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INCREASING DURABILITY IN STEEL WIRE ROPE INSTALLED IN SPECIAL TRANSPORT EQUIPMENT

Summary. In this paper, special transport-handling equipment is presented for the purpose of pushing wagons in a trans-shipment facility. During operations, the rotary tilter device, which is an integrated part of the trans-shipment process, leads to excessive wear of the steel wire rope used for pushing wagons by means of a pusher system. Therefore, the main task was to propose a suitable design modification of the given pusher system in order to eliminate excessive wear of the rope and, in turn, to prolong operational durability for this steel wire rope, as well as for the whole technical system.

Keywords: steel rope durability, wagon, tilter, pusher system, driving station

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1. INTRODUCTION

Possible causes of rope damage have been addressed by certain authors [5,6,7,8]. In paper [9], the author describes a mathematical and geometric model for computer simulation of wire ropes, while [10] presents friction losses in gearboxes are analysed with regard to their efficiency. For users of steel wire rope, durability is the most important attribute, as it affects the economy of the device with the steel cable as a part. Of the many factors affecting the service life of wire rope, loading is very important. The size of loading is one of the variables, by which we characterize the device. A rope bears static, dynamic and bending forces during its operation. Their calculation is quite simple. If we know the static load of the rope, we can calculate the specific load of the rope, as well as the pressure between the rope and the sheave (i.e., drum or pulley). On the basis of the calculated values and by comparing them with table values, we can assume, with a certain level of probability, the achievement of a certain durability or service life.

The durability of the rope is affected by many factors, which can be divided into the following three basic groups [1]:

- production (rope design, quality of production, cross section and strength of wires etc.)
- service (maintenance, environment, methods of operation etc.)
- factors influenced by a device (material and shape of the grooves of a pulley, diameter and angle of wrap around the support and deflection rollers, angle of attack of the rope to a pulley etc.)

Individual factors do not act alone. In other words, during the operation of the rope, current and mutual interaction of several factors occur. From the static load of the rope, it is possible to calculate its specific load, as well as the pressure between the rope and pulley (sheave, drum). From calculated values of these variables, it is possible to assume the achieved durability of the steel wire rope in the corresponding device. Static load of the rope is determined by the sum of all loadings of the devices working together on that rope, including its own weight, if the rope is moving in a vertical plane. In addition to this calculation, it is possible to obtain loading values by using different measuring devices (load limiters, scales, load cells), which show the real values of the load at the specific time. From these measurements, we can find that forces in the rope are not constant, but change during the working cycle.

The change of load size is caused by the action of dynamic forces arising from different stimuli, such as:

- resistance against the load movement due to passive resistance
- acceleration or deceleration of the load movement at the start or finish of lifting equipment
- longitudinal and transverse oscillations of wire rope

2. ANALYSIS AND CALCULATION OF SPECIAL PUSHER SYSTEM

The maximum rope tensile force S is determined according to the real value of the operational load and using the ratio value of block and tackle. The next step involves the calculation of the rope diameter, after which the mean radius of the rope drum is calculated.

According to the STN ISO 4308/1 (STN 27 0050), the rope diameter d (mm) is obtained by using the following relation:

$$d = C.\sqrt{S}\ , \tag{1}$$

where C is the coefficient of rope selection and S is the maximum loading of the rope (in N), which is obtained by means of the following coefficients:

- nominal operational loading
- weight of the pulley block
- block and tackle ratio value
- efficiency of the block and tackle ratio value

The minimal value of the load carrying capacity is:

$$F_0 = S\ Z_p\ \ , \tag{2}$$

where S is the maximal loading of the rope (in N), while Z_p is the minimal value of the rope safety coefficient, which is selected according to the classification of the hoisting mechanism (from M1 to M8, Table 3).

