigated distribution of the temperature field on one tooth of a cycloidal reducer, assuming that the load and the heat transfer are the same for all the teeth. Olejarczyk et al. [10] investigated operating temperatures of mineral and synthetic oils. A lot of researchers studied thermal stability of worm gears [11-13]. Xue and Xu [14] investigated the heat transfer mechanism of each element in a cylindrical helical gearbox. Zivkovic [15] studied planetary gearbox and established a relation between mechanical losses and the amount of heat dissipated to the ambient air.

Based on the review of the literature, it can be concluded that the lubricant operating temperature has a crucial impact on the thermal stability of a gearbox. Optimal lubrication can be ensured only within the right range of operating temperature. Therefore, the aim of this paper is to define a simplified theoretical model to predict the operating temperature of the lubricant and the amount of heat dissipated only through the housing.

2. ENERGY BALANCE IN THE CYCLOIDAL REDUCER

In order to establish the thermal stability and to determine the operating temperature of the lubricant in the cycloidal reducer, it is necessary to define the place and the manner, i.e. the elements on which the mechanical energy is converted into heat and the way the heat is dissipated.

In general, the contacts between the elements of a cycloidal reducer can be divided into load-independent and load-dependent contacts [6,16]. Apart from the housing parts, the load-independent contacts are mainly related to the lubricant and the shaft sealing elements. Load-dependent contacts can be divided into the contacts:

- in the cycloid disc bearing;
- between the output rollers and holes in the cycloid disc;
- between the output rollers and their pins;
- between the ring gear rollers and the cycloid disc teeth;
- between the ring gear rollers and their pins;
- in the input shaft bearings;
- in the output shaft bearings.

The entire mechanical energy converted into heat during these contacts is absorbed by the lubricant and transferred by convection to all internal parts, including the inner surface of the housing. The heat is dissipated to the ambient air through a relatively large outer surface of the housing, through the housing foot, but also through free parts of the shaft, regardless of their relatively small surface areas. Table 1 shows the predominant ways of heat transfer between the cycloidal reducer elements.

Table 1: The ways of heat transfer in the cycloidal reducer

Heat transfer	The predominant mode of heat transfer
Conduction	All internal elements in contact with each other
Convection	All internal elements in contact → lubricant
	Lubricant → air in the gearbox
	Housing → ambient air
	Housing foot → ambient air
	Input/output shaft → ambient air
	Input/output shaft coupling → ambient air
Radiation	Housing → ambient air
	Housing foot → ambient air
	Input/output shaft → ambient air
	Input/output shaft coupling → ambient air

Since a simplified model is used, when determining the equilibrium temperature of the lubricant, the heat dissipated through the housing foot and as well as through the input and output shaft of the cycloidal reducer is neglected.

3. EFFICIENCY OF THE CYCLOIDAL REDUCER

As a result of many years of research, numerous models are available to predict the efficiency of cycloidal reducers. One of the most commonly used models was defined by Malhotra [3]. It is rather simple, easy to use and it gives good results. The total efficiency is determined based on the following expression:

$$\eta = \frac{T_{in} \cdot 2\pi - W_f}{T_{in} \cdot 2\pi} \tag{1}$$

where: T_{in} – input shaft torque; W_f – total work done by the friction force.

The total work done by the friction force (W_f) is the integral of the sum of elementary work: friction in the bearing of cycloid disc, rolling friction between the output rollers and the holes in the cycloid disc, rolling friction between the cycloid discs and the ring gear rollers, sliding friction between the output rollers and their pins, as well as sliding friction between the ring gear rollers and their pins. The comprehensive expression can be written in the following form [6]:

$$\begin{aligned} W_f = {} & \frac{f_{r1} \cdot D_{SR} \cdot z_1}{d_{kt}} \int_0^{\frac{2\pi}{z_1}} F_E(\beta) \cdot d\beta + z_1 \cdot \left(f_{r2} + \frac{\mu_{s1} \cdot d_{VK}}{2} \right) \cdot \int_0^{\frac{2\pi}{z_1}} \sum_{j=1}^{q} F_{Kj}(\beta) \cdot d\beta + \\ & + (z_1 + 1) \cdot \left(f_{r3} + \frac{\mu_{s2} \cdot d_0}{2} \right) \cdot \int_0^{\frac{2\pi}{z_1}} \sum_{i=1}^{p} F_{Ni}(\beta) \cdot d\beta \end{aligned} \tag{2}$$

