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Selecting the elastic coupling and lubrication system to a single speed reducer to a mathcad computer-assisted

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Abstract. The Mathcad interface allows users to combine a variety of different elements (math, descriptive text, and support images) in the form of a natural-readable worksheet. Since mathematics is essential for the program, mathematics is inherently alive, dynamically recalculating, as upstream values change. This allows easy manipulation of input variables, hypotheses and expressions, which in turn are updated in real time. The following example serves to highlight Mathcad's capabilities rather than to provide specific details about the functionality of the individual product.

This paper aims to make selection of the elastic coupling and lubrication system for the design of a single stage speed reducer using MathCad program.

1. Introduction

Couplings make a permanent or intermittent connection between two consecutive elements of a transmission for the purpose of transmitting the rotation motion and torque moment, without modifying the motion law.

From the coupling definition mode, the main function of these couplings is: transmission of the movement and the torque moment.

The wide range of coupling applications has required attachment and other additional functions:

- ⇒ compensation of position deviations of coupled elements (axial, radial, angled or combined) due to execution and / or assembly errors;
- ⇒ protection against shocks and vibrations;
- ⇒ interruption of the link between the two elements;
- ⇒ limitation of the transmitted task;
- ⇒ speed limitation;
- ⇒ the limitation of the transmission of the load.

The conditions that the couplings have to fulfill are:

- ❖ safety in operation;
- ❖ reduced gauge dimensions;
- ❖ easy assembly and dismantling;
- ❖ to be statically and dynamically balanced;
- ❖ to ensure high durability.

Transmission of motion and torque through the gears is achieved by contacting the flanges of the gear wheels that are engaged. In the tooth interaction area there is a relative movement accompanied by friction, which leads to various forms of wear. To alleviate this phenomenon, a lubricating environment is introduced between the friction surfaces, which aims at avoiding the direct contact of the flanks, which

would lead to their grip, as well as to increase the efficiency and cooling the gear. In certain special situations, the lubricant (liquid) is also used in the hydraulic control circuit of the transmission.

2. Elasting coupling selection

The elastic coupling is both transmitting the moment from the reducer to the next working machine but is dumping vibrations and shock in some degree.

The moment for calculation is the one from shaft II

$$M_{tc} = C_s M_{tII} = 1 \cdot 1,606 \cdot 10^5 = 1,606 \cdot 10^5 \text{ Nmm} \quad (1)$$

where

M_{tc} – is the moment to be used in calculation, M_{tII} – is the moment coming from the shaft II,

$$M_{tII} = 1,606 \cdot 10^5 \text{ Nmm}$$

$C_s=1$ is the service coefficient.

The flange elastic coupling is given in two versions, but we'll consider the normal version N.

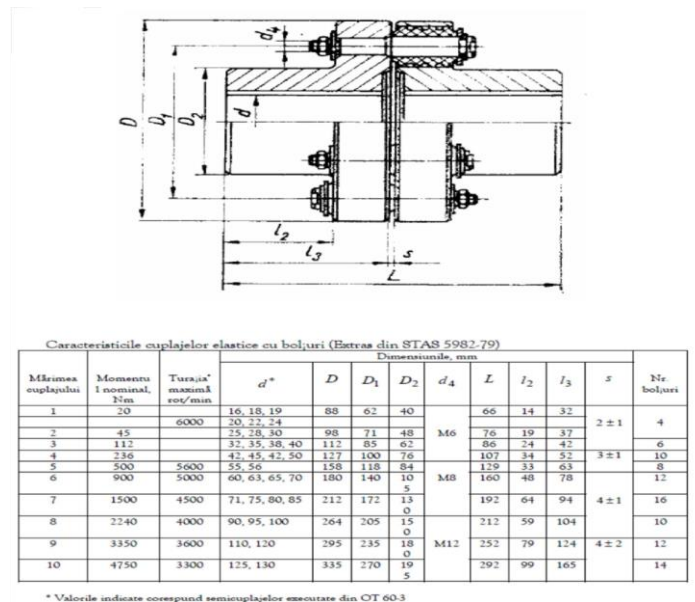


Figure 1

3. Lubrificatiting system selection

The lubricating system will be selected as function of the kinematic and loading parameters of the reducer, the type of gears and their materials. For us are important the speed of the shaft with the biggest moment to transmit.