The minimal value of the rope drum diameter and the individual rope pulley diameters are calculated from the minimal value of the rope diameter (1), using the values h_1, h_2 and h_3, with regard to the classification of the given hoisting mechanism (see Table 1):

$$D_1 \geq h_1\ d;\quad D_2 \geq h_2\, d\ ;\quad D_3 \geq h_3\, d\ \ , \tag{3}$$

where D_1 is the mean diameter of the drum (mm), D_2 is the mean diameter of the pulley (mm), D_3 is the mean diameter of the balancing pulley (mm), d is the minimal diameter of the rope (mm), h_1 is the selective coefficient for the drum, h_2 is the selective coefficient for the pulley, and h_3 is the selective coefficient for the balancing pulley.

Table 1

Selective coefficients h_1, h_2, h_3

Classification of mechanism	***Drums*** h_1	***Pulleys*** h_2	***Balancing pulleys*** h_3
M 1	11.2	12.5	11.2
M 2	12.5	14.0	12.5
M 3	14.0	16.0	12.5
M 4	16.0	18.0	14.0
M 5	18.0	20.0	14.0
M 6	20.0	22.4	16.0
M 7	22.4	25	16.0
M 8	25.0	28	18.0

At the same time, the rope tensile force is also the circumferential force, which is acting on the mean radius of the drum; in turn, the torque on the drum shaft is also known. Selection of the driving system and the total transmission ratio are determined with regard to the rope tensile force and the hoisting speed.

However, it is necessary to calculate the number of the rope windings on the drum, in order to create the required friction force between the rope and the drum.

There is well-known relation defined according to Euler, i.e., $\frac{F_1}{F_2} = e^{\alpha f}$, which describes a phenomenon of the belt friction, which occurs between the rope and the drum. The force value F_1 is the input tensile force in the rope (tension on the pulling side), the force value F_2 is the output tensile force in the rope (tension on the resisting side), f is the coefficient of friction between the rope and drum, α is the wrap angle around the rope drum and e is the base of the natural logarithm.

Table 2 offers information about the required number of rope windings on the drum (or the wrap angle) with regard to the $e^{\alpha f}$ values.

Table 2

Required number of rope windings for various coefficients of friction between the rope and the drum

Values $e^{\alpha f}$			
Number of windings	***f = 0.13***	***f = 0.15***	***f = 0.18***
0.5 (α = π)	1,503	1,602	1,758
1.0 (α = 2π)	2,260	2,565	3,090
2 (α = 4π)	5,105	6,583	9,550
3 (α=6π)	11,534	16,890	29,512
4 (α=8π)	26,062	43,331	91,202
5 (α=10π)	58,884	111,170	281,840
6 (α=12π)	133,040	285,230	870,960

Table 3

Z_p values

Classification of mechanism	M 1	M 2	M 3	M 4	M 5	M 6	M 7	M 8
Value of Z_p	2.5	2.5	3.0	3.5	4.0	4.5	5.0	5.0

The specific load of steel wire rope is a relatively simple and characteristic value for steel wire ropes. It allows for the determination of working conditions of the rope and its load. The specific load of the rope is defined [2] as the ratio between the maximum static load of the rope and its cross section:

$$\sigma_m = \frac{S}{A} \quad (Pa) \; , \tag{4}$$

where S is the maximum static load of steel wire rope (in N), while A is the bearing cross section of wire rope (in m^2).

Figure 1 [2] illustrates the course of dependence in relation to the specific load of steel wire rope on fatigue cycles, which represent the durability or life of steel wire rope.