where: f_{r1} – lever arm of the rolling friction of the cycloid disc $(f_{r1} = \mu_{r1} \cdot d_{kt} / 2)$; μ_{r1} – coefficient of rolling friction in the cycloid disc bearing; $F_E(\beta)$ – current value of the eccentric cam force; D_{SR} – mean diameter of the cycloid disc bearing $(D_{SR} = (D_{CZ} + d_{CZ}) / 2)$; D_{CZ} – outer diameter of the cycloid disk bearing; f_{r2} – lever arm of the rolling friction of the output roller $(f_{r2} = \mu_{r2} \cdot D_{VK} / 2)$; μ_{r2} – coefficient of rolling friction between the output rollers and the hole in the cycloid disc; $F_{Kj}(\beta)$ – current value of the output force on the *j-th* output roller; q – current number of the output rollers participating in the load transfer (if the total number of output rollers (u) is even, then $(q = u / 2)$, if it is an odd number, then $(q = (u-1) / 2)$); f_{r3} – lever arm of the rolling friction of the ring gear rollers $(f_{r3} = \mu_{r3} \cdot D_0 / 2)$; μ_{r3} – coefficient of the friction between the ring gear rollers and the cycloid disc; $F_{Ni}(\beta)$ – current value of the normal force on the *i-th* ring gear roller; p – current number of ring gear rollers participating in the load transfer. If the total number of the ring gear rollers is even, then $(p = z_2 / 2)$, and if it is an odd number, then $(p = (z_2 + 1) / 2)$.

4. THE QUANTITY OF DISSIPATED HEAT

Defining a mathematical model for the amount of heat dissipated through the housing wall is very complex due to the specific geometry of the housing. Therefore, a simplified model (Figure 1) is used with the following approximations:

- the housing is homogeneous and isotropic with flat vertical and cylindrical horizontal surfaces;
- the inside surface of the housing is in contact only with a constant-temperature lubricant.

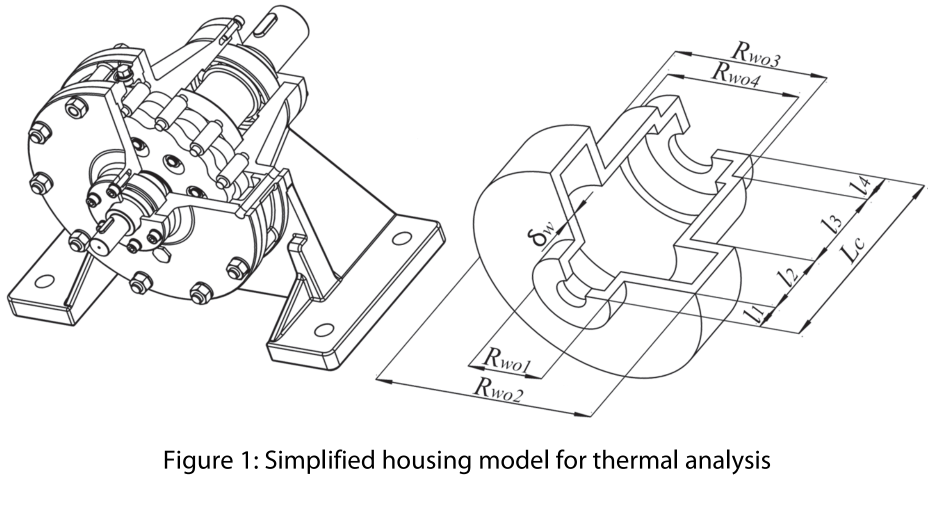


Figure 1: Simplified housing model for thermal analysis

4.1 The amount of heat dissipated through the flat wall of the housing

For the one-dimensional heat flow through the flat wall (Q_{hf}), which receives heat from the lubricant by convection on the inside and transfers heat to the ambient air by convection and radiation on the outside (Figure 2), the following equations can be written [17-18]:

$$Q_{hf} = \alpha_{lub} \cdot A_{wi,f} \cdot \left(t_{lub} - t_{wi,f}\right) = \frac{\lambda_w}{\delta_w} \cdot A_{we,f} \cdot \left(t_{wi,f} - t_{wo,f}\right) = \left(\alpha_{air,fcon} + \alpha_{air,rad}\right) \cdot A_{wo,f} \cdot \left(t_{wo,f} - t_{air}\right) \tag{3}$$

$$Q_{hf} = \frac{\left(t_{lub} - t_{wi,f}\right)}{1/\alpha_{lub} \cdot A_{wi,f}} = \frac{\left(t_{wi,f} - t_{wo,f}\right)}{\delta_w / \lambda_w \cdot A_{we,f}} = \frac{\left(t_{wo,f} - t_{air}\right)}{1/\left(\alpha_{air,fcon} + \alpha_{air,rad}\right) \cdot A_{wo,f}} \tag{4}$$

$$Q_{hf} = \frac{\left(t_{lub} - t_{wi,f}\right)}{R_{f,lub}} = \frac{\left(t_{wi,f} - t_{wo,f}\right)}{R_{f,wal}} = \frac{\left(t_{wo,f} - t_{air}\right)}{R_{f,air}} \tag{5}$$

