7th International Quality Conference
May 24th 2013
Center for Quality, Faculty of Engineering, University of Kragujevac

Milos Matejic[1)]
Mirko Blagojevic[1)]
Zorica Djorđevic[1)]
Nenad Marjanovic[1)]
Nenad Petrovic[1)]

1) Faculty of Mechanical Engineering, University of Kragujevac, Serbia {mmatejic,mirkob, zoricadj, nesam, npetrovic}@kg.ac.rs

COMPARATIVE ANALYSIS OF DIFERENT TYPE REDUCERS FOR WINCH DRUM DRIVING UNIT

Abstract: *Thanks to modern technological advances, today is developed so many of different types of mechanical transmissions. They are differences in performance, dimensions, mass, efficiency ratio, and of course price. In the framework of this paper is to set the output parameters (puling force, rotations per minute) the concept of the ship's winch drum are developed and elaborated. Performed by calculation of its operating parameters (power and rotations per minute of electric motors, power transmission and gear ratio) its three-dimensional model has been made. Then is analyzed the possibility of implementing different types of reducers due to the fact that there is a large difference in the torque, rpm and efficiency ratio in relation to the mass and overall dimensions. The paper also presents the results of mutual comparisons given reducers as well as guidelines for the selection of optimal solutions.*

Keywords:*winch, drive unit, drum, reducer, comparative analysis*

1. INTRODUCTION

Thanks to modern technological advances, many different types of mechanical transmissions are developed today. Nearly all mechanical systems contain at least one or more types of mechanical transmission. Among the mechanical powered, transmission gearboxes are the most widely used. During the development of the industry, many types of reducers have appeared: coaxial reducers, reducers with conical gears, worm reducers, planetary reducers, cycloid drives, harmonic drives, etc.

Conventional reducers include gearboxes with parallel shaft axes and with cylindrical gears, which are single stage or multistage reducers, [1, 2]. Reducers with conical gears are used when the input and output shaft axis need to be intersecting [3]. Quite often, reducers, which are a combination of cylindrical and bevel gear pairs are used. Cylindrical and conical gear reducers are characterized by a very high level of efficiency, while their overall dimensions and mass are greater than other types of reducers.

Worm reducers are usually made in a single stage or in two stages. They are used in situations where the input and output axis need to bypass [4]. Worm gear reducers are characterized by smaller overall dimensions and mass compared to conventional gear reducers, and have a lower level of efficiency. The greatest advantage of worm gears compared to their conventional counterparts is a much higher transmission ratio. In practice, different types of gear pairs, cylindrical, conical, and worm, are combined in one reducer housing. And this type of reducer was and still is the subject of many studies.

Planetary gearboxes are classified

under special types of reducers. Their work is based on the planetary movements of spur gears. They are derived in several concepts as single stage and multistage reducers. Small overall dimensions and mass, and a high efficiency ratio, and high transmission ratio characterize them. The most attention in studies of these reducers is given to dynamic analysis [5, 6].

Cycloid drives are also special types of gearboxes whose operation is based on the rotational motion of cycloid gears, which drives an eccentric shaft. Very small overall dimensions and mass, and a high efficiency ratio characterize these transmissions. They are also derived as single-stage or multi-stage reducers. For this type of reducer, research trends include dynamic behavior as well as efficiency levels [7, 8].

Harmonic drives are the newest type of mechanical transmission [9]. Their operation is based on the rotation of the central flexible gear, which has one tooth less than the fixed central gear. They are attributed for having a very high transmission ratio, and a lower level of efficiency in comparison to other mechanical transmissions.

All these reducers are different in performance, size, mass, level of efficiency, and of course price.

In the guidelines of this paper the calculation and design of a drum for a ship winch is given. An analysis of the transmission choice has been performed in order to select the optimal solution.

2. DRUM DESIGN OF THE SHIP WINCH

For the design of the ship's winch drum, input parameters are primarily mass of the cargo and the velocity of the pulling cargo. In addition to these two basic parameters, some more baseline characteristics are defined [10]. Input parameters for the calculation and construction of the ship's winch drum in this paper are given in Table 1.

Table 1. Input parameters

Mass of winching cargo m_C	250[kg]
Pulling speed (in the highest layer) V_W	100[m/min]
The highest layer drum diameter D_{HL}	600[mm]
Drum pipe length L	500[mm]
Wire diameter d_W	12[mm]
Wire length L_W	500[m]
Multi-layer spooling	5÷20

After defining the input parameters, the next step is the ship's winch drum calculation. Calculation of ship's winch drum consists of previous power calculations of the winch drum, the previous calculation of the drum body, the final power and angular speed calculation of the drum and final calculation of the winch drum dimensions.

2.1 Previous power calculation of the winch drum

In the previous calculation of the winch drum, winch power is determined based on mass and velocity of the cargo (pulling or lifting). Determination of power P_D is based on the expression (2.1).

$$P_D = \frac{M_D \cdot \omega_D}{\eta_B^2} \tag{2.1}$$

Given are:

η_B –efficiency ratio in the drum bearings,

ω_D – angular speed of the drum, [s^{-1}].

Pulling torque on the drum M_D is calculated according to (2.2), in function of the radius of the highest layer wound r_{HL} and pulling force in the rope F_W.

$$M_D = r_{HL} \cdot F_W \tag{2.2}$$

Pulling force in the rope and drum angular velocity is calculated using the following expression:

$$F_W = m_C \cdot g \tag{2.3}$$

$$\omega_D = \frac{\nu_D}{r_{HL}} \tag{2.4}$$

g–gravity acceleration [m/s²].

The presented expression leads to values for the power on a particular drum P_D=4, 16[kW].

2.2 Previous calculation of the drum body

The previous calculation requires determining of the drum body outer pipe diameter and number of rope layers, which is wrapped in a function of: the required length of the drum body, the required diameter of the wrapped rope top layer as well as the length and diameter of the rope which is wrapped. In this paper, an algorithm has been developed to determine the length of the wrapped rope on the drum with specified parameters (length of the drum pipe, the outer diameter of the drum pipe, number of wrapping layers, and the diameter of the rope). With this algorithm in a few iterations it is very easy determine the outer pipe diameter and number of wrapping layers.

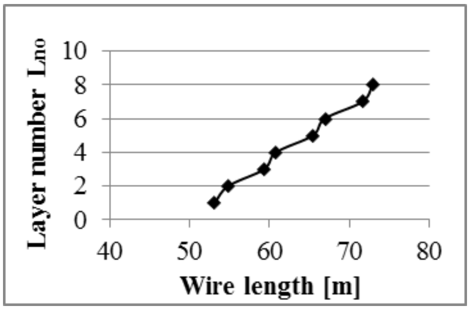

Figure 1. Spooling rope in layers

When choosing the number of wrapping layers and outer drum pipe diameter, the number of wrapped layers must be kept to a minimum, and that the diameter of the top layer should be as close as possible to the required value from the input parameters for a smaller speed variation of the pulling velocity of the cargo. The proposal of the optimal design solutions in this study has been obtained from the fourth iteration. The following parameters were obtained: the number of wrapping layers is 8, the outer drum pipe diameter of 400 [mm] and diameter of the wrapped top layer of 592 [mm].

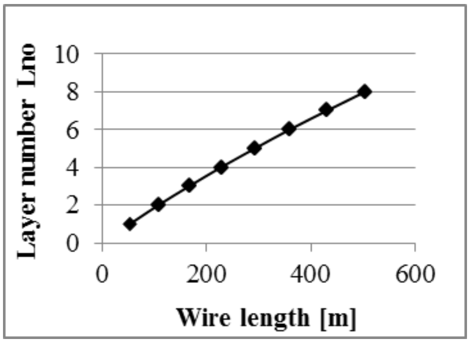

Figure 2. Spooling rope length by layer

As a result the algorithm makes the graph or rope spooling in layers (Figure 1) and the graph of spooling rope to a certain layer (Figure 2).

2.3 Final power calculation on winch drum

The final power calculation on the winch drum is the same as the previous calculation based on expressions (2.1), (2.2), (2.3) and (2.4). All sizes, which were outlined in the previous calculations of the winch drum remain the same, except for the radius of the top layer wrapped r_{HL}. For the final calculation of capacity, r_{HL}, is calculated from the expression:

$$r_{HL} = \frac{D_{HL} - d_W}{2} \tag{2.5}$$

Substituting the new values of r_{HL} in terms of the previous calculations of the parameters are obtained power P_D= 4, 18 [kW] and the drum speed n_D = 55±5 [min^{-1}], which are applicable for choosing the drum driving unit.

2.4 Final calculation of the drum pipe

In the final calculation of the drum pipe defines the minimum permissible wall thickness of the drum, which depends on the approved material for drum production, as well as the pulling force in the rope F_W. For the steel drum S355J2G3 steel was adopted, with a yield stress of R_{eH}= 355 [N/mm2]. Tube wall thickness, t_p, is determined by the expression [11].

$$t_p = \frac{F_W}{\sigma_d \cdot d_W} \cdot C \tag{2.6}$$

Figuring in the equation (2.6) are pulling force F_W, rope diameter d_W, allowable stress σ_d and wrapping constant C. Allowable stress is calculated from the expression:

$$\sigma_d = R_{eH} \cdot S \tag{2.7}$$

Security degree S, is adopted as recommended. Constant C by DNV standard [12] for a single layer wrapping is 1.0, whereas for multilayer wrapping it has a constant value of 1.75.

Substituting these values into the expression (2.6) the minimum wall thickness must be greater than 1.5 [mm]. Other values for the design of the drum are adopted according to recommendations or based on experience, while for higher loads in addition to deterministic calculations finite element analysis can be used.

2.5 Drum design of winch

After making a detailed analysis of the calculation and a list of demands, it was accessed to the design of the ship's winch drum. The designed winch drum is shown in Figure 3.

It can be seen in Figure 3 that for the design of the winch drum a lot of attention is paid to stiffening the sides of the drum and the drum sleeves. Stiffening sleeves can be made and the inside of the drum winch

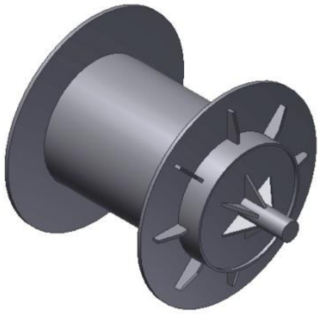

Figure 3. Winch drum

3. INSTALLATION POSSIBILITYOF DIFFERENT TYPES OF REDUCERS

In the projected winch drum can be fitted with almost any type of reducer. This heading shoes the comparative analysis of installing a two-stage bevel reducer, a worm reducer and a single planetary reducer.

3.1 Installation analysis of two-stage bevel reducer

Installation of the two-stage bevel reducer on to the designed winch drum was taken into consideration for the following reasons: compact construction, gear positioning capabilities in several positions and in several ways and the high efficiency ratio of the this particular mechanical transmission. The selected two-stage bevel reducer is one produced by Nord Drive Systems [13], labeled SK 9042.1 - 112 MH4. The characteristics of this gear are given in Table 2.

A preview of the two-stage bevel reducer installation on the winch drum is given in Figure 4.

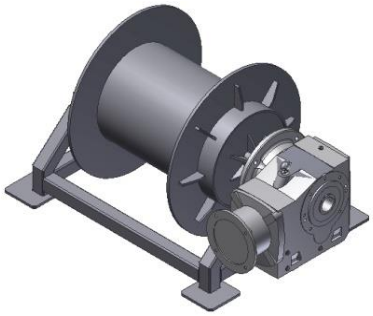

Figure 4. Installation of a two-stage bevel reducer on the winch drum

3.2 Installation analysis of worm reducer

The possibility of installing a one stage worm reducer is taken into consideration due to the small overall dimensions and mass. Selected worm reducer is produced by Techniche Antriebselemente (T.E.A), [14]. The characteristics of the reducer, labeled I 130, are given in Table 2.

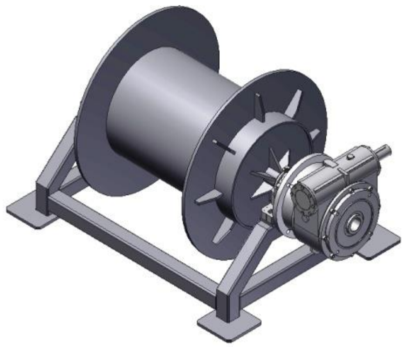

Figure 5. Installation of a single-stage worm reducer on the winch drum

This drive is dimensionally smaller than the previous one, it has a smaller mass, but it has a much lower level of efficiency. A lower level of efficiency automatically means the need for a stronger powertrain. Worm reducer can be installed on the ship's winch drum in several different positions as well as in several different types of gear unit housings. Installation of a worm reducer is shown in Figure 5.