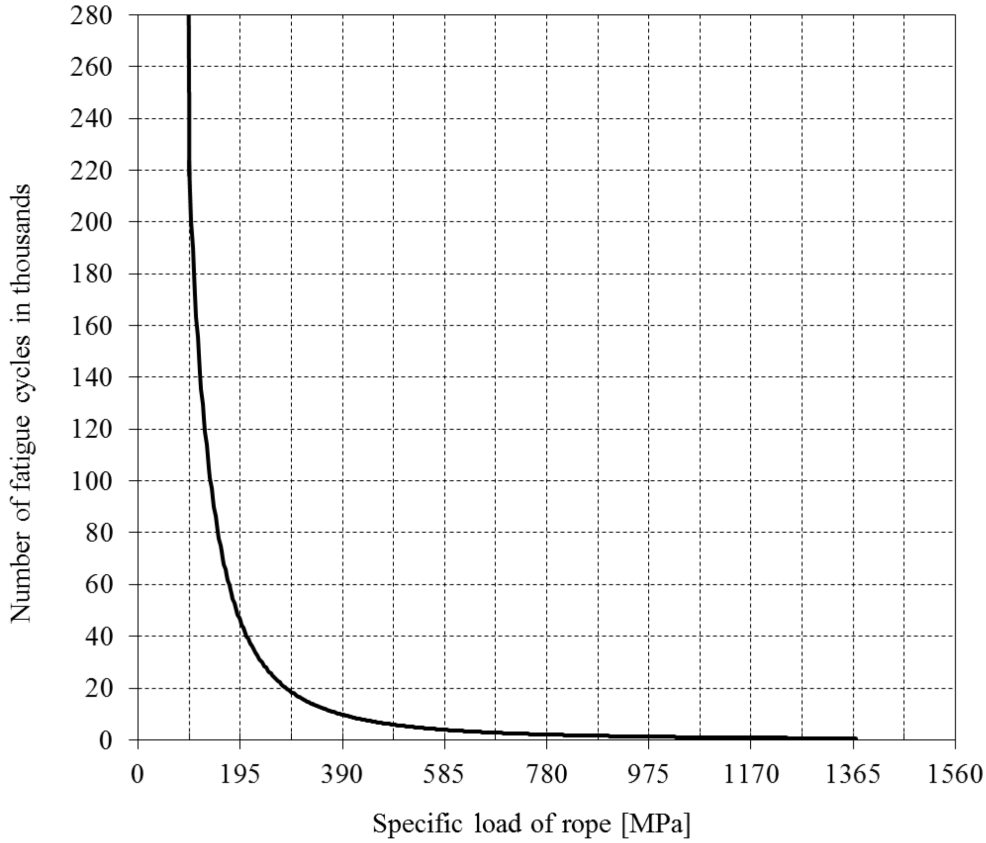

Fig. 1. Curve of specific load of steel wire rope

Pressure between the rope and sheave also affects the durability of steel wire ropes and is dependent on the load of the rope. The size of the maximum pressure, with which the rope acts on the sheave (pulley, drum), can be calculated according to the following formula:

$$p_{max} = \frac{3\,S}{D_i\,d} \quad (Pa) \tag{5}$$

where p_{max} is the maximum pressure, with which the rope acts on the sheave (pulley, drum) (in Pa), S is the maximum static load of steel wire rope (in N), D_i is the diameter of the sheave (pulley, drum) (in m), and d is the diameter of the wire rope (in m).

According to the above-mentioned formula, it can be seen that the size of pressure between the rope and sheave at a given static load can be reduced by increasing the diameter of the sheave (pulley, drum) and the diameter of the rope. That said, significantly increasing the diameter of the rope from the point of view of stress is not particularly important. The ratio between D_i and d, which is defined according to individual lifting and towing equipment, ranges from 25 to 100. The recommended pressure values are given in Table 4.

Currently, the Eastern Slovak Trans-shipment Yards (ESTY) provide trans-shipment for over 90% of raw materials and goods imported to Slovakia by rail from Eastern Europe and Asia [3]. The significance of the ESTY' status enhances their uniqueness as the largest of their kind, offering a comprehensive range of services from broad gauge (BG) (1,520 mm) to normal gauge (NG) (1,435 mm), which is applicable to Slovakia.

The crucial technological device for reloading involves the rotary tippler wagon on BG track with a load capacity of 100 t (see Figure 2.). Tippler is a rotating device in the shape of a keg, into which the wagons are inserted. After stopping the carriage and fixing the wagon, the whole tippler turns upside down around the longitudinal axis of the wagon. After tipping the wagon and stabilizing the tilter in the basic position, the wagon is then pulled from the tilter by a pusher located on the high ramp at Facility III (see Figure 3).