$$Q_{hf} = \frac{\left(t_{lub} - t_{air}\right)}{R_{f,total}} \tag{6}$$

$$R_{f,total} = R_{f,lub} + R_{f,wal} + R_{f,air} \tag{7}$$

where: t_{lub} – lubricant temperature; $t_{wi,f}$ – temperature of the inner flat wall of the housing; $t_{wo,f}$ – temperature of the outer flat wall of the housing; t_{air} – ambient temperature; $A_{wi,f}$ – inner surface area of the flat wall of the housing; $A_{wo,f}$ – outer surface area of the flat wall of the housing; $A_{we,f}$ – equivalent surface area of the flat wall $\left(A_{we,f} = \left(A_{wi,f} + A_{wo,f}\right)/2\right)$; α_{lub} – coefficient of heat transfer from the lubricant to the inner wall of the housing (according to Jelaski [19], the value of the coefficient (α_{lub}) ranges from $150 \div 300 W/(m^2K)$, but it is usually taken to be $\alpha_{lub} = 200 W/(m^2K)$ [20]); $\alpha_{air,fcon}$ – coefficient of convective heat transfer from the outer flat wall of the housing to the ambient air; $\alpha_{air,rad}$ – coefficient of heat transfer by radiation from the outer wall of the housing to the ambient air; λ_w – coefficient of thermal conductivity (for cast steel $\lambda_w = 50 W/mK$) [21]; δ_w – equivalent thickness of the vertical flat wall of the housing; $R_{f,lub}$ – thermal resistance to heat transfer from the lubricant to the inner wall; $R_{f,wal}$ – thermal resistance to heat transfer through the flat wall; $R_{f,air}$ – thermal resistance to heat transfer from the outer flat wall to the ambient air.

Since heat always flows from a higher to a lower temperature area in order to equalize the temperatures, it is assumed that $\left(t_{lub} > t_{air}\right)$.

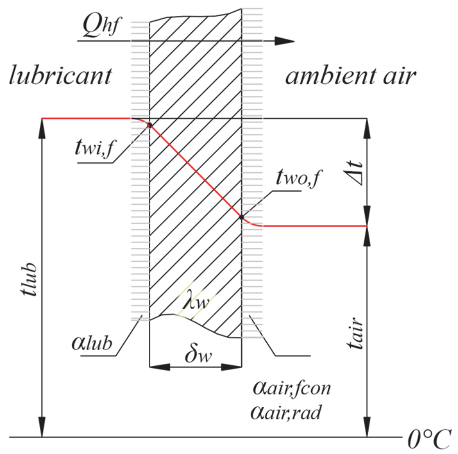


Figure 2: Heat dissipation through the vertical flat wall of the housing (Δt – the temperature difference between the lubricant and the ambient air)

If the outer surface temperature of the housing wall (t_{wo}) is known, Churchill and Chu [22-23] proposed the following correlation equations to determine the heat transfer coefficient $(\alpha_{air,fcon})$:

$$\alpha_{air,fcon} = \frac{N_{u,fo} \cdot \lambda_{air}}{H_{wo}} \tag{8}$$

$$N_{u,fo} = 0,68 + \frac{0,670 \cdot R_{a,fo}^{\ 1/4}}{\left[1 + \left(0,492 / P_{r,fo}\right)^{9/16}\right]^{4/9}} \qquad R_{a,fo} < 10^9 \tag{9}$$

$$N_{u,fo} = \left\{0,825 + \frac{0,387 \cdot R_{a,fo}^{\ 1/6}}{\left[1 + \left(0,492 / P_{r,fo}\right)^{9/16}\right]^{8/27}}\right\}^2 \qquad R_{a,fo} > 10^9 \tag{10}$$

$$R_{a,fo} = G_{r,fo} \cdot P_{r,fo} \tag{11}$$

$$R_{e,fo} = \frac{\rho_{air} \cdot \mathrm{w}_{air} \cdot H_{wo}}{\mu_{air}} = \frac{\mathrm{w}_{air} \cdot H_{wo}}{\nu_{air}} \tag{12}$$

$$P_{r,fo} = \frac{c_{p,air} \cdot \mu_{air}}{\lambda_{air}} \tag{13}$$

$$G_{r,fo} = \frac{\beta \cdot g \cdot H_{wo}^{\ 3} \cdot \left(t_{wo} - t_{air}\right)}{\nu_{air}^{\ 2}} \tag{14}$$

where: $N_{u,fo}$ – Nusselt number; $R_{e,fo}$ - Reynold number; $P_{r,fo}$ – Prandtl number; $G_{r,fo}$ – Grashof number. The ambient air parameters include: ρ_{air} – density; w_{air} – air velocity; μ_{air} – dynamic viscosity; ν_{air} – kinematic viscosity; $c_{p,air}$ – specific heat capacity at the constant pressure; λ_{air} – coefficient of thermal conductivity. The rest of the quantities are: H_{wo} – height of the outer flat wall; β – coefficient of thermal expansion $\left(\beta = 1/t_{ekv}\right)$; t_{ekv} – equivalent temperature $\left(t_{ekv} = \left(t_{wo,f} - t_{air}\right)/2\right)$, g – acceleration of gravity $\left(g = 9.81 m/s^2\right)$.