The speed of gears at their pitch diameters are:

$$v_1 = \frac{\pi d_{w1} n_1}{60.000} = \frac{\pi \cdot 77,830 \cdot 2.464}{60.000} = 10,041 [m/s] \quad (2)$$

$$v_2 = \frac{\pi d_{w2} n_2}{60.000} = \frac{\pi \cdot 172,170 \cdot 1.100}{60.000} = 9,916 [m/s] \quad (3)$$

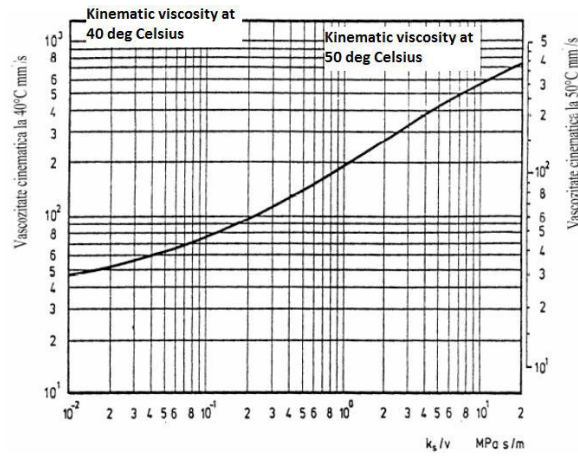


Figure 2
Kinematic viscosity of reducer oils

Where d_{w1} is the pitch diameter for the pinion and n_1 is its speed, rot/min.
Some recommendations are made for lubricants:

- If $v = (0...0,4)\text{m}^2/\text{s}$ one may use graphite as lubricant
- If $v = (0...0,8)\text{m}^2/\text{s}$ grease
- If $v = (0,8...4)\text{m}^2/\text{s}$ grease or oil
- $v > 4\text{m}^2/\text{s}$ only oil (our case)

In order to select the viscosity of the oil the above chart is to be used where on Ox axis is the Ratio ks/v where ks is the Stribeck pressure:

Already calculated:

- Tangential force

$$F_{t1} = F_{t2} = \frac{M_{tpinion}}{d_1} = 932,785\text{N} \quad (4)$$

- Pinion width

$$b_1 = 65\text{mm} \quad (5)$$

- Transmission ratio

$$u = 2.24 \quad (6)$$

- Diameter d_1

$$d_1 = m_t z_1 = \frac{z_1 m_n}{\cos(\beta)} = 76,869\text{mm} \quad (7)$$

- The rolling factor

$$\left(\frac{x_1 + x_2}{z_1 + z_2} = \frac{0,6931}{106} = 0,0065 \right)$$

$$Z_H = \sqrt{\frac{\cos\beta_b}{\cos^2\alpha_t \tan\alpha_{wt}}} = 2,34 \quad (8)$$

- the contact length factor

$$\begin{aligned}\varepsilon_a &= 1,55 \\ \varepsilon_\beta &= 2,288 \\ \Rightarrow Z_\varepsilon &= 0,81\end{aligned}\quad (9)$$

⇒ **Stribeck pressure**

$$\begin{aligned}k_s &= \frac{F_{t1}}{b_1 d_1} \cdot \frac{u+1}{u} \cdot Z_H^2 Z_\varepsilon^2 = \\ &= \frac{932,785}{65 \cdot 76,869} \cdot \frac{2,24+1}{2,24} (2,34)^2 \cdot (0,81)^2 = 0,97\end{aligned}\quad (10)$$

⇒ **Ratio**

$$\frac{k_s}{v_1} = \frac{0,97}{10,041} = 0,097\quad (11)$$

From the diagram in fig. 2 and with regard to the ratio k_s/v for the 50 degree Celsius we'll have the viscosity

$$30[\text{mm}^2/\text{sec}] = 30 [\text{centistocs}].$$

From the below table we'll take the oil type TIN 25 EP

Oil symbol	Kinematic viscosity 50°C v_{50} (cSt)	Viscosity index IV	Freezing point (°C)	Fire point (°C)
TIN 25 EP	21-26	60	-25	195
TIN 42 EP	37-45	60	-25	210
TIN 55 EP	50-57,5	60	-20	220
TIN 82 EP	82-90	60	-20	230
TIN 125 EP	130-140	60	-15	235
TIN 200 EP	200-220	70	-10	240
TIN 300 EP	230-300	70	0	255

Figure 3
The oil type

The lubricating system may be:

- ▶ Immersed lubrication is used for speeds below 12 m/s and each gear has to be immersed inside the oil bath (our case).
- ▶ With forced oil circulation is for speeds above 20 m/s is using a pumping system with pipes, heat exchangers, filters etc. This is not our case.