Table 4

Pressure in the groove drive roll (MPa) [4]

Lifting machine group	***Rope speed*** (ms^{-1})								
	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 90/hour
II - duty cycle up to 40% or the number of cycles up to 90/hour
III - duty cycle up to 40% or the number of cycles up to 60/hour
IV - duty cycle up to 20% or the number of cycles up to 30/hour

3. DESIGN MODIFICATION OF TRANSPORT-HANDLING EQUIPMENT

After turning the tilter, the contents of the wagon will be flown through a steel grate into the tray (a slip hopper), the volume of which should ensure retention of the material (it has the volume of about two BG wagons, i.e., approximately 134 t of transported material). Two mills (drum crushers) are installed in the area above the grate, which will be put into operation if the ore is frozen or creates a larger chest, such that, by passing over the entire grid, seized lumps or insufficiently unfrozen material are crushed. The bottom tray is finished with four scraper belt conveyors, which evenly shovel spilled material from wagons. Positioned on the conveyor belt is a continuous scale for indicative weighing of interlaced material. The weighed amount of the interlaced material from the conveyor belt passes through the separator of items, onto a buckle conveyor and then onto a reverse belt into the loaded NG wagon, which stands on the track as static weight. The static railway scale provides official weighing of trans-shipped goods. After weighing the wagon, an NG carriage assembly will pass the length of the wagon. The subsequently empty wagon is weighed, while the loading of the corresponding quantity of material continues. After filling the entire set of

NG wagons, the pusher rolls back the assembly in front of the construction of the conveyor belt, where the set is attached to the locomotive. The service staff then provides automatic disconnection from the pusher, such that the full set can be pulled away from the loading area.

The pusher was put into operation in its current form in 2009. Its technical parameters are as follows:

- pulling power = 80 kN
- travel speed = 0.328 m/s^{-1} (19.68 m/min^{-1})
- drag rope - STN 25 02 4324.57, F_u = 387.1 kN
- electric motor - 1LAS 220-4AA, 37 kW, 1,475 rpm
- transmission - TSA 031371-07 gearbox

The weakest link in this solution for shifting wagons towards a tilter appears to be the drag rope, whose expected lifetime is at least three years.

Fig. 2. Facility III: high ramp with tilter

Fig. 3. Reel of pusher and carriage of pusher

However, the rope was torn after nearly a year of operations. Using another tension member, for example, chains (cell or Gall's), is not suitable because the rope represents the ideal solution, given its elastic properties in relation to engagement.

Therefore, an analysis of the possible causes of damage of the rope was carried out, which revealed that there was a synergistic effect of several partial injuries.

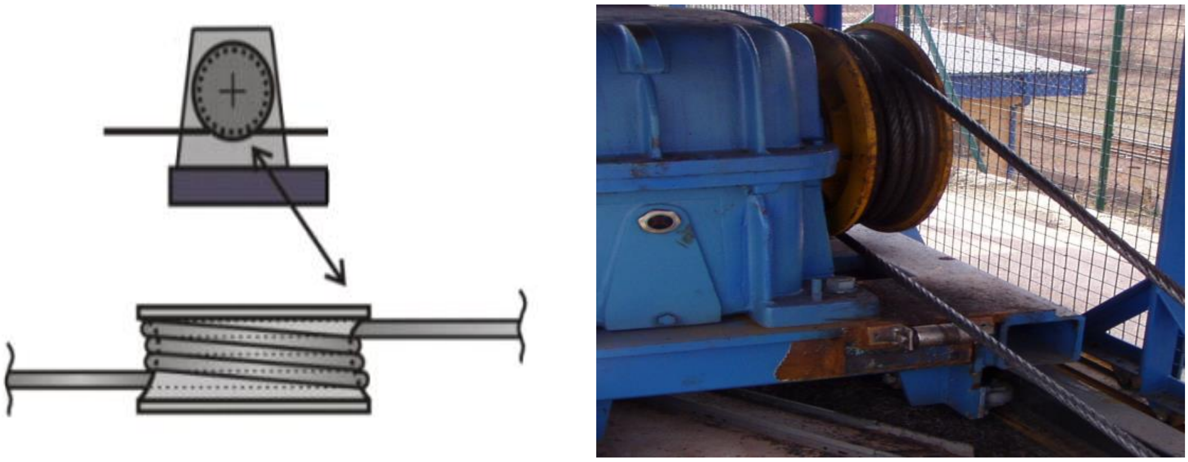

Fig. 4. Driving drum of a continuous winch (SPIL)

