DESIGN OF CHAINS OF CHAIN CONVEYOR AS PART OF MOBILE WORKING MACHINE MODIFICATION

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Abstract. The article presents a case study focused on working activity of a commercially produced mobile working machine. This machine serves as a wood semi-automatic splitting machine. Its function consists in transporting a wood log to the machine, where the operator cuts it into selected lengths of blocks using a chainsaw. The cut logs are subsequently split using a splitting wedge controlled by a hydraulic piston. The saw, the hydraulic piston and the belt conveyor are driven hydraulically from a pump driven by an electric motor with an output of 9.5 kW. The machine manufacturer states in the documentation that the processing capacity of wooden logs is 5 to 8 $m^3 \cdot h^{-1}$. However, this figure is impossible to achieve in real operation, when experimental work managed to process a maximum of 1 $m^3 \cdot h^{-1}$. The main reason why it was not possible to converge to the manufacturer's theoretical values was explained by the difference in the input data. The machine manufacturer states in the documentation that the maximum diameter of the input log is 500 mm. A log of this size will not actually fit into the machine. Even when using the maximum diameter of the log that can be inserted into the machine (approx. 450 mm), there is a problem with its transportation. The semi-automatic splitting machine is sold together with an input roller conveyor 1 meter long without a drive. Therefore, it is necessary to manually transport the log to the machine up to the saw, which represents approximately another meter of distance along the sheet metal part of the machine structure with significant friction. Manual transport of the log was replaced by purchasing an additional belt conveyor driven by a hydraulic motor as a replacement for the original roller conveyor. The log either hit the machine structure or it was stopped by passive resistances when the conveyor slipped. Therefore, it is necessary to modify the input conveyor consisting in the design of the chain conveyor chains.

Keywords: analysis, splitting machine, design, working machine.

Introduction

Preparing firewood is indeed a demanding process. It requires more physical effort, starting from handling the wood mass to the final product, i.e. a log intended for a fireplace or a stove [1]. At home, people mostly use hand tools, or they can use electric machines such as wood splitters. Companies engaged in professional preparation and processing of firewood are mostly equipped with semi-automatic or fully automatic splitting machines that can load the wood mass, measure it, saw it, split it and transport the finished logs to a predetermined location. These devices can handle several operations at once, which is why they are more efficient and therefore more economical compared to using manual labour. However, not all splitting devices are optimally designed, as is the case with the solved commercially produced machine (Fig. 1). Therefore, the main goal of the entire research is to modify this machine.

Fig. 1. **Mobile working machine of a commercial producer**

The object of this work is the input chain conveyor that transports the wood log to the machine. The main problem is its low efficiency, which results in a decrease in the efficiency of the machine. For this reason, the article discusses the design of a new input chain conveyor, the aim of which is to increase the efficiency of log transport and thereby achieve higher efficiency of the entire machine. Conveyors are widely used handling machines, and they find application in many sectors of industry. Conveyors

DOI: 10.22616/ERDev.2025.24.TF066

can transport logs of woods, mechanical parts, but they are also used in mines [2], the construction industry [3; 4], in intermodal terminals [5; 6], in the food industry [7; 8] and elsewhere. Usually, they include drive and driven shafts [9; 10]. Based on the operational conditions, they are exposed to various types of loads [11-14].

Materials and methods

Since this is a structural design of the conveyor over the entire transport distance and since it is a mobile working machine, it must be considered that the conveyor must not interfere with a car or quad bike when turning during its transport [15]. Two variants were considered when designing the conveyor. The first variant represents the use of a belt conveyor, where the supporting part of the conveyor would consist of individual rollers mounted in bearing housings located on the machine frame. In the case of using one belt, the hydraulic motor would have to be placed at the beginning of the conveyor and the tensioning device in the return station of the conveyor. Due to the folding of the conveyor during transport, the tensioning device would have to be released each time. The first variant could be further improved by using two separate belts and therefore two conveyors, one of which would be folded. In such a case, the tensioning device would not have to be released during transport. In that case, the hydraulic motor would be placed on the machine frame, from which it would drive both conveyors using a chain or belt. The second variant would consist of a chain conveyor design consisting of two separate chain conveyors. One of them would be folded during transport of the machine. The hydraulic motor would be placed on the machine frame and would drive the shaft on the conveyor using a chain. Of the above designs, the second variant was selected for the application. The reason is also the fact that the chain conveyor is more suitable for transmitting larger forces. With a chain conveyor, there is no slippage between the chain and the sprocket, as is the case with a belt conveyor when there can be slippage between the belt and the drum. Also, from a maintenance point of view, the second variant is more suitable, because a significantly smaller number of bearing housings that need to be lubricated will be used. The entire conveyor design will be based on the use of an analytical calculation model, which starts from determining the volume V of transported logs according to the relationship (1):

$$V = \frac{\pi \cdot d^2}{4} \cdot l, \tag{1}$$

where d – maximum log diameter resulting from the machine design, m;
l – maximum log length, m.

The maximum log diameter is d = 0.5 m, the maximum log length is of l = 2 m. Then, the log weight m is then given by (2):

$$m = \rho \cdot V, \tag{2}$$

where ρ – density of the processed wood log, kg·m^{-3}.

As a dominant processed material is supposed beech with the density ρ = 990 kg·m^{-3}. The calculation of the loading forces in the conveyor during its operation is carried out based on the release method using the scheme shown in Fig. 2 and Fig. 3.

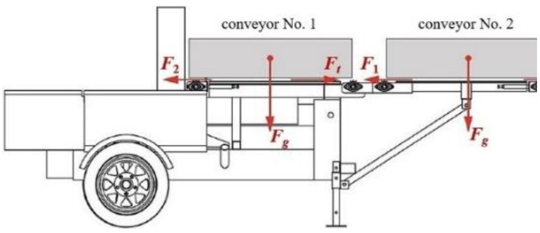

Fig. 2. **Force ratios of the proposed chain conveyor**

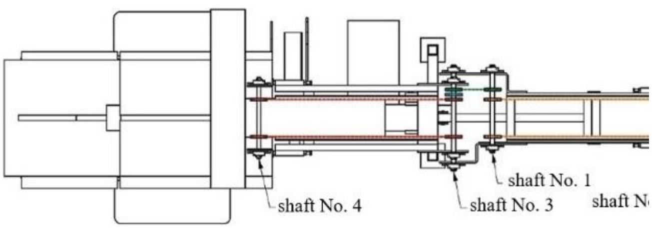

Fig. 3. **Drive concept of the proposed chain conveyor**

According to the relation (3), the static load of one conveyor F_g is determined, where half of the load weