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JUSTIFICATION ANALYSIS OF THE APPLICATION OF CYLINDRICAL ROLLER BEARINGS WITHIN THE UNIVERSAL MOTOR HELICAL GEAR REDUCERS

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Abstract: Within the universal motor helical gear reducer, single row deep groove ball bearings are usually used, but for larger driver dimensions and heavily loaded shafts, spherical roller bearings are used. In order to achieve the maximum load capacity of the gearbox, within the same axis height, relatively large single-row ball bearings are necessary. In some cases, due to dimensions constraints, they cannot be installed in the gearbox, so many manufacturers, in certain cases, use somewhat smaller and more expensive single-row cylindrical roller bearings. Though these bearings are not specifically designed to transmit axial forces, they can bear axial loads. The idea of this paper is to point out this problem and the benefits that come from the use of cylindrical roller bearings.

Keywords: helical gear reducers, roller bearings

1. INTRODUCTION

Universal motor helical gear reducers, depending on the gear ratio, are made as single-stage, two-stage, three-stage, but also as multistage gear reducers.

Single-stage reducers are always made in the housing for single-stage units. However, there are manufacturers who, due to lower demand, do not produce them, but cover their transmission ratios with slightly more expensive two-speed gearboxes (for example company ROSSI [1]).

Two-speed gearboxes can be produced in special housing for only two-speed gear units (STOBER [2]) or in a universal housing for two- and three-speed gearboxes (SEW [3,4]) or in a single-speed housing connected with another single-speed housing (REGAL [5]).

It is indisputable that manufacturers who produce two-speed gearboxes in special housing for two-speed gear units have slightly lower production costs than those who produce two-speed gearboxes in universal housing for two- and three-speed gearboxes or in double single-speed housings.

Three-speed gearboxes can be assembled in universal housing for two- and three-stage gearboxes (SEW [3,4]) or as a combination of two- and single-stage gearbox housings (REGAL [5]). Some manufacturers have both types of three-stage gearboxes in their product range (NORD [6]). The production costs of three-stage gearboxes assembled in the housing for three-stage reducer are lower than the costs of three-stage gearboxes built by connecting two and one-stage gearbox housings.

Four-stage reducers are usually assembled in double two-speed gearbox housings; five-stage units are produced as a combination of two- and three-stage housings, etc.

Which production concept will be adopted depends on the market orientation of the gearbox manufacturer. The greatest demand for gearboxes is usually in the range of speeds from n = 20 to 50 rpm, which roughly corresponds to gear ratios in the range from i = 30 to 70, i.e. in the area of higher values of gear ratios of two-stage gearboxes and lower gear ratios of three-stage gearboxes. Until recently, it was common for single-stage gearboxes to be manufactured with a maximum gear ratio of about 6.3; two-stage with about 30, and three-stage with about 100. However, nowadays gear ratios are completely different. Single-stage gearboxes are made with a gear ratio of up to 15 [6], two-stage gearboxes over 50 [6] and three-stage gearboxes over 300 [6]. This increasing of gear ratios enabled application of simpler (lower stage) and thus cheaper gearboxes, which work with a higher degree of efficiency. Although, it should be noted that gearboxes with conceptually higher transmission values are always more expensive than gearboxes with lower values of gear ratios. It is because of the larger central distance and thus larger gears, but for those gear ratios, where a lower-stage reducer is offered instead of a multi-stage one, a certain saving is realized which compensates for the general increase in production costs. Besides increasing gear ratios, almost all manufacturers have also managed to increase the load capacity of their gearboxes, which in some cases has managed to offer significantly cheaper gearboxes. With such interventions, the leading manufacturers managed to improve their market position, although the competition soon did the same. However, the competition was not so strong because small producers, mainly due to high investment costs in casting tools, were not financially able to keep up with the interventions of large manufacturers. [7,8]

Increasing the gear ratio and load capacity of universal motor gear reducers has been achieved in a very simple way, as described further. First, the manufacturers reduced the diameter of the pinion. Instead of mounting the pinion on the electric motor shaft (or gearbox input shaft), the manufacturers started to press the pinion into the shaft of a gear motor (or gearbox input shaft). Thus, they managed to provide larger gear ratios, for the same central distance, with a slight increasing of the driven gear diameter.

The second intervention was reducing the teeth number of the pinion, and in some cases increasing the module value. The result was increasing the gear ratio and, if the module value is higher, the load capacity of the gear pair is increased. This has been achieved thanks to advances in gear manufacturing technology (mainly grinding) that allows the production of gears with a small teeth number (usually 8 or less).

The third improvement was achieved by increasing the central distances, so the diameters of the driven gears are expanded and the gear ratio is increased. The gearbox housing got sideways expansion, but it did not significantly affect the overall dimensions of the gearbox. This modification also required the opening of the slow chamber, although some manufacturers opened both chambers, usually from the top, to allow easy installation of large gears in the gearbox housing. Implementing this procedure, the gearbox housings became a bit stiffer and the higher production accuracy was achieved (greater alignment of the openings), with a somewhat more complex installation.

The fourth step was abandoning the coaxial gearbox housing. In order to input as low torque as possible in the gearbox, the output gear pair should have the highest possible gear ratio. Since the installation space is limited by the axis height of the gear unit, for example in a two-stage coaxial gear reducer, increasing the gear ratio of the output gear pair increases its central distance. Therefore, the space for installing the driven gear in the first gear pair is reduced, i.e. the driven gear in the first gear pair must be smaller. For the increased central distance, the pinion in the first gear pair must be larger in order to maintain the alignment, thus lowering the gear ratio of the first gear pair. In order to avoid this gear ratio lowering, the concept of coaxial gearboxes was abandoned. The pinion of the first gear pair remains of the same diameter, but it is closer to the driven gear, i.e. its central distance is smaller so that the input and output shafts of the gearbox are parallel and no more coaxial.

The fifth modification was the changing of forces supporting the so-called fifth gear shaft, where the driving gear of the output gear pair is located. Due to the limited installation dimensions, single-row rigid ball bearings usually cannot accept all the forces that support the bearing, so the single-row cylindrical roller bearings are installed. This solved the force supporting problem in bearing as well as the transmission of larger torques, although in principle there was doubt whether these cylindrical roller bearings will be able to reliably accept the relatively large axial forces that, in some cases, occur on them.

2. PROBLEM DESCRIPTION

Within universal motor gear reducers, output gear pairs are the most expensive components of gearboxes, so in order to reduce production costs, they are kept in both two-stage and three-stage versions. The driving gear shaft of the output pair, so-called the fifth gear shaft, was supported in three bearings at older solutions. This solution was not suitable if the gear ratio should be increased because it did not allow the installation of large gears in the high-speed chamber. Therefore, the fifth gear shaft is supported in the usual way, with two bearings. In such a solution, the second gear (for two-stage gearbox) or fourth gear (for three-stage gearbox) is overhung (located as a console), so it is necessary that the fifth gear shaft, in place of the inner support, has a relatively large diameter and thus a large bearing. It is the case for certain solutions (Figure 1) and in one of the directions of rotation, when the inner bearing has to accept the axial force. Of course, the directions of the helix angles are defined in such a way that the axial forces on the gears are overridden as much as possible. However, this requires two sets of output gear pairs, with different helix directions of the tooth flanks, for the two-stage and three-stage variants. This concept increases the cost and complicates production and maintenance. Therefore, some gearbox manufacturers have given up of two gear pairs, so they use only one output gear pair and override axial force in a two-stage variant. In that case, the

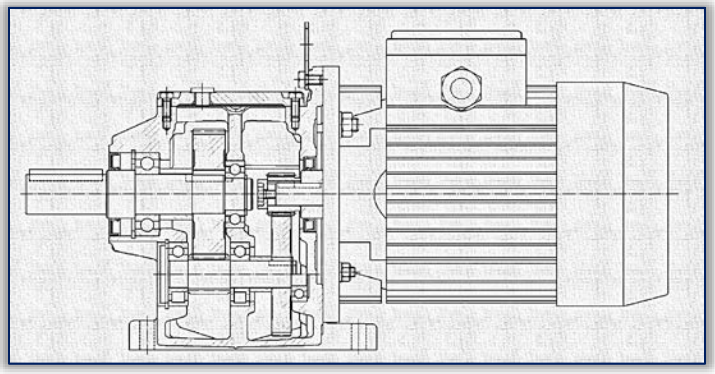

Figure 1. Characteristic solution of a two-stage universal motor gearbox with a fifth gear shaft, at higher torques (and higher axle heights), mounted with single-row cylindrical roller bearings (ROSSI [1])

bearings operate at higher speeds, while with a three-stage gearbox, they keep the same gears with the same helix direction of the output pair. In the three-stage variant, this means that axial forces are summed up and the bearings are loaded much harder. This is the necessary reason to install stronger (larger) bearings, but due to space limitations, this is usually not possible. Thus, many manufacturers started to install single-row cylindrical roller bearings on the fifth gear shaft (Figure 1).

Since the housing supports for both bearings are drilled in the same pass, absolute alignment of the openings is ensured so that there is no problem with the installation of cylindrical roller bearings. The only problem is the stiffness of the shaft and the real possibility of reliable transmission of relatively large axial forces that, in some cases, can support roller bearings.

It should be emphasized that the first gear pair in a two-stage and three-stage gearbox is used from a single-stage reducer (in three-speed gearboxes from the first smaller gearbox size). Gear ratios are usually applied according to the standard row R20, so only one output gear pair is necessary. In the case of two-stage gearboxes, due to their higher load capacity, two output gear pairs are used in order to cover a part of the gear ratio of a single-stage gearbox. If the gear ratio of single-stage gear reducers is given according to the row R10, due to the reduction of production costs, then it is necessary to use two output gear pairs, with different gear ratios, in order to achieve the gear ratio for two-stage and three-stage gear unit with the standard row R20.

If the forces of the fifth gear shaft are considered, the reactions in the supports are obtained as quite large and it is necessary to provide a quite large (strong) bearing to satisfy operating life of 10.000 hours (corresponding to a gearbox life of 5 years, with 50 working weeks per year, working in one shift with a 40-hours working week [9]).

In the first calculation, a two-stage gear reducer with older concepts will be considered. It is assembled in a universal housing for two-stage and three-stage gearboxes, in the case of both rotation directions and in case of axial forces are usually overridden (Figure 2 and 3). In the second calculation, a three-stage gear reducer will be researched, when the forces are added (Figures 4 and 5) and when axial forces are usually overridden (Figures 6 and 7). The following data are given:

≡ Axis height $h = 115$ mm
≡ Characteristically lengths of the fifth gear shaft: $a = 32$ mm, $b = 32$ mm, $c = 26$ mm i $d = 16{,}5$ mm.

Nominal torque $T_{2N} = 450$ Nm, and for the two-stage version, for the highest gear ratio, it is $T_{2N} = 360$ Nm since it cannot be loaded more.

— The highest gear ratio of the output gear pair $i_{3xmax} = 6.54$
≡ The teeth number of the fifth gear with the highest gear ratio $z_5 = 11$
≡ The module of the fifth gear with the highest gear ratio $m_{n5/6} = 1.5$ mm
≡ The helix angle of the output gear pair with the highest gear ratio $\beta_{5/6} = 12°$
≡ The direction of the helix angle of the fifth gear – left direction, in the case that axial forces are being overridden in three-stage variant – right direction.
≡ The angle of the axis of the fifth gear in the housing $\varphi_5 = 31.008°$ (Figures 2, 3, 4, 5, 6 and 7)

— The highest gear ratio of the second gear pair $i_{2xmax} = 5.36$
≡ The teeth number of the fourth gear with the highest gear ratio $z_4 = 60$
≡ The module of the fourth gear with the highest gear ratio $m_{n3/4} = 1.5$
≡ The helix angle of the second gear pair with the highest gear ratio $\beta_{3/4} = 20°$
≡ The direction of the helix angle of the fourth gear – right direction
≡ The angle of the axis of the fourth gear in the housing $\varphi_4 = 88.137°$ (Figures 4, 5, 6 and 7)

— The highest gear ratio of the first gear pair $i_{1xmax} = 7.64$ in two-stage variant
≡ The teeth number of the second gear with the highest gear ratio $z_2 = 84$
≡ The module of the second gear with the highest gear ratio $m_{n1/2} = 1$ mm
≡ The helix angle of the first gear pair with the highest gear ratio $\beta_{1/2} = 30°$
≡ The direction of the helix angle of the second gear – left direction
≡ The angle of the axis of the second gear in the housing $\varphi_2 = 37.383°$ (Figure 2 and 3)

— The highest gear ratio of the first gear pair $i_{1xmax} = 6.36$ in three-stage variant
≡ The teeth number of the second gear with the highest gear ratio $z_2 = 70$
≡ The module of the second gear with the highest gear ratio $m_{n1/2} = 1$ mm
≡ The helix angle of the first gear pair with the highest gear ratio $\beta_{1/2} = 30°$
≡ The direction of the helix angle of the second gear – left direction
≡ The angle of the axis of the second gear in the housing $\varphi_2 = 27.239°$

≡ Calculating the value of the velocity rotation of electric motor $n_{em} = 1450$ min^{-1},
≡ Designation of output (A) ball bearing 6203, and roller bearing NUP2203
≡ Designation of input (B) ball bearing 6203, and roller bearing NUP2203.

First, a simplified bearing life calculation was performed [10] for the two-stage variant and the least loaded support case, using the equation:

$$L_{10h} = \frac{10^6}{60 \cdot n} \cdot \left(\frac{C}{P}\right)^p \tag{1}$$

Applied output torque is $T_{2N} = 360$ Nm (because of the limited torque of the first gear pair) and the highest gear ratio is analyzed (Table 1). However, at this reduced torque, the outer ball bearing (A) does not satisfy (Table 1).

Table 1: Results of the calculation of a two-stage gear reducer when the axial forces are partially overridden for the least loaded case $T_{2N} = T_6 = 360$ Nm and the highest gear ratios $i_{1/2} = 7.64$ and $i_{5/6} = 6.64$

Two-stage gear reducer – $T_{2N} = 360$ Nm			Axial forces are partially overridden		
Right-handed direction of rotation					
Forces in supports	$F_{Ar} = 3907$ N	$F_{Br} = 2399$ N	$F_a = 721$ N	$n_{5min} = 190$ rpm	
Designation	Bearing A	Bearing B	L_{nmh}, h	Bearing A	Bearing B
Ball bearing 6303, d = 17 mm, D = 47 mm, C = 14300 N, $C_o = 6550$ N, $f_o = 12$, $P_u = 275$ N					
Bearing A accepts axial force (1)					
L_{10h}, h	4301	18578		1935	17649
Bearing B accepts axial force (2)					
L_{10h}, h	4301	18532			
Roller bearing NUP2203 d = 17 mm, D = 40 mm, C = 23800 N, $C_o = 21600$ N, $P_u = 2650$ N					
Bearing A accepts axial force (1)					
L_{10h}, h	36004	182640		18002	146112
Bearing B accepts axial force (2)					
L_{10h}, h	35996	132787			
Left-handed direction of rotation					
Forces in supports	$F_{Ar} = 3823$ N	$F_{Br} = 2489$ N	$F_a = -721$ N	$n_{5min} = 190$ rpm	
Ball bearing 6303, d = 17 mm, D = 47 mm, C = 14300 N, $C_o = 6550$ N, $f_o = 12$, $P_u = 275$ N					
Bearing A accepts axial force (1) and (2)					
L_{10h}, h	4591	16635			
Bearing B accepts axial force					
L_{10h}, h	4591	16635			
Roller bearing NUP2203 d = 17 mm, D = 40 mm, C = 23800 N, $C_o = 21600$ N, $P_u = 2650$ N					
Bearing A accepts axial force (1) and (2)					
L_{10h}, h	38698	161560			
Bearing B accepts axial force					
L_{10h}, h	38698	119928			

Note: (1) bearing A accepts axial force regardless of its action direction and (2) bearing A accepts axial force if the force acts towards it and in case axial force acts towards bearing B then bearing B accepts axial force.

By replacing it with roller bearings, it was established that this roller bearing can carry the applied load at the highest gear ratio.

By taking a more detailed bearing calculation, according to the equation:

$$L_{nmh} = a_1 \, a_{SKF} \, L_{10h} \tag{2}$$

more favourable results are obtained, but the ball bearings still do not satisfy.

In the calculation, the value $a_1 = 1$ is used. For ISO VG 220 oil, at the expected operating temperature of $\theta = 70$°C, it follows that kinematic viscosity of the oil is $v = 55$ mm^2/s. For mean bearing diameter $d_m = (d + D) / 2 = (17 + 47) / 2 = 32$ mm and the speed velocity that occurs in this calculation $n_{5min} = 190$ rpm, it follows that $v_1 = 90$ and $K = 0.61$. For the case of mild contamination, it follows that $\eta = 0.3 - 0.5$ and it is adopted that $\eta = 0.5$. Partial results of detailed calculation are given in Table 1.

Based on the detailed bearing calculation, it can be assumed that when using roller bearings, with lower gear ratios, it is probably not necessary to reduce the nominal load capacity of the reducer (T_{2N}).

It should be noted that in practice there are only two characteristic design solutions:

1. when bearing A accepts axial force regardless of the direction of its action and
2. when bearing A accepts axial force if it acts toward it, and in the case of axial force acts towards bearing B then this bearing accepts axial force (Tables 1 and 2).