3.3 Installation analysis of planetary reducer

The compactness of the planetary gear design allows the drive to be placed inside the drum pipe, which significantly reduces the dimensions of the winch. The C concept of planetary reducer has been adopted for the purposes of this paper, which is designed, in one of the previous works of the authors [10]. Characteristics of the planetary reducer which is considered here are given in Table 2.

Table 2. Reducer's properties

Reducer type	*1*	*2*	*3*
Output rpm of reducer n_2[min^{-1}]	52	56	60
Output torque of reducer T_2, [Nm]	740	710	730
Electric motor power P_{EM}, [kW]	4	5,27	5
Reducer ratio i_r	27,9	25	20
Reducer efficiency ratio η	0,97	0,79	0,9
Reducer mass, [kg]	110	45	89

Given are three types of reducers, and those are:

1 – two-stage bevel reducer,
2 – single-stage worm reducer,
3 – planetary reducer.

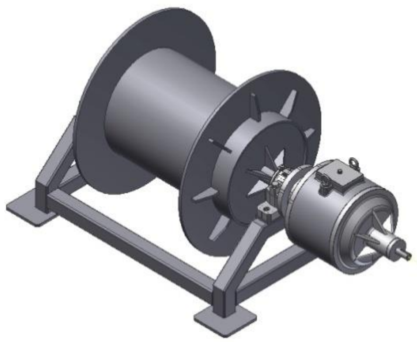

Figure 6. Installation of a planetary reducer on the winch drum

The installation of the planetary gear without inserting into the drum pipe is shown in Figure 6. Figure 6 also shows that if the drive is not inserted into the drum , the size increases significantly due to the alignment of the reducer and winch drum axis.

3.4 A comparative analysis of different types of installing reducers

All three drives meet the discussed criteria for winch drum-mounted drives. These reducers differ in power required for their operation, level of efficiency, mass and overall dimensions.

The two-stage bevel reducer has the greatest mass of all, while the worm reducer has the smallest mass (shown in Figure 7).

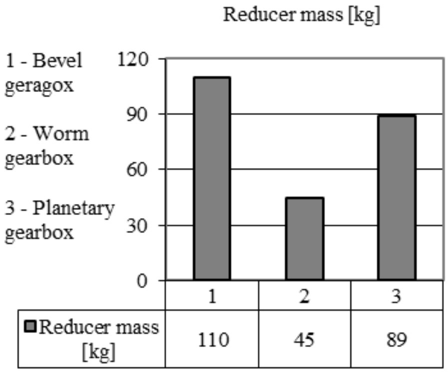

Figure 7. Comparison of drives according to mass

In addition to maximum mass, double-bevel gear have the highest moment of inertia when stopping.

Shown in Figure 8 is a comparison of reducers by their efficiency ratio. The two-stage bevel reducer has the highest level of efficiency, while worm reducer has the smallest.

Figure 9 shows a comparison of the reducers relative to the driving power. The most power is needed to drive the worm reducer, and the least for the two-stage bevel reducer.

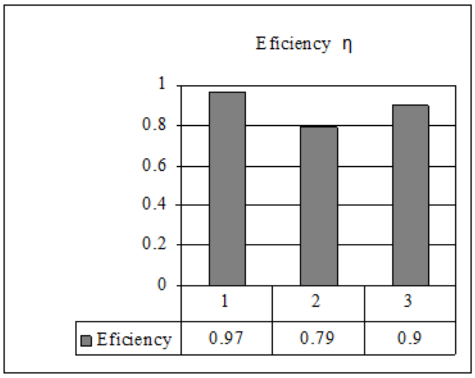

Figure 8. Comparison of drives according to their efficiency ratio

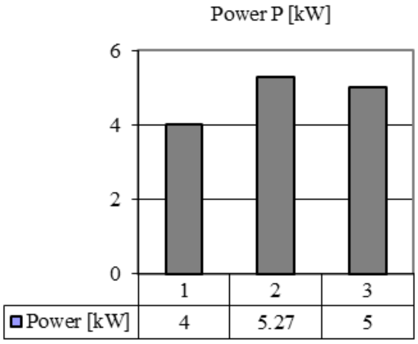

Figure 9. Comparison of drives according to power needed for operation

4. CONCLUSION

By analyzing the characteristics of the given three reducers, it can be concluded that the optimal solution must be found in the additional demands that would be placed before completing the design of the winch.