For practical calculations, an empirical expression which gives acceptable results can also be used [24]:

$$\alpha_{air,fcon} = 7 + 12 \cdot \sqrt{w_{air}} \tag{15}$$

The heat transfer coefficient $\left(\alpha_{air,rad}\right)$ can be determined based on the expression [18]:

$$\alpha_{air,rad} = \varepsilon \cdot \sigma \cdot \left(t_{wo}^{\ 2} + t_{air}^{\ 2}\right) \cdot \left(t_{wo} + t_{air}\right) \tag{16}$$

where: ε – housing emissivity coefficient which depends on the material, temperature and surface quality; σ - Stefan-Boltzmann constant $\left(\sigma = 5.67 \cdot 10^{-8}\, W/m^2K^4\right)$ [18].

If the lubricant velocity inside the housing $\left(\mathrm{w}_{lub}\right)$ is known, the coefficient of heat transfer (α_{lub}) from the lubricant to the inner wall of the housing (for the Reynolds number in the range of $5 \cdot 10^5 < R_{e,fi} < 10^7$) can be determined based on the expression [22]:

$$\alpha_{lub} = 0.037 \cdot R_{e,fi}^{\ 0,8} \cdot P_{r,fi}^{\ 1/3} \cdot \frac{\lambda_{lub}}{H_{wi}} \tag{17}$$

4.2 The amount of heat dissipated through the cylindrical wall of the housing

For one-dimensional heat flow through the cylindrical wall $\left(Q_{hc}\right)$, which receives heat from the lubricant by convection on the inside and transfers heat to the ambient air by convection and radiation on the outside (Figure 3), the following equations can be written [17-18]:

$$\mathrm{Q_{hc} = R_{wi} \cdot \pi \cdot L_c \cdot \alpha_{lub} \cdot \left(t_{lub} - t_{wi,c}\right) = 2\pi \cdot L_c \cdot \frac{\lambda_w}{\ln\left(\frac{R_{wo}}{R_{wi}}\right)} \cdot \left(t_{wi,c} - t_{wo,c}\right) = R_{wo} \cdot \pi \cdot L_c \cdot \left(\alpha_{air,ccon} + \alpha_{air,rad}\right) \cdot \left(t_{wo,c} - t_{air}\right)}. \tag{18}$$

$$Q_{hc} = \frac{\left(t_{lub} - t_{wi,c}\right)}{1/\left(R_{wi} \cdot \pi \cdot L_c \cdot \alpha_{lub}\right)} = \frac{\left(t_{wi,c} - t_{wo,c}\right)}{ln\left(R_{wo} / R_{wi}\right) / 2\pi \cdot L_c \cdot \lambda_w} = \frac{\left(t_{wo,c} - t_{air}\right)}{1/\left(R_{wo} \cdot \pi \cdot L_c \cdot \left(\alpha_{air,ccon} + \alpha_{air,rad}\right)\right)} \tag{19}$$

$$Q_{hc} = \frac{\left(t_{lub} - t_{wi,c}\right)}{R_{c,lub}} = \frac{\left(t_{wi,c} - t_{wo,c}\right)}{R_{c,wal}} = \frac{\left(t_{wo,c} - t_{air}\right)}{R_{c,air}} \tag{20}$$

$$Q_{hc} = \frac{\left(t_{lub} - t_{air}\right)}{R_{c,total}} \tag{21}$$

$$R_{c,total} = R_{c,lub} + R_{c,wal} + R_{c,air} \tag{22}$$

where: $t_{wi,c}$ – temperature of the inner cylindrical wall of the housing; $t_{wo,c}$ – temperature of the outer cylindrical wall of the housing; R_{wi} – equivalent inner radius of the cylindrical housing wall $\left(R_{wi,ekv} = \left(\sum R_{wi} \cdot l_i\right) / \sum l_i\right)$; R_{wo} – equivalent outer radius of the cylindrical housing wall $\left(R_{wo,ekv} = \left(\sum R_{wo} \cdot l_i\right) / \sum l_i\right)$; L_c – length of the cylindrical housing; $\alpha_{air,ccon}$ – coefficient of convective heat transfer from the outer cylindrical housing wall to the ambient air; $R_{c,lub}$ – thermal resistance to heat transfer from the lubricant to the inner cylindrical wall; $R_{c,wal}$ – thermal resistance to heat transfer through the cylindrical wall; $R_{c,air}$ – thermal resistance to heat transfer from the outer cylindrical wall to the ambient air.