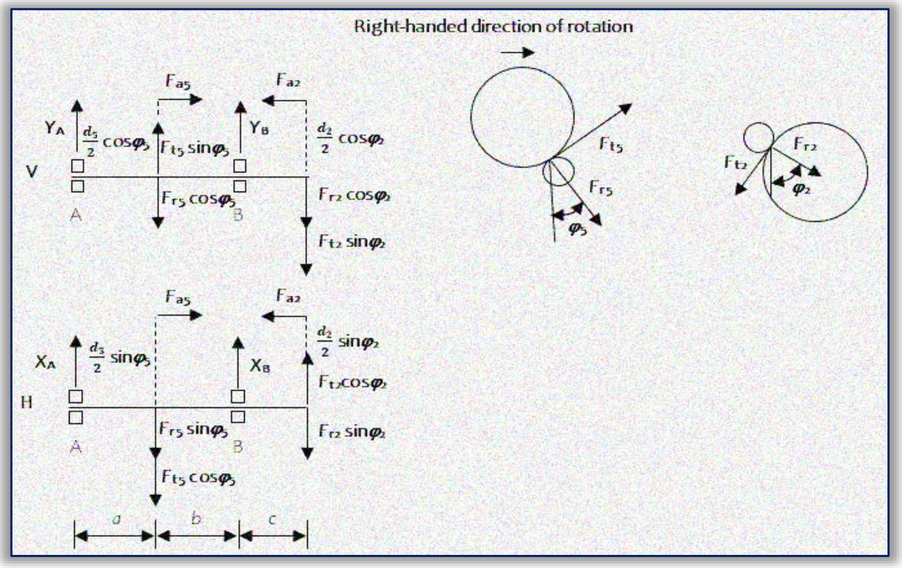

Figure 2. Analysis of forces on the fifth gear shaft for a two-stage gearbox in case the axial forces are partially overridden (right-handed direction of rotation of the output shaft)

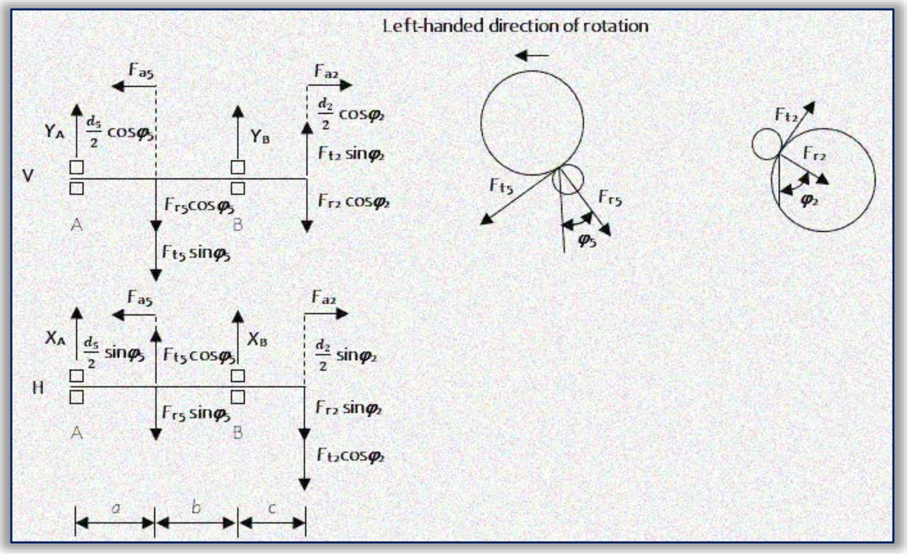

Figure 3. Analysis of forces on the fifth gear shaft for a two-stage gearbox in case the axial forces are partially overridden (left-handed direction of rotation of the output shaft)

Further calculations showed that the predicted nominal load of T_{2N} = 450 Nm (Table 2) can be partially carried by roller bearings even at the highest gear ratio, which leads to the conclusion that with increased load capacity it is completely justified to use roller bearings.

Table 2: Results of the calculation of a two-stage gear reducer for the worst case in Table 1 for the nominal loading case $T_{2N} = T_6 = 450$ Nm and the highest gear ratios $i_{1/2} = 7.64$ and $i_{5/6} = 6.64$

Two-stage gear reducer – $T_{2N} = 450$ Nm Axial forces are partially overridden				
Right-handed direction of rotation				
Forces in supports	$F_{Ar} = 4884$ N	$F_{Br} = 2999$ N	$F_a = 901$ N	$n_{5min} = 190$ rpm
Roller bearing NUP2203 d = 17 mm, D = 40 mm, C = 23800 N, $C_o = 21600$ N, $P_u = 2650$ N				
Bearing B accepts axial force (2)				
L_{10h}, h	17119	63160		

Note: (2) bearing A accepts axial force if the force acts towards it and in case axial force acts towards bearing B then bearing B accepts axial force.

A similar calculation was performed for a three-stage reducer when the axial forces on the fifth gear shaft are summed up (Figures 4 and 5) and Tables 3 and 4.

