

Analysis of rope transmission from the point of view of rope life

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Abstract This paper analyses the possibility of increasing the life of the rope on the shifting device working on the traction transmission of the driving force. In the original solution, the service life of the rope did not even reach the designed value. By increasing the rope wrap angle on the drive drum, replacing the original spil drum with a rope with grooves for the rope, and changing the rope drive transmission, the rope load values changed to lower than allowed by the standard.

Keywords rope durability, angle of lap, rope gearing,

1. INTRODUCTION

At present, the transshipment facility in Čierna nad Tisou handles more than 90% of raw materials and goods imported to Slovakia by rail from Eastern Europe and Asia [8]. The importance of the status of the transshipment point is enhanced by its uniqueness of the largest transshipment point with a comprehensive range of services from big gauge (BG) - 1520 mm - to normal gauge (NG) - 1435 mm in Slovakia. The transshipment area with a rotary tipper is one of the most modern and efficient facilities of its kind in Europe. The feed of wide gauge wagons to the tipper space is realized by a traction rail vehicle (locomotive). The handling of empty wagons from the tipper and on the ramp is ensured by a wide-gauge rope-moving device. The feed of normal gauge wagons for loading is also realized by a sliding track vehicle, which places the first wagon under the hopper and further manipulation is ensured by a normal gauge sliding device with traction drive.

2. MATERIALS AND METHODS

Technical parameters of the main tipper equipment:

Load capacity of the rotary tipper	100t
The weight of a full railway carriage	about 92 t
Wagon length	14,040 mm
Wagon height	3275 mm

Unloading cycle (feed, tipping, removal)	5 min.
Operation	24 hour
Annual transshipment performance	3 million tons.

The ratio of interlaced material is 80% iron ore substrates and 20% coal substrates and coke. Volume ratios are given in Tab.1.

Tab.1: Volumes of tipped commodities

Raw material	Share	Tipped volume t/year	Utilization of waggon t/waggon
Orw, sinter	cca 20 %	600 000	69 ton
Coke	cca 10%	300 000	35 ton
Coal	cca 10 %	300 000	68 ton
Pellets	cca 60 %	1 800 000	69 ton
Total amount transferred		3 000 000	

The maximum number of wagons in the set ready for unloading is recommended to be 27 (wide gauge) and the corresponding number of normal gauge wagons needs to be prepared 33 to ensure a smooth transshipment.

The displacement of wagons on the track is provided by a displacement device with the following parameters:

tensile force = 80 kN,
travel speed = 0.328 m / s (19.68 m / min),
tow rope - \varnothing 25 STN 02 4324.57 $F_u = 387.1$ kN,
electric motor - ILAS 220-4AA, 37kW, 1475 rpm. / min.
transmission TSA 031371-07;

In traction drive, the required driving force is created by the friction of the rope on the circumference of the drum (type of spool) or the traction sheave. As the required driving force is generated by friction, specially shaped grooves are used in addition to smooth discs and drums. The disadvantage of drive drums (especially the type of spil) is that the rope is constantly moving on the drum, unless the drum is parabolic like the head of a winch. In this case, the rope starts to move when the angle of inclination β (Fig. 1) of the drum shell exceeds the angle of friction. The service life of the rope is significantly reduced by sliding.

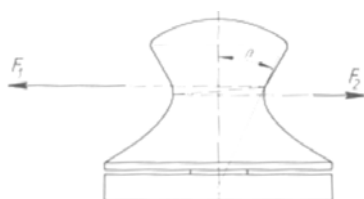


Fig.1 Friction drum type spil [8]

In such cases, Euler's equation is used to calculate the force in the rope:

$$F_1 \leq F_2 \cdot e^{\alpha f} \quad (1)$$

F_1, F_2 - there are forces in the rope,

α - is the angle of contact,

f - coefficient of friction between rope and drum (pulley).

Transmitted circumferential force:

$$F_c = F_1 - F_2 \leq F_2(e^{\alpha f} - 1) \quad (2)$$

Tab. 2 shows the most frequently used values of the coefficient of friction between the steel rope and the base material on the drum (pulley).

Material of drum (pulley)	Coefficient of friction
Steel, cast iron	0,12
Rubber with fabric	0,22
Plastics	0,25
Light metal	0,35

An increase in the transmitted circumferential force is most often achieved by increasing the wrap angle. For drive drums, the rope is wrapped around the drum several times. The driving ability can be increased by using special groove shapes (Fig.2), while the value of the coefficient of friction f changes to f' but at the expense of the life of the rope.

The shifting device at the transshipment point in Čierna nad Tisou is equipped with a traction drive with a parabolic friction drum (Fig.3). If we consider the coefficient of friction between the wagon wheel and the rail $f_1 = 0.1$, at a maximum wagon weight of 92 t, a tensile force in the rope $F_1 = 92\,000 \cdot 9,81 \cdot 0,1 = 90\,252$ N is required to overcome the frictional resistance. The traction of the shunting device is dimensioned at 80 kN, which means that it is exceeded by less than 13% when the wagon is full. When using a rope with a diameter of 25 mm and a construction according to STN 02 4324.57, the load capacity of the rope is $F_u = 387.1$ kN.

