ANNALS OF FACULTY ENGINEERING HUNEDOARA
- INTERNATIONAL JOURNAL OF ENGINEERING
Tome IX (Year 2011). Fascicule 1. (ISSN 1584 - 2665)

[1.]Siniša KUZMANOVIĆ, [2.]Milan RACKOV

THE INFLUENCE OF AXIAL LOAD AT OUTPUT SHAFT OF UNIVERSAL WORM AND HELICAL-WORM GEAR UNITS ON THEIR THERMAL POWER CAPACITY

[1, 2] UNIVERSITY OF NOVI SAD, FACULTY OF TECHNICAL SCIENCES, SERBIA

ABSTRACT: Universal worm and helical-worm gear units are among the mechanisms that operate with a relatively low level of efficiency for which their thermal power capacity is paid extremely high attention. Value of thermal power limit for gearboxes with free input shaft is particularly defined in the catalogue, enabling their correct choice, i.e. enables the timely assessment of the needs of taking certain procedures in order to overcome problems that may arise due to excessive heating of reducers. Thermal power capacity of motor gear reducer is taken into account when defining a range, i.e. when combining (connecting) the motor and gearbox which is made according to the catalogue, so that the problem is not noticeable to the customer. Today, in an era of tough competition, it is necessary to consider the impact of external overhung and axial loads applied to output shaft on the thermal power capacity of gearbox, so that it could be eventually taken into account when gear reducer is selected. This paper deals with the problem of reducing of thermal power capacity of gearboxes due to external loads of output shaft, i.e. it deals with additional heating due to increased power loss in the bearings. At the end it is concluded (as expected) that the effect of those loads is negligible and there is no need to take them into account when selecting the gear unit, because it does not achieve any effect.
KEYWORDS: worm and helical-worm gear units, thermal power capacity

INTRODUCTION

When choosing a universal worm and helical-worm gear reducer, service factor is selected from the catalogs of almost all manufacturers of gearboxes according to the service nature (uniform, medium and heavy), operating time during a day (0 to 24 hours), starting frequency - number starting during an hour (from 0 upwards), ambient temperature, the effective operating of reducer in an hour (so called ED factor), permissive overhung and axial loads of the output shaft (and input shaft for gear reducers with solid shaft) and thermal power capacity, accounting that the electric motor drives gearbox. However, when the large overhung and axial loads are applied, in this case only axial load on the output shaft, it comes to additional heat generating of gear reducer and thus reducing its thermal capacity. This can cause excessive heating of gear unit (usually above 80°C, or even 100°C), which may, mainly due to changes in size, have bad influence on their operating. Therefore, in this paper it is necessary to consider the influence of external axial load on the thermal capacity of the worm and helical-worm gear unit and, perhaps, suggest ways to not occur problems due to excessive heating of gearbox.

THE AIM OF THE STUDY

The main objective of this paper is to point out the importance of thermal capacity of worm and helical-worm gear units, as well as the influence of the external axial load on the output shaft to the value of this capacity.

PROBLEM INTERPRETATIONS

Universal worm and helical-warm gear units can be delivered with motor or with free input shaft. If they are delivered with electric motors, they can be delivered with special motors, so called geared motors, or with standard (IEC) motors. What electric motors will be used depends on the attitude of the manufacturers company as well as specific demands of the customer [1]. If gear units are delivered with free input shaft, they can have usual solid input shaft and with IEC motors interface.

Large manufacturers usually use special motors, which are characterized by special flanges, special diameters of output shafts, stronger bearings and better sealing solution, so they have a number of advantages (easier, cheaper and more compact design, the possibility of achieving higher gear ratios, greater permitted force of the motor shaft and better tightness). Since they are buying large quantities of such motors, they get them quickly and at almost a price of standard motors, so that this procedure is completely payable to manufacturer. In addition, these manufacturers usually have their own factory of electric motors, so that they do not have practically this problem.

Small and medium manufacturers of gear units usually use standard IEC motors, although it is not the rule, mainly because of lower cost and short delivery time, and all the benefits of special motors they try to compensate by suitable way of installing motor to the gear unit. Since it is difficult to make up a lot of advantages of special motor, in practice there are different construction solutions of installing gear unit with standard IEC motors that are directly, or with IEC motors interface, connected for the housing of gear unit.

Gear units with standard IEC motors are delivered by large manufacturers, who use special geared motors, especially when customers require. For example, when customer wants to install motors on purchased gear units by himself. It is usually case when they think they can do cheaper or faster service of their motors, or in case of export of gear units in the country, where there are factories of electric motors, which wants with a large taxes on motors to protect their products from foreign competition, and customers are payable to buy electric motors, so they buy gear units with free input shaft motor, usually, with IEC motors interface, which allow them much easier and more secure mounting IEC motors, so that there is no possibility to install motor incorrectly.

Regardless of the type of the applied electric motor, it must consider that power losses originated in the gearbox must be delivered to the surroundings [1]:

$$P_L = P_{in}(1-\eta) = Q \le Q_o = \alpha\, A\, \Delta\Theta \tag{1}$$

where: P_L - losses in the gearbox, P_{in} - input power of gear reducer, η - efficiency of gear reducer, Q - heat flux caused by originated losses, Q_o - maximum heat flux that can be transmitted to the ambient, α – coefficient of heat transmission, A - the surface area of housing of gear reducer that can exchange heat, $\Delta\Theta$ – temperature difference,
where $\Delta\Theta = \Theta - \Theta_o$,
where: Θ - the temperature of surface of reducer housing, usually it is considered that maximum temperature is Θ = 80…100°C [1] and Θ_o - temperature of ambient where the gearbox operates.

From equation (1) it follows that the value of the thermal power capacity (P_Q) is [3]. This means that thermal power limit is the greatest power in the input at which, in a permanent operating, obtained losses in the gearbox can be transferred around without excessive heating of gear reducer (Fig.1).

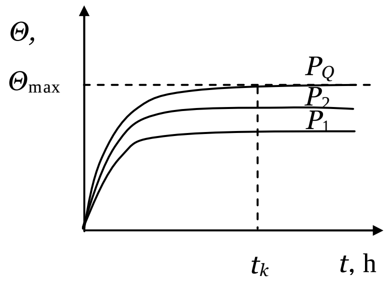

Figure 1. Graphical review of heating gear reducers depending on the input power (where t - time of heating, t_k - critical time when maximum permissive temperature is achieved (Θ_{max}), Θ - temperature of the gearbox housing, P - input power of gearbox)

$$P_{in} \le P_Q = \frac{\alpha A \Delta\Theta}{1-\eta} \tag{2}$$

It should take into account that the speed of heating depends exclusively on operating regime, input power, thermal inertia of gear reducer (of its mass) and selected cooling method. So, when choosing gear, among other requirements it must be met the following condition: $P_{in} \le P_Q$.

When developing a catalog of geared motor, i.e. when the manufacturers compose the assembly of driving and gear unit (geared motor), they concern to fulfill this condition, and they account that the normal temperature of outside air is Θ_o = 20°C. When gear reducer operates at higher ambient temperatures, the value of thermal power capacity is corrected by special coefficient.

However, when selecting gearbox with free input shaft, the customers are referred by manufacturers (in their catalogs) to detailed procedure of gearbox selection, so that problem must be considered by customers (designers) that make procedure for gearbox installation, in order to avoid possible accidents that may occur in the overheated gearboxes.

❖ DESCRIPTION OF WAYS OF SOLVING PROBLEMS

When selecting gear reducer with free input shaft, it must be provided:

$$T_N \ge T_{out} f_B \tag{3}$$

where: T_N - nominal torque, T_{out} - output torque, f_B - minimum value of the service factor.

When selecting motor gear reducer it is indirectly defined by service factors [2, 4, 5]:

$$f_{B\mathrm{perm}} \ge f_B \tag{4}$$

where: f_{Bperm} – permissive value of service factors given in the catalogs for each motor power, speed and size of reducer (it is determined by expression $f_{Bperm} = T_N / T_{out}$),

f_B – service factor defined according to the type of loading, operating time in hours during a day, the number of cycles during an hour, the ambient temperature, the effective operative duration during an hour and, eventually, the desired life of gearbox.

So, the choice of gear reducer with solid input shaft is based on its load torque (T_N), or motor service factor (equat.5), as well as the permissive values of radial ($F_{Ri\ perm}$) and axial ($F_{Ai\ perm}$) loads of the free input shaft of gearbox (for gearboxes with solid input shaft) and radial ($F_{Ro\ perm}$) and axial ($F_{Ao\ perm}$) loads of the output shaft (for both types of gear units), Fig.2.