Ball bearings do not satisfy even in the three-stage variant, so only roller bearings should be used. A detailed calculation of the bearing, which is not performed here, would certainly show that the nominal loading capacity of the reducer (T_{2N}) should not be reduced at lower gear ratio values.

A similar calculation was performed for a three-stage gearbox when the axial forces on the fifth gear shaft are partially overridden (Figures 6 and 7) and Table 4.

Table 3: Results of the calculation of the three-stage gear reducer when the axial forces are summed up, i.e. they are not overridden, in case of load $T_{2N} = T_6 = 450$ Nm and the highest gear ratios $i_{1/2} = 6.36$; $i_{3/4} = 5.36$ and $i_{5/6} = 6.64$

Three-stage gear reducer – $T_{2N} = 450$ Nm			Axial forces are summed up	
Right-handed direction of rotation				
Forces in supports	$F_{Ar} = 4704$ N	$F_{Br} = 2615$ N	$F_a = 2232$ N	$n_{5min} = 43$ rpm
Ball bearing 6303, d = 17 mm, D = 47 mm, C = 14300 N, $C_o = 6550$ N, $f_o = 12$, $P_u = 275$ N				
Bearing A accepts axial force (1)				
L_{10h}, h	8484	63383		
Bearing B accepts axial force (2)				
L_{10h}, h	10889	18503		
Roller bearing NUP2203 d = 17 mm, D = 40 mm, C = 23800 N, $C_o = 21600$ N, $P_u = 2650$ N				
Bearing A accepts axial force (1)				
L_{10h}, h	46101	605621		
Bearing B accepts axial force (2)				
L_{10h}, h	85717	183142		
Left-handed direction of rotation				
Forces in supports	$F_{Ar} = 5858$ N	$F_{Br} = 3462$ N	$F_a = -2232$ N	$n_{5min} = 43$ rpm
Ball bearing 6303, d = 17 mm, D = 47 mm, C = 14300 N, $C_o = 6550$ N, $f_o = 12$, $P_u = 275$ N				
Bearing A accepts axial force (1) and (2)				
L_{10h}, h	5638	27315		
Bearing B accepts axial force				
L_{10h}, h	5638	13161		
Roller bearing NUP2203 d = 17 mm, D = 40 mm, C = 23800 N, $C_o = 21600$ N, $P_u = 2650$ N				
Bearing A accepts axial force (1) and (2)				
L_{10h}, h	26020	237915		
Bearing B accepts axial force				
L_{10h}, h	41283	97609		

Note: (1) bearing A accepts axial force regardless of its action direction and (2) bearing A accepts axial force if the force acts towards it and in case axial force acts towards bearing B then bearing B accepts axial force.

Table 4: Results of the calculation of the three-stage gear reducer when the axial forces are overridden in the case of load $T_{2N} = T_6 = 450$ Nm and the highest gear ratios $i_{1/2} = 6.36$; $i_{3/4} = 5.36$ and $i_{5/6} = 6.64$

Three-stage gear reducer – $T_{2N} = 450$ Nm			Axial forces are overridden	
Right-handed direction of rotation				
Forces in supports	$F_{Ar} = 4819$ N	$F_{Br} = 2330$ N	$F_a = -1184$ N	$n_{5min} = 43$ rpm
Ball bearing 6303, d = 17 mm, D = 47 mm, C = 14300 N, $C_o = 6550$ N, $f_o = 12$, $P_u = 275$ N				
Bearing A accepts axial force (1)				
L_{10h}, h	10128	89603		
Bearing B accepts axial force (2)				
L_{10h}, h	10128	49376		
Roller bearing NUP2203 d = 17 mm, D = 40 mm, C = 23800 N, $C_o = 21600$ N, $P_u = 2650$ N				
Bearing A accepts axial force (1)				
L_{10h}, h	63643	889378		
Bearing B accepts axial force (2)				
L_{10h}, h	79092	452606		
Left-handed direction of rotation				
Forces in supports	$F_{Ar} = 5801$ N	$F_{Br} = 3884$ N	$F_a = 1184$ N	$n_{5min} = 43$ rpm
Ball bearing 6303, d = 17 mm, D = 47 mm, C = 14300 N, $C_o = 6550$ N, $f_o = 12$, $P_u = 275$ N				
Bearing A accepts axial force (1) and (2)				
L_{10h}, h	5806	19344		
Bearing B accepts axial force				
L_{10h}, h	5806	19344		
Roller bearing NUP2203 d = 17 mm, D = 40 mm, C = 23800 N, $C_o = 21600$ N, $P_u = 2650$ N				
Bearing A accepts axial force (1) and (2)				
L_{10h}, h	37141	162212		
Bearing B accepts axial force				
L_{10h}, h	42650	117037		

Note: (1) bearing A accepts axial force regardless of its action direction and (2) bearing A accepts axial force if the force acts towards it and in case axial force acts towards bearing B then bearing B accepts axial force.