4. Reducer thermal calculation

When in service the gears flanks are in friction on to another and the oil inside the oil bath where the gears are immersed is agitated and so heat is developing. The power leaving the reducer P_{II} is smaller than the incoming one P_I , the difference is lost by heating P_p :

$$P_p = P_I - P_{II} = P_a + P_l + P_u \quad (12)$$

Where:

P_a – the power lost due to gear friction;

P_l – lost in bearings;

P_u – lost by agitating the oil.

The total efficiency for the reducer is:

$$\begin{aligned} \eta_R &= \frac{P_{II}}{P_I} = \eta_{a12} \eta_l^2 \eta_{u1} \eta_{u2} = \\ &= 0,996 \cdot (0,99)^2 \cdot 1 \cdot 0,877 = 0,855 \end{aligned} \quad (13)$$

$$\Rightarrow P_{II} = P_I \eta_R = 15,826 \quad (14)$$

➤ For the gearing the power lost due friction is:

$$\begin{aligned} \eta_{a12} &= 1 - \frac{\pi \mu \varepsilon_a}{f \cos \beta} \left(\frac{1}{z_1} + \frac{1}{z_2} \right) = \\ &= 1 - \frac{\pi \cdot 0,1 \cdot 1,56}{5 \cdot \cos(15^\circ)} \left(\frac{1}{33} + \frac{1}{73} \right) = 0,996 \end{aligned} \quad (15)$$

Where:

$\mu = 0,1$ – friction coefficient

$\varepsilon_a = 1,56$ – engagement factor;

$\beta = 150$ – flank inclination angle

$f = 5$ – it's a factor which is 2 for straight teeth gears, (5 for helical gears and 2...3 for high speed gearing).

η_{12} – the efficiency of the gearing,

$$\eta_{12} = 0,996$$

η_l – the efficiency of a bearing pair and

$$\eta_l = 0,99$$

➤ As for the power lost due oil agitation:

$$\begin{aligned} \eta_{u1} &= 1 - \frac{P_u}{P_1} = 1 - \frac{1}{P} \left(\frac{b_1 h_1 v_1^{\frac{3}{2}}}{1,3 \cdot 10^4} \right) = \\ &= 1 - \frac{1}{30} \left(\frac{65 \cdot 0,10 \cdot 0,041^{3/2}}{1,3 \cdot 10^4} \right) = 1 \end{aligned} \quad (16)$$

$$\begin{aligned}\eta_{u2} &= 1 - \frac{P_u}{P_{II}} = 1 - \frac{1}{P} \left(\frac{b_2 h_2 v_2^{\frac{3}{2}}}{1,3 \cdot 10^4} \right) = \\ &= 1 - \frac{1}{30} \left(\frac{62,5 \cdot 15 \cdot 9,916^{\frac{3}{2}}}{1,3 \cdot 10^4} \right) = 0,877\end{aligned}\quad (17)$$

$\eta_{u1}; \eta_{u2}$ – the efficiency lost due to oil agitation.

$$\begin{aligned}\eta_{u1} &= 1 \\ \eta_{u2} &= 0,877\end{aligned}$$

Where:

b – is the gear width,

h – the immersion depth of the gear (is 3xtooth height for driven gear),

v – gear speed,

P=18,5 kW – the transmitted power.

Taking into account the thermal equilibrium equation the heat produced during service is dissipated during service in environment through the reducer housing by convection of the surrounding air and radiation.

The equation is:

$$P_p = P_c$$

Where:

Pp – lost power,

Pc – power dissipated via housing to the environment.

So that:

$$\begin{aligned}P_I - P_{II} &= KS(t - t_0) \\ \Rightarrow t &= \frac{[(P_I - P_{II}) + KS t_0]}{KS} = \frac{[3 + 0,18 \cdot 0,29 \cdot 20]}{0,18 \cdot 0,29} = \\ &= 68,7^\circ C < 90^\circ C\end{aligned}\quad (18)$$

Where:

P_{II} =15,826 kW – is the reducer power;

K – global heat exchange between housing-environment, (K = 0.12...0.18 kW/(m² grd) for normal service reducers).