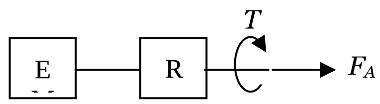

Figure 2. Schematic review of a loading of the output shaft of geared motor (EM – electric motor, R – reducer, F_A – axial load)

Additionally, the choice of gear reducer is also based on thermal capacity (equation 2), where it should take into account that thermal capacity depends on the ambient temperature, as well as on the size (and sometimes the shape and position of mounting) of gear reducer. Its values can be obtained as a table or a diagram (Fig.3).

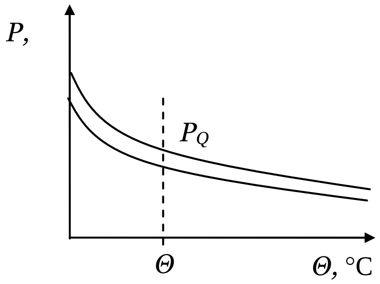

Figure 3. Graphic display of thermal capacity for particular size of gear reducer

Thermal capacity is a little different for geared motor and gear reducer with solid shaft (with classic input or with input for IEC motors), because the fan of electric motor of geared reducer provides some greater air circulation and thus better cooling of reducer, while, due to the heating of electric motor, gearbox is subjected to somewhat larger heating from the motor. Sometimes these cooling and heating quantities can be canceled, and sometimes unfortunately not, so they should be separately shown in the diagram (or table).

In order to reduce the production cost of electric motors, it is going on a maximum reduction of material consumption, which causes faster heating of the motor, so that today insulation material of class F is installed in motors (which allow their heating up to 150°C). Of course, the fan of electric motor does not allow reaching this temperature, but certainly because of higher temperatures of motor it comes to a stronger heating of gear unit, especially if the motor has bigger number of starting during an hour, and particularly if it is a motor with a brake which additionally heats the reducer.

Manufacturers of gear reducers are aware of this problem and take into account the thermal capacity of their gearboxes and try to increase it. They usually manage this by increasing the surface area of housing (i.e. by placing ribs on the surface of housing of gear unit), or by increasing the coefficient of heat transmission by defining of such forms of housing that will provide better air circulation around it, which is driven by a fan of electric motor (this is only applied for geared motor), or by placing a special fan (by manufacturers) on the worm shaft of worm and helical-worm gear reducer.

Operating regime of gear units has also major impact on their thermal capacity. For example, service nature, operating time and number of starting can strongly affect on the heating and thus the thermal capacity of gear reducer. Especially different combinations of these parameters can strongly affect on the heating which is considered by service factor. The calculation of their actual impact is quite complex and can not be accurately described by mathematics, but very accurate values can be obtained by concrete measurements.

In the case that condition is not satisfied (equation 2), it is necessary to adopt a larger (stronger) gearbox, with a larger surface area participating in the exchange of heat, or it is need to use the system for cooling oil. For smaller sizes of gear reducers it is cheaper to select larger gearbox, while in medium and large size of reducer it is rational to use oil cooling system. The system consists of filter, circulating pumps, overflow and several classic valves, piping and heat exchanger with fan and electric motors.

The existence of an external axial load on the output shaft, which permissive limit values can be found in the catalogs of manufacturers of gear reducers, causes additional load of bearings of gearbox and the occurrence of additional friction in them (whose approximate value amounts $F_{\mu A} = \mu\ F_{Aperm}$ - friction in the bearing due to the external axial force) and it causes additional heating of gear unit. Since there is no additional overhung load, bearing will be subjected to maximum permissive axial force according to the catalog.

Additional losses of power in the bearing (P_L) can be calculated by the equation:

$$P_L = 1.05 \times 10^{-4}\ M\ n \qquad (5)$$

where: n – number of revolution of output shaft, min^{-1}; M – total frictional moment of bearing

Total frictional moment of bearing (M) depends on several frictional moments as follows:

$$M = M_0 + M_1 + M_2 + M_3 \qquad (6)$$

where: M_0 - load independent frictional moment, Nmm; M_1 – load-dependent frictional moment, Nmm; M_2 – axial load-dependent frictional moment, Nmm; M_3 – frictional moment of seals, Nmm