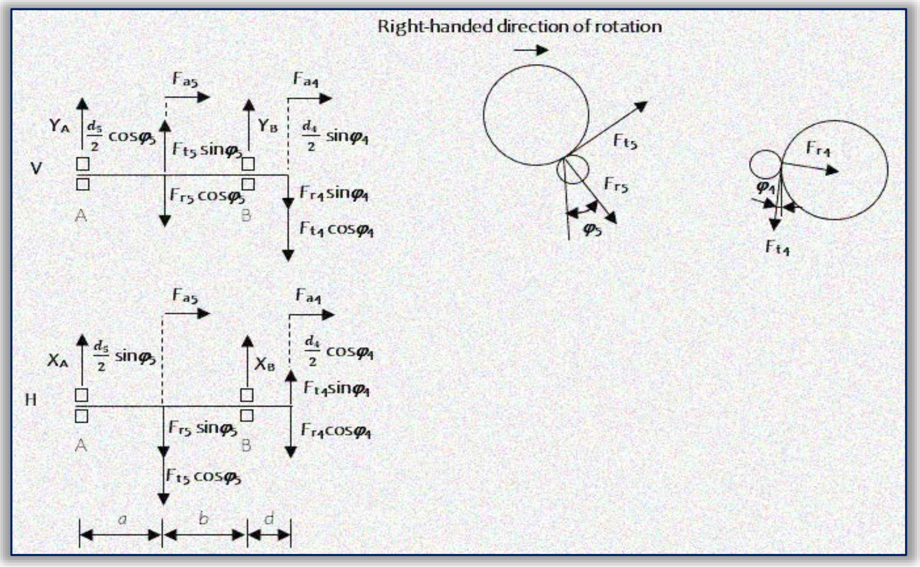

Figure 4. Analysis of the forces on the fifth gear shaft for a three-stage gearbox in case the axial forces are summed up, i.e. with one set of output gears (right-handed direction of rotation of the output shaft)

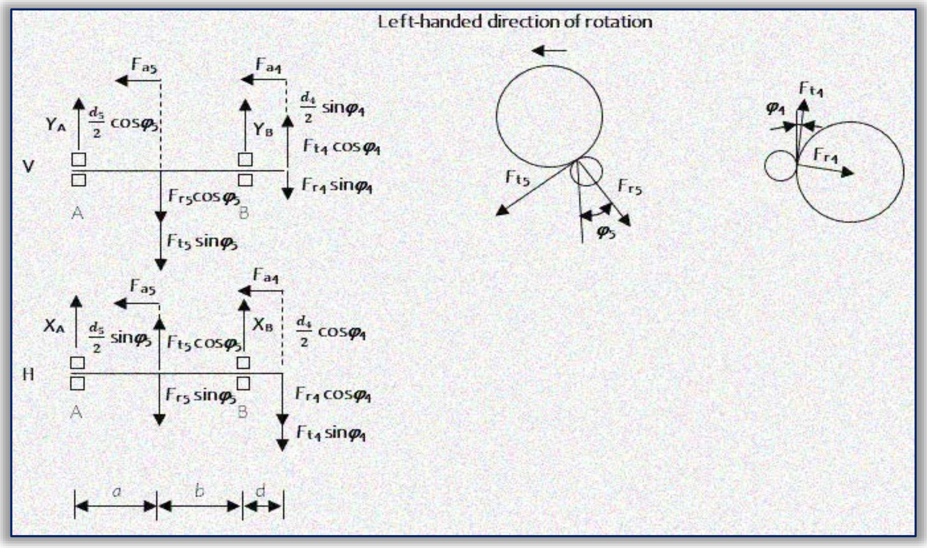

Figure 5. Analysis of the forces on the fifth gear shaft for a three-stage gearbox in case the axial forces are summed up, i.e. with one set of output gears (left-handed direction of rotation of the output shaft)