To determine the external coefficient ($\alpha_{air,ccon}$), Hilpert [25-26] suggested the following expression:

$$\alpha_{air,ccon} = \frac{N_{u,co} \cdot \lambda_{air}}{2R_{wo}} \tag{23}$$

$$N_{u,co} = C_2 \cdot R_{e,co}{}^{m} \cdot P_{r,co}{}^{\frac{1}{3}} \tag{24}$$

$$R_{e,co} = \frac{2R_{wo} \cdot W_{air} \cdot \rho_{air}}{\mu_{air}} \tag{25}$$

$$P_{r,co} = \frac{c_{p,air} \cdot \mu_{air}}{\lambda_{air}} \tag{26}$$

where: C_2, m – constants dependent on the range of the Reynolds number, as stated in Table 2.

Table 2: Values of constants depending on the range of the Reynolds number [25]

$R_{e,co}$	C_2	m	$R_{e,co}$	C_2	m
$4 \cdot 10^{-1} \div 4 \cdot 10^{0}$	0.989	0.330	$4 \cdot 10^{3} \div 4 \cdot 10^{4}$	0.193	0.618
$4 \cdot 10^{0} \div 4 \cdot 10^{1}$	0.911	0.385	$4 \cdot 10^{4} \div 4 \cdot 10^{5}$	0.027	0.805
$4 \cdot 10^{1} \div 4 \cdot 10^{3}$	0.683	0.466			

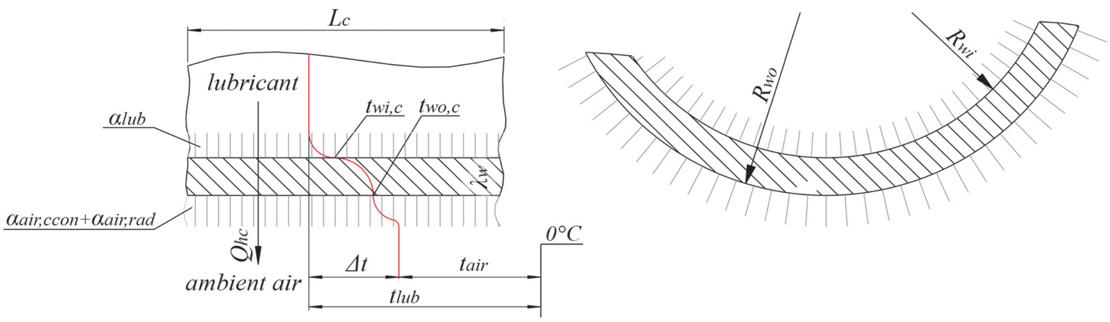


Figure 3: Heat dissipation through the cylindrical wall of the housing (Δt – the temperature difference between the lubricant and the ambient air)

5. MATHEMATICAL MODEL

After certain period of operation at constant torque, thermal equilibrium will be reached, i.e. the amount of mechanical energy converted into heat (Q_G) will equal the amount of heat dissipated (Q_R). This can be mathematically written as:

$$Q_G = Q_R \tag{27}$$

The amount of mechanical energy converted into heat can be determined based on the expression:

$$Q_G = P_{in} \cdot (1 - \eta_{cr}) \tag{28}$$

where: P_{in} – input power, η_{cr} – efficiency of the cycloidal reducer.

However, for a simplified model where only heat transfer through the housing is taken into account, the amount of heat dissipated can be determined based on the expression:

$$Q_R = Q_{hf} + Q_{hc} \tag{29}$$

By equating the equations (28) and (29), a thermal equilibrium equation is obtained:

$$P_{in} \cdot (1 - \eta_{cr}) = Q_{hf} + Q_{hc} \tag{30}$$

and by substituting the equations (6) and (21) in (30), a comprehensive expression is obtained in the following form:

$$P_{in} \cdot (1 - \eta_{cr}) = \frac{(t_{lub} - t_{air})}{R_{f,lub} + R_{f,wal} + R_{f,air}} + \frac{(t_{lub} - t_{air})}{R_{c,lub} + R_{c,wal} + R_{c,air}} \tag{31}$$

After rearranging, the final expression for calculation of the lubricant equilibrium temperature is obtained, provided that all the heat is transferred through cylindrical and flat walls:

$$t_{lub} = \frac{P_{in} \cdot (1 - \eta_{cr})}{\dfrac{1}{R_{f,lub} + R_{f,wal} + R_{f,air}} + \dfrac{1}{R_{c,lub} + R_{c,wal} + R_{c,air}}} + t_{air} \tag{32}$$