A two-stage bevel reducer has the highest efficiency ratio and requires the least power, so from that aspect it is the optimal solution. From the standpoint of overall dimensions and mass, this gear is the least favorable option.

A single stage worm reducer is the best in the mass criteria for installation, as well as the overall dimensions. However, in efficiency ratio for this type of reducer is by far the least favorable solution.

The third considered reducer, a C concept planetary reducer, according to the criteria of mass is between the single-stage worm reducer and two-stage bevel reducer. However, it has a lower efficiency ratio than the bevel reducer, but higher than the worm reducer.

In the event that the most compact design of ship winches is required, the optimal solution would be a planetary gear because of the possibility of its installation into drum of the winch.

In the event that the lightest construction is required, the optimal solution would be a single-stage worm reducer due to its low mass.

Finally, the need to give the minimum power required to run the winch drum, the optimal solution would be a two-stage bevel gear.

Optimal selection of gears for this construction largely depends on the requirements that must be placed before elaborating them.

REFERENCES:

[1] Zhu, L., & Wang, G. (2013). Distribution on Transmission Ratio of Cylindrical Gear Reducer. *Advanced Materials Research, 605-607.*

[2] Zhang, J., & Zhang, H. P. (2012). Optimization Design of the Two-stage Helical Cylindrical Gear Reducer Based on MATLAB. *Advanced Materials Research, 452-453*, 1174-1178.

[3] Yoon, J. H., & Yang, D. Y. (1990). A three-dimensional rigid-plastic finite element analysis of bevel gear forging by using a remeshing technique. *International Journal of Mechanical Sciences, 32*(4), 277-291.

[4] Jelaska, D. (2012). *Worm Gear Drives, in Gears and Gear Drives.* Chichester:John Wiley & Sons, Ltd.

[5] Inalpolat, M., & Kahraman, A. (2009). A theoretical and experimental investigation of modulation sidebands of planetary gear sets. *Journal of Sound and Vibration, 323*(3–5), 677-696.

[6] Al-shyyab, A., & Kahraman, A. (2005). Non-linear dynamic analysis of a multi-mesh gear train using multi-term harmonic balance method: period-one motions. *Journal of Sound and Vibration, 284*, 151-172.

[7] Blagojević, M., Marjanović, N., Đorđević, Z., Stojanović, B., & Dišić, A. (2011). A New Design of a Two-stage Cycloidal Speed Reducer. *Journal of Mechanical Design (ASME), 133*(8).

[8] Blagojević, M., Kočić, M., Marjanović, N., Stojanović, B., Đorđević, Z., Ivanović, L., & Marjanović, V. (2012). Influence of Friction on the Cycloidal Speed Reducer Efficiency. *Journal of the Balkan Tribological Association, 18*(2), 217-227.

[9] Tuttle, T. D. (1996). A nonlinear model of a harmonic drive gear transmission. *Robotics and Automation, 12* (3), 368 - 374.

[10] Matejić, M. (2012). *Design and calculation of ship winch in CAD software* (Master thesis, Faculty of engineering scinces of University in Kragujevac).

[11] Moe, A. K. (2006). *Design Consideration For Winch* (Master thesis, National Universityof Singapore).

[12] Det Norske Veritas (DNV), Oct 2005, Offshore Standard DNV-OS-D101, Chap.2 Sec.5, pp. 49-56.

[13] Nord Drive Systems, G1000 Catalogue IE2, catalogue,2012,pp 313.

[14] Techniche Antriebselemente, T. E. A, GmbH, Druckfehler, Maß- und Konstruktionsänderungenvorbehalten, catalogue 2011, Sec V pp. 181.