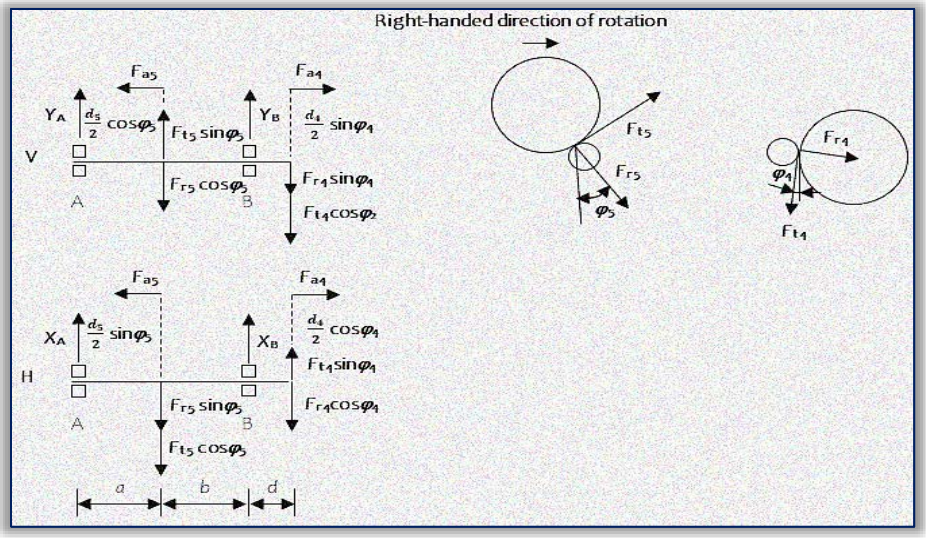

Figure 6. Analysis of forces on the fifth gear shaft for a three-stage gearbox in case the axial forces are overridden, i.e. there are two sets of output gears (right-handed direction of rotation of the output shaft)

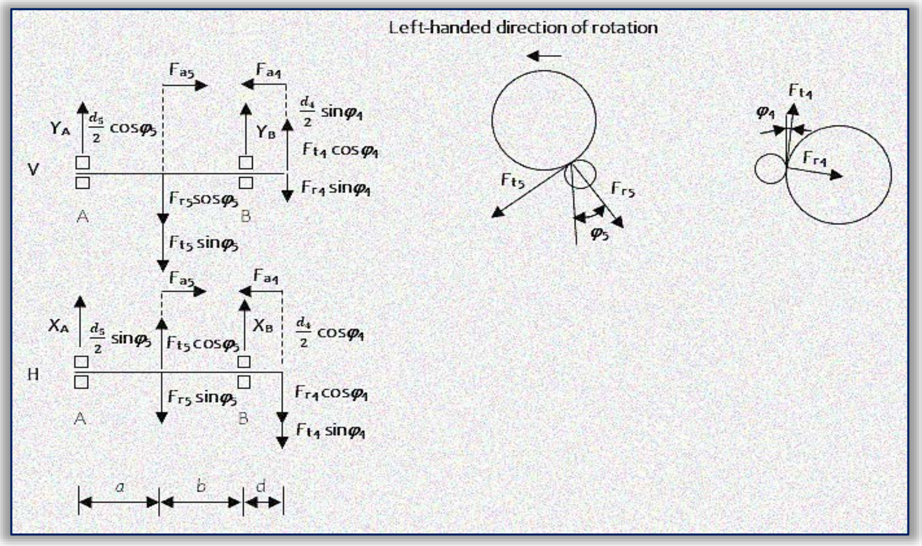

Figure 7. Analysis of forces on the fifth gear shaft for a three-stage gearbox in case the axial forces are overridden, i.e. there are two sets of output gears (left-handed direction of rotation of the output shaft)

Ball bearings also do not satisfy in this case either, so only roller bearings should be used.

With the simplified calculation method, it comes out that roller bearings almost satisfy the operating life in every case. A detailed calculation of the bearing, which is not performed here, would certainly show that the nominal loading capacity of the reducer (T_{2N}) should not be reduced at lower gear ratio values.

3. CONCLUSIONS

Based on the given calculation and analysis of the obtained results, it is concluded that with the increase of the gearbox loading capacity, as is expected, it is necessary to use stronger bearings. Most of the gear reducer manufacturers decided to use cylindrical roller bearings, due to maintaining the same installation dimensions. Since stronger bearings are used, there is a question of whether they could carry a larger axial force, which would result in using only one set of output gear pairs.

Some gearbox manufacturers use two sets of output gears with different helical teeth directions in order to override axial forces on the gears of the fifth gear shaft. Based on the calculations, it follows that the difference in the service life of critical bearings, where axial forces are overridden or summed up, is relatively large - approx. 40%. Since the bearing life for both rotation directions is usually above the projected 10.000 hours, it follows that the production of gearboxes with only one set of output gear pairs is the best option. In this case, production costs of gear reducer manufacturing are much lower, especially since the output gear pair is the largest gear in the gear unit.

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