Under normal operating conditions, according to Castrol as one of the leading lubricant manufacturers, Optigear EP mineral oil provides good lubrication up to 88 ℃, Optigear Synthetic X PAO based synthetic oil ensures good lubrication up to 100 ℃, Optigear Synthetic 800 synthetic PAG based oil offers good lubrication up to 120 ℃, while Castrol Tribol GR 100-0 PD lithium grease based on mineral oil enables good lubrication up 140 ℃ [27-28]. However, in order to maintain the proper functioning of the cycloidal reducer (lubricants, seals, dilatation of components), the limit temperature should not exceed 65-70 ℃, i.e. it should not exceed the ambient temperature by more than 40℃ [29]. If the temperature is higher, it is necessary to install forced ventilation or increase the heat dissipation area.

6. MODEL TESTING AND SIMULATION RESULTS

This study was performed on a cycloidal reducer with the characteristics given in Table 3 [30]. In order to calculate the heat transfer coefficient $(\alpha_{air,rad})$, the value of the temperature was assumed $(t_{wo} = 80℃)$, while the other quantities were calculated using the given expressions.

Table 3: Basic characteristics of the cycloidal reducer

Characteristics	Value	Characteristics	Value
Input power, $P_{im}\ (kW)$	4	Coefficient of rolling friction, μ_{r1}	0.005
Input speed, $n_{im}\ (min^{-1})$	1420	Coefficient of rolling friction, $\mu_{r2} = \mu_{r3}$	0.0045
Reducer ratio, u_{CR}	13	Coefficient of sliding friction, $\mu_{s1} = \mu_{s2}$	0.05
Output torque, $T_{ou}\ (Nm)$	332	Coefficient of sliding friction, μ_{s3}	0.04

Pitch circle radius of the housing roller, $r\,(mm)$	86	Average air temperature, t_{air} $(℃)$	20
Eccentricity, $e\,(mm)$	4	Average air velocity, w_{air} (m/s)	1.25
Housing roller diameter, $D_0\,(mm)$	14	Thermal conductivity of the housing, λ_w (W/mK)	50
Diameter of the housing roller pin, $d_0\,(mm)$	8	Heat transfer coefficient for lubricant, α_{lub} $W/(m^2K)$	200
Number of output rollers, u	8	Outer surface area of the flat walls of the housing, $A_{wo,f}$ (m^2)	0.0832
Output roller diameter, $D_{VK}\,(mm)$	14	Outer surface area of the cylindrical walls of the housing, $A_{wo,c}$ (m^2)	0.1048
Diameter of the output roller pin, $d_{VK}\,(mm)$	7	Inner surface area of the flat walls of the housing, $A_{wi,f}$ (m^2)	0.0741
Diameter of the cycloid disc hole, $D_{OCZ}\,(mm)$	22	Inner surface area of the cylindrical walls of the housing, $A_{wi,c}$ (m^2)	0.0943
Pitch circle diameter of output rollers, R_{Oiz} (mm)	47.5	Equivalent thickness of the flat walls of the housing, δ_w (mm)	11
Inner diameter of the bearing, $d_{CZ}\,(mm)$	35	Equivalent radius of the outer cylindrical walls, R_{wo} (mm)	181
Outer diameter of the bearing, $D_{CZ}\,(mm)$	40	Equivalent radius of the inner cylindrical walls, R_{wi} (mm)	170
Emissivity coefficient of the housing, ε	0.8	Length of the cylindrical housing, $L_c\,(mm)$	150

Figure 4 shows the change in the lubricant equilibrium temperature (t_{lub}) in relation to the change in the load of the machine (T_{load}) working at a constant speed $(n_{ou} = 109.23\ min^{-1})$. The simulation was performed for three different heat transfer coefficients (α_{lub}) which varied in the range from 150 to 250 $W/(m^2K)$ and which depend on the following lubricant parameters: density (ρ_{lub}), flow velocity (w_{lub}), dynamic viscosity (μ_{lub}), kinematic viscosity (ν_{lub}), specific heat capacity at constant pressure $(c_{p,lub})$ and thermal conductivity coefficient (λ_{lub}). The load (T_{load}) varied in the range from 25 Nm to the nominal 332 Nm, while the overload of the electric motor was not simulated. A significantly higher lubricant temperature (t_{lub}) was obtained because the model lacked heat transfer through the foot, shafts and couplings. The temperature (t_{lub}) changes depending on the ratio between the amount of heat generated and the amount of heat dissipated. The values of the characteristic points in Figure 4 are shown in Table 4. With an increase in the load of the working machine (T_{load}), the impact of the value of the heat transfer coefficient (α_{lub}) on the lubricant equilibrium temperature (t_{lub}) increases.