The least reliable element, in terms of an element that harms the rope the most, appears to be the driving drum of a continuous winch (SPIL), through which the driving force is transmitted from the engine to the rope. By multiple wrapping of the rope on the drum, in order to ensure a sufficient angle of wrap and the resulting required driving force, there is no ideal stacking of the rope on the drum (Figure 4, left). Due to the influence of dynamic processes during the shifting of wagons, however, overlapping of individual loops of rope occurs. This causes increased wear of the rope, which, along with associated compression stress, leads to its rapid degradation. Visual verification of this excessive wear highlights the amount of “milled” tiny particles of rope wires around the drive drum (Figure 4, right).

Fig. 5. The principle of the proposed change of rope drive

By examining the market opportunities, rope sheave production approaches and design capabilities for changing the existing equipment, the solution with the principle shown in Figure 5 is proposed. This solution was found to increase the angle of wrap to comply with the required excess friction force, as well as remove the rope crossing and increase the diameter of the drive drum to the maximum possible rate, i.e., from 600 mm to 800 mm. The given value of 800 mm is limiting for the preservation of the existing drive; otherwise, it would be necessary to change the motor and gearbox as well. Modifying the cable transfer is also proposed. Until now, cable transmission 1 has been in use; for the proposed adjustment, we assume one more sheave to be inserted, which will increase the cable transfer to i = 2, resulting in changes in the strength of the pull rope to a half value.

The recalculation of given conditions, according to Equation (5), results in the following data: rope diameter = 25 mm, diameter of the drive wheel D = 600 mm, cable transfer i = 1, when considering that the maximum driving force S = 80 kN, which is the maximum pressure in the wrap p_{max} = 16 MPa.

Given that the rotation capacity of trans-shipment (for continuous operations) is 2.5 to 2.8 Mt/yr, the performance of the tilter [2], including handling of BG wagons, which equates to 67 t/5 min. It means that a tippler is able to serve 12 wagons per hour. According to Table 1, it should be from lifting machine group IV, where the number of cycles is up to 30/hour and for which the recommended maximum pressure is about 10.2 MPa for the travelling velocity, v = 0.328 ms^{-1}. It can be seen that, for the maximum pulling force, the permissible value is exceeded 1.57 times. A BG wagon load is 67 t (for an NG wagon, it is 55 t). Wagons are drawn to the tilter one by one, i.e., the maximum pressure in the wrap rope for the considered coefficient of friction between the wheel and the rail wagon (i.e., 0.1) will be p_{max} = 13.1 MPa. In other words, the value is again exceeded 1.28 times. In the proposed adjustment, when pulling one BG wagon into the tilter, the maximum pressure p_{max} = 9.8 MPa would also create pressure in the wrap drum with a diameter of 800 mm. If we were to use the rope gear i = 2, then the maximum pressure would be p_{max} = 4.9 MPa, which is less than half of the recommended maximum pressure value for the given duty cycle.

4. CONCLUSION

This article has highlighted how the specific load of steel wire rope, as well as the pressure between the rope and sheave, pulley or drum, will enable the assessment of the durability or life of such rope during a particular operation. At the same time, the example of the pusher used with the tilter has shown that suitable design modification of installed and routed cables can considerably extend the durability of this transport-handling device.

This paper was elaborated in the framework of the following projects: VEGA1/0197/14 – research on new methods and innovative design solutions in order to increase efficiency and to reduce emissions of transport vehicle driving units, together with the evaluation of possible operational risks: VEGA 1/0198/15 – research on innovative methods for emission reduction of driving units used in transport vehicles and the optimization of active logistic elements in material flows, in order to increase their technical level and reliability; and KEGA 021TUKE–4/2015 – development of cognitive activities focused on innovations in educational programmes within the discipline of engineering, as well as the building and modernization of specialized laboratories, which are specified for logistics and intra-operational transport.

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