The frictional moment (M_0) is not influenced by bearing load but by the hydrodynamic losses in the lubricant and depends on the viscosity and quantity of the lubricant and also the rolling velocity. It dominates in high-speed, lightly loaded bearings and is calculated using:

$$M_0 = 10^{-7} f_0 (\nu n)^{2/3} d_m^3 \quad (7)$$

if $\nu n \geq 2000$ or using

$$M_0 = 160 \times 10^{-7} f_0 d_m^3 \quad (8)$$

if $\nu n < 2000$, where: d_m - mean diameter of bearing (for particular bearing $d_m = 0.5 (d + D) = 0.5 (30 + 72) = 51$ mm)

f_0 – a factor depending on bearing type and lubrication (for particular bearing $f_0 = 1$); ν – kinematic viscosity of the lubricant at the operating temperature, mm^2/s (for operating temperature $\Theta = 40°$C)

The load dependent frictional moment (M_1) arises from elastic deformations and partial sliding in the contacts and predominates in slowly rotating, heavily loaded bearings. It can be calculated from:

$$M_1 = f_1 P_1 d_m \quad (9)$$

where: f_1 – a factor depending on bearing type and load

for particular bearing and load: $$f_1 = (0.0006 \ldots 0.0009)\left(\frac{F_{R\,\mathrm{perm}}}{C_0}\right)^{0.55} \quad (10)$$

P_1 - the load determining the frictional moment, N, for particular bearing and load:

$$P_1 = 3 F_{A\,\mathrm{perm}} - 0.1 F_{R\,\mathrm{perm}} = 3 F_{A\,\mathrm{perm}} \; (F_{R\,\mathrm{perm}} = 0) \quad (11)$$

Frictional moment (M_2) which depends mostly on the axial load can be calculated as follows:

$$M_2 = f_2 F_{A\,\mathrm{perm}} d_m \quad (12)$$

where: f_2 – a factor depending on bearing design and lubrication (for particular bearing design and lubrication f_2 =0.006)

The frictional moment (M_3) of the seals for a sealed bearing can be estimated and for particular bearing it is calculated as M_3 = 18 Nmm. For a smaller size of gear reducer (with shaft height h = 80 mm) orientation values of frictional moments and additional losses of power in worm and helical-worm reducer are calculated and shown in Table. 1. Based on carried out calculation it follows that the additional power losses in the gearbox, with the maximum permissive axial load of the output shaft, amounts about up to 3.63%. For lower speed ratio, power loss is less, not bigger than 2%. The power loss is bigger than only overhung load subjects the output shaft, but it is not so high and many manufacturers of gear reducer completely ignore it. When making the instruction for selecting gearbox, manufacturers of gear reducers, ignore the influence of external loads on the thermal capacity of gear unit and thus considerably simplify their selection of gear reducer.

Table 1. Results of calculation of a typical worm gear reducer without a fan with shaft height 80 mm

Thermal capacity - P_Q, W	1500	920	280
Permissive axial force of output shaft - $F_{A\,\mathrm{perm}}$, N	5520	7800	7800
Speed ratio - u	5.4	26	79
Revolution number of output shaft - n, min^{-1}	259	54	18
Load independent frictional moment - M_0, Nmm	7.66	6.7	6.07
Load-dependent frictional moment - M_1, Nmm	289.14	494.14	494.14
Axial load-dependent frictional moment - M_2, Nmm	1689.12	2386.8	2386.8
Frictional moment of seals - M_3, Nmm	18	18	18
Total frictional moment of bearing - M, Nmm	2003.92	2905.63	2905.01
Additional power losses in gear reducer - P_L, W	54.5	16.47	5.49
Percent ratio of power losses - $\frac{P_L}{P_Q} \cdot 100$, %	3.63	1.79	1.96

❖ CONCLUSION

Based on the conducted analysis it can be seen that the external axial loads of the output shaft of worm and helical-worm gear reducers have a small influence on the change of thermal capacity, usually about 2%, or for higher output speed up to 3.6%. Therefore, manufacturers of gear reducers ignore it with a full right, i.e. they do not take external forces into account when selecting gearbox and do not make correction in thermal capacity. This power loss would be more important for higher transmitted power with high output speed, when this 3.6% power loss is not negligible value, but it is a very rare case.

❖ REFERENCES

[1] KUZMANOVIĆ, S., *Universal Gear Reducers with Cylindrical Gears*, University of Novi Sad, Faculty of Technical Sciences, Novi Sad, 2009. (in Serbian)

[2] Catalog SEW Eurodrive, Movimot Gearmotors, Edition 04/2004

[3] SKF General Catalogue, 2007

[4] Catalog Nord, Constant speeds, G1000/2008, Getriebebau Nord, Hamburg

[5] Catalog Siemens, Flender Gear Units, Catalog MD 20.1-2009