S =0,29 m² - the reducer housing surface.

t₀ =20 0 C the environment temperature

Having the preliminary sketch of the housing we have:

Length = 0,348 m,

Width = 0,135 m and

Height = 0,181 m

we'll have the surface

$$S=2(0,384 \times 0,135 + 0,384 \times 0,181 + 0,135 \times 0,181) = 0,29 \text{ m}^2$$

The allowable reducer temperature is 80°C - 90°C. Above that the oil properties are worsening.

5. MathCad software application for designing

The calculation program MATHCAD initial design data is entered. Enter formulas and standards chosen dates thereafter. The program calculates and generates results.

MATHCAD application program looks like this:

Introduce your data in the yellow fields

Lubrificant selection

$$\text{raport} = \frac{ks}{v1}$$

VARIABLE

$$\text{dw1} := 82.53$$

$$\text{dw2} := 167$$

$$\text{n1} := 2464$$

$$\text{n2} := 1100$$

FORMULAS

$$v1 = \frac{\pi \cdot \text{dw1} \cdot \text{n1}}{60000}$$

$$v2 = \frac{\pi \cdot \text{dw2} \cdot \text{n2}}{60000}$$

REZULTS

$$v1 = 10.041$$

$$v2 = 9.916$$

$$\text{Ft1} := 932.785$$

$$\text{b1} := 65$$

$$\text{b2} := 62.5$$

$$\text{u} := 2.24$$

$$\text{ZH} := 2.34$$

$$\text{Z}\epsilon := 0.81$$

$$\text{d1} := 76.869$$

$$ks = \frac{\text{Ft1}}{\text{b1} \cdot \text{d1}} \cdot \frac{u + 1}{u} \cdot \text{ZH}^2 \cdot \text{Z}\epsilon^2$$

$$ks = 0.97$$

$$\text{raport} = 0.097$$

Termal calculation

$$\mu := 0.1$$

$$\epsilon\alpha := 1.56$$

$$\beta_{\text{grade}} := 15$$

$$\beta = \frac{\beta_{\text{grade}} \cdot \pi}{180}$$

$$\beta = 0.262$$

$$\text{f} := 5$$

$$\text{z1} := 33$$

$$\text{z2} := 73$$

$$\eta_{a12} = 1 - \left(\frac{\pi \cdot \mu \cdot \epsilon\alpha}{f \cdot \cos\beta} \right) \left(\frac{1}{z1} + \frac{1}{z2} \right)$$

$$\eta_{a12} = 0.096$$

$$\text{PI} := 30$$

$$\text{h1} := 0$$

$$\text{h2} := 0.877$$

$$\eta_{u1} = 1 - \frac{1}{PI} \left[\frac{b1 \cdot h1 \cdot (v1)^{\frac{3}{2}}}{13000} \right]$$

$$\eta_{u2} = 1 - \frac{1}{PI} \left[\frac{b2 \cdot h2 \cdot (v2)^{\frac{3}{2}}}{13000} \right]$$

$$PII = PI \cdot \eta_R$$

$$\eta_R = 0.855$$

$$PII = 15.826$$

$$S := 0.29$$

$$K_{kw} := 0.18$$

$$t_0 := 20$$

$$\eta_{u1} = 1$$

$$\eta_{a2} = 0.877$$

$$\eta_l := 0.99$$

$$t_{reductor} := \frac{(PI - PII) + K_{kw} \cdot S \cdot t_0}{K_{kw} \cdot S}$$

$$\eta_R = \eta_{a12} \cdot (\eta_l)^2 \cdot \eta_{u1} \cdot \eta_{u2}$$

$$t_{reductor} = 68.756$$

6. Conclusions

Designing and making a machine is an extremely complex process that builds on the accumulation of knowledge and requires great experience gained from previous achievements.

In order to achieve the final product, a series of intermediate steps are required that are closely related to each other.

By introducing the Mathcad software in design of machine is made interactive and easier data processing. In application created can be modified easily input data to obtain more rapid project results. The program calculates and generates results.

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