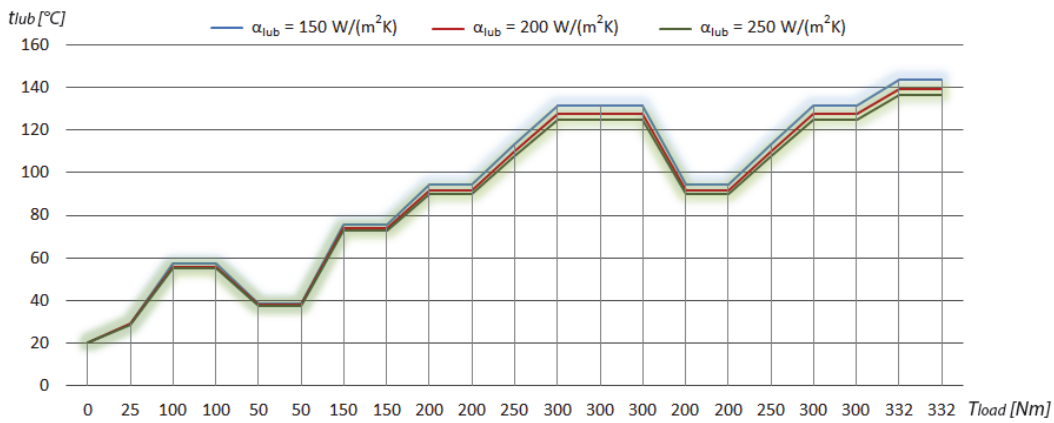


Figure 4: Lubricant equilibrium temperatures in relation to the ratio between the amount of heat generated and the amount of heat dissipated for different values of the heat transfer coefficient (α_{lub})

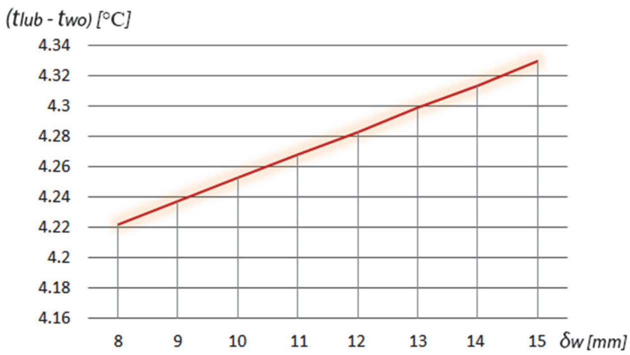
Table 4: Lubricant equilibrium temperatures at characteristic points for different values of the coefficient (α_{lub})

$T_{load}\,(Nm)$	0	25	50	100	150	200	250	300	332
$\alpha_{lub} = 150\left(W/(m^2K)\right)$	22 ℃	29.3 ℃	38.59 ℃	57.22 ℃	75.81 ℃	94.44 ℃	113 ℃	131.6 ℃	143.5 ℃
$\alpha_{lub} = 200\left(W/(m^2K)\right)$	22 ℃	28.96 ℃	37.92 ℃	55.87 ℃	73.79 ℃	91.74 ℃	109.7 ℃	127.6 ℃	139.1 ℃
$\alpha_{lub} = 250\left(W/(m^2K)\right)$	22 ℃	28.76 ℃	37.52 ℃	55.06 ℃	72.57 ℃	90.12 ℃	107.6 ℃	125.1 ℃	136.4 ℃

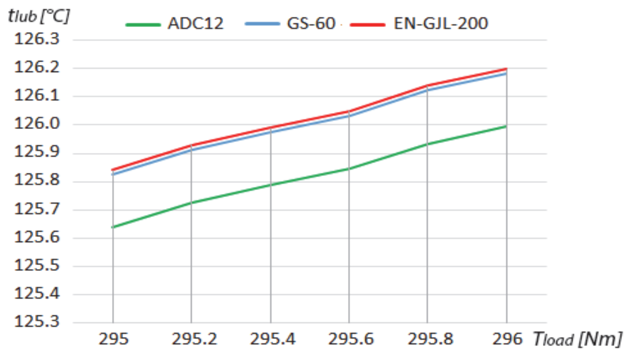
Figure 5.a shows how the temperature difference between the lubricant and the outer wall of the housing ($t_{lub} - t_{wo}$) changes as a result of a change in the wall thickness (δ_w). For this simulation, the approximation $(R_{wi} = R_{wo} - \delta_w)$ was introduced. The load (T_{load}) had a constant value of 50 Nm, while the wall thickness varied in the range from 8 to 15 mm. Based on this simulation it can be concluded that the change in the wall thickness (δ_w) has no significant effect on the temperature difference. With an increase of 7 mm in the wall thickness (δ_w), the temperature difference increases by 0.108 ℃.