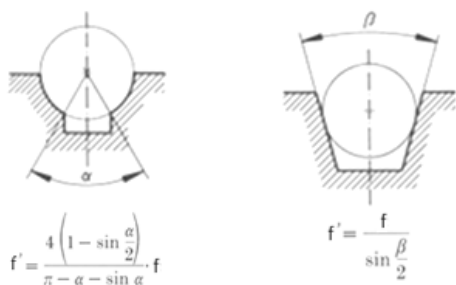


Fig.2 Special groove shape adjustment to increase the coefficient of friction [8]

Thus, the required pulling force represents 23.3% of its carrying capacity, which is satisfactory in the present case. From the point of view of reliability, the tow rope appears to be the weakest link in this solution for moving the wagons onto the tipper. His life expectancy was at least three years. When the device was put into operation, the rope lasted less than a year.

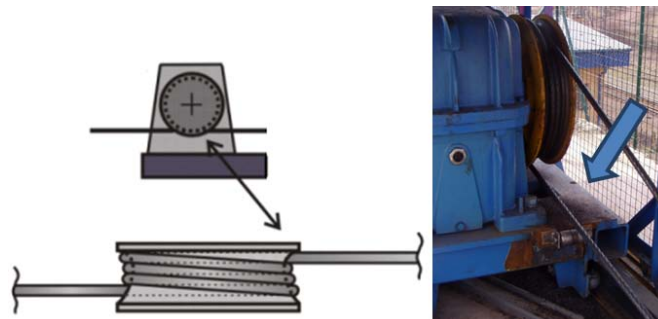


Fig. 3 Driving drum of continuous winch (Spil)

The driving drum of the continuous winch (Spil), through which the driving force is transmitted from the motor to the rope, appears to be the element which causes the most damage to the rope. By winding the rope on the drum several times to ensure a sufficient wrapping angle and the resulting driving force, the rope does not fit ideally on the drum (Fig. 3 on the left), but the individual loops of the rope overlap due to dynamic processes. This causes increased wear of the rope and the associated compressive stress leads to its rapid degradation. The visual proof is the "milled" small particles of rope wires around the drive drum (Fig.3 on the right). The use of other traction elements, such as chains (link or Gall), is not suitable, because the rope is an ideal solution precisely because of its elastic properties when engaged.

By analysing the possibilities of remedying this situation, two basic solutions were considered, taking into account the minimal intervention in the original construction of the power station. As a first solution, the exchange of the drive drum (continuous winch - spools) for a friction drum or a pair of friction drums with cut grooves for laying the rope came into consideration, as can be seen in the schematic diagram in Fig.4. The solution in Fig. 4a (left) provides a higher wrapping angle than the solution in Fig. 4b (right) with the same number of rope loops.

Both methods allow two-way operation with the same drive parameters. The only problem that still needed to be solved was the placement of the second friction drum, as the original transmission did not allow it. The second solution that came to mind was to reduce the force in the rope. In the original design of the shifting device, the ends of the rope were fastened with rope clamps to the shifting carriage (Fig. 5). One end of the rope on the side of the continuous winch (Fig.5a), the other end on the side of the wagons (Fig.5b). In this case, the rope gear is $i = 1$. If the rope gear $i = 2$ were used, the force in the rope would be halved, but it would be necessary to use a rope half as long and two extra pulleys (Fig.6).



Fig.4 Friction drum with controlled placement of the rope



Fig.5 Attaching the ends of the ropes to the trolley

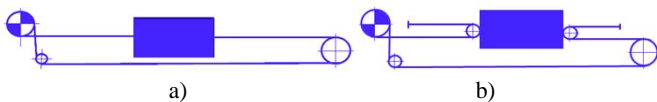


Fig.6 Drive of a trolley with rope transmission a) $i = 1$ and b) $i = 2$

The most important attribute for a steel rope user is its durability [1-5]. In addition to the force in the rope, the pressure between the rope and the pulley (drum) significantly affects its service life in such types of equipment. Pressure between the rope and sheave also affects the life time of steel wire ropes and is dependent on the load of the rope. The size of the maximum pressure, by which acts the rope on the sheave (pulley, drum), can be calculated according to the following formula

$$p_{max} = \frac{3 \cdot F_{max}}{D \cdot d} \quad [Pa] \quad (3)$$

where: p_{max} – maximum pressure, by which acts the rope on the sheave (pulley, drum) in [Pa],
 F_{max} - maximum static load of steel wire rope in [N],
 D – diameter of the sheave (pulley, drum) in [m],
 d – diameter of the wire rope in [m].

According to [6] the recommended pressure values are given in Table 3. In [7] the course of the dependence of the specific load of the steel rope on the fatigue cycles, which represent the durability / service life of the steel rope, is shown.

Tab.3 Pressure in the groove drive roll [MPa] [6]

Group of elevator machine	Rope speed [m/s]								
	0,3	0,5	0,7	1,0	1,4	2,0	2,8	4,0	Over 4,0
I	8,2	7,2	6,3	5,7	5,6	4,2	3,8	3,5	3,5
II	8,9	8,0	7,1	6,5	5,9	5,2	4,8	4,5	-
III	9,6	8,6	8,0	7,3	6,6	6,2	-	-	-
IV	10,2	9,4	8,8	8,2	7,7	-	-	-	-

- I - duty cycle over 40% or the number of cycles greater than the 90/hour,
- II - up to 40% duty cycle or the number of cycles up to 90/hour,
- III - up to 40% duty cycle or the number of cycles up to 60/hour,
- IV - up to 20% duty cycle or the number of cycles up to 30/hour.

By examining market opportunities [9], rope pulley manufacturers and design options for changes to existing equipment, a solution was proposed with the principles shown in Figures 4a and 6b. This solution increased the belt angle to meet the required excessive frictional force, eliminated the crossing of the rope and increased the diameter of the drive drum to the maximum possible extent, i.e.

from 600 mm to 800 mm. The stated value of 800 mm was limiting for the maintenance of the existing drive, otherwise it would be necessary to change the engine and transmission. Hitherto, rope gear 1 has been used, the proposed modification assumes the insertion of one additional pulley (on both sides of the trolley), thus increasing the rope gear to $i = 2$, thus reducing the force in the rope to half. The recalculation of the given conditions leads to the following data: For rope diameter 25 mm, drive wheel diameter $D = 600$ mm, rope gear $i = 1$, considering the maximum driving force $F_{max} = 80$ kN, the maximum groove pressure $p_{max} = 16$ MPa. Due to the fact that the rotational capacity of the transshipment (in continuous operation) is 2.5 to 2.8 million tons per year, the output of the dump truck [6], including the handling of wide gauge wagons, is 67 t / 5 min. This means that the tipper is able to serve 12 wagons per hour. According to Table 3, it should be IV. elevator machine group with the number of cycles up to 30 / hour, for which max. pressure about 10.2 MPa for movement speed $v = 0.328$ m / s. It can be seen that for the maximum pulling force, the allowed value is exceeded 1.57 times. The wagons are pulled on the tipper one by one, i.e. the maximum pressure in the wrapped rope for the considered coefficient of friction between the wheel and the rolling stock $f = 0,1$ will be $p_{max} = 13,1$ MPa for the maximum load capacity of the wide gauge wagon 67 tonnes. The value is thus again exceeded 1.28 times. For a maximum wagon weight of 92 tonnes, this value would be $p_{max} = 18$ MPa, which would mean exceeding the maximum allowable pressure value of 1.76 times. In the proposed modification (rope transmission $i = 2$, drive drum diameter $D = 800$ mm, rope diameter $d = 25$ mm), when pulling 1 wide gauge wagon into the dump truck, the maximum pressure $p_{max} = 4.9$ MPa would act between the rope and the drum (for the wagon 67 tonnes), which is less than half the recommended maximum pressure for a given duty cycle. For a maximum wagon weight of 92 tonnes, this value would be $p_{max} = 6.8$ MPa, which would be suitable for this case as well.

3. CONCLUSION

If we need to monitor the life of the rope, then the calculation of its specific load (p_{max}) is a suitable alternative for this type of equipment. Ultimately, after analysing the actual condition and the proposed modifications as well as the options for rebuilding the existing facility, modifications were proposed that were accepted by the facility operator. Another drive wheel (drive with sufficient belt angle - Fig. 4a) was inserted into the sliding frame, which is hydraulically tensioned, as shown in Fig. 7, the rope transmission was adjusted to $i = 2$ according to the diagram shown in Fig. 4. 6b.

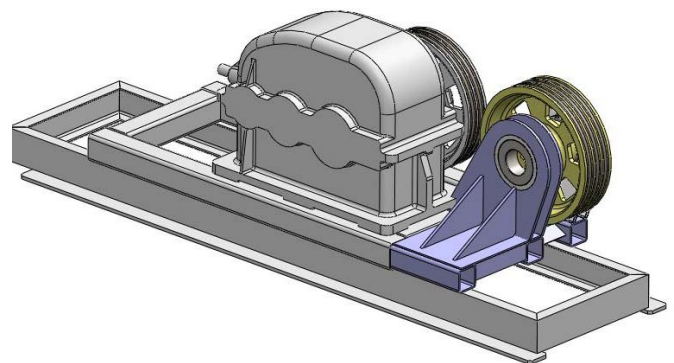


Fig.7 Auxiliary construction for inserted pulley

Such a design modification achieved the required service life of the rope and at the same time the reliability of the entire device for moving the wagons into the dump truck.

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