Figure 5.b shows the change in the lubricant equilibrium temperature (t_{lub}) in relation to the change of the coefficient of thermal conductivity (λ_w), which depends on the type of the housing material. The use of grey cast iron EN-GJL-200 ($\lambda_w = 48 W/mK$) [31], cast steel GS-60 $(\lambda_w = 50 W/mK)$ and aluminum alloy AlSi11Cu3 / ADC-12 $(\lambda_w = 92 W/mK)$ [32] was analyzed. The load (T_{load}) varied in the range from 295 Nm do 296 Nm.

Based on this simulation, it can be concluded that aluminium alloy provides the best heat dissipation, while there is no significant difference between the grey cast iron and cast steel due to the extremely small surface area of the housing.



a)



b)

Figure 5: (a) Dependence of the temperature difference $(t_{lub} - t_{wo})$ on the change in the housing wall thickness; (b) Dependence of the lubricant equilibrium temperature (t_{lub}) on the housing material

Figure 6. shows the change in the lubricant equilibrium temperature (t_{lub}) in relation to the change in the ambient air velocity (w_{air}). In case the gearbox is located indoors, the speed of 1.25 m/s is usually taken [33]. The operation of the gearbox located outdoors, where the values of the air velocity are 2 m/s and 3 m/s, was also analysed. The load (T_{load}) varied in the range from 300 Nm to the nominal 333 Nm. Based on this simulation, it can be concluded that the air velocity significantly affects the heat dissipation and the equilibrium temperature.

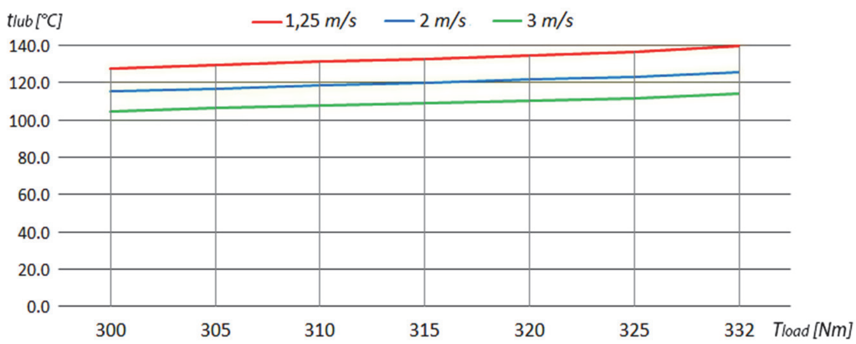


Figure 6: Dependence of the lubricant equilibrium temperature on the ambient air velocity

7. CONCLUSION

A model to predict both the amount of heat dissipated from the outer surface of the housing and the lubricant equilibrium temperature has been developed in this paper. The model is based on the correlation between the mechanical energy converted into heat and the amount of heat dissipated to the ambient air. Due to the specific geometry of the housing, two approximations have been introduced: that the housing is a homogeneous and isotropic medium with flat and cylindrical surfaces, and that the inner side of the housing is in contact only with a constant-temperature lubricant.

An actual cycloidal reducer with a nominal power of 4 kW, a nominal torque on the output shaft of 332 Nm and a transmission ratio of 13 was studied. Based on the testing performed, valuable conclusions have been drawn.

When the cycloidal reducer is loaded up to the nominal 332 Nm, the equilibrium temperature (t_{lub}) changes depending on the balance between the amount of heat generated and the heat dissipated. The value of the heat transfer coefficient (α_{lub}) has no significant effect at lower loads, but at higher loads its impact is prominent.

From the aspect of geometry, it can be noticed that the change in the wall thickness (δ_w) does not have significant effects on the temperature difference between the lubricant and the outer wall $(t_{lub} - t_{wo})$. With an increase in the wall thickness (δ_w) by 7 mm, the temperature difference increases by 0.108 ℃.

Tests on different housing materials have shown that aluminium alloy (AlSi11Cu3/ADC-12) provides the fastest heat dissipation, while with grey cast iron (EN-GJL-200) and cast steel (GS-60) the heat dissipation is much slower.

As for the weather conditions, the velocity of the air (w_{air}) significantly affects the heat dissipation and, consequently, the value of the equilibrium temperature (t_{lub}). By changing the velocity from 1.25 m/s to 3 m/s, at the same load, the equilibrium lubricant temperature (t_{lub}) decreases by 25 ℃.

Our further studies will be devoted to expanding the model to include the impact of the housing foot, shafts and couplings on the heat dissipation, as well as to performing experimental verification of the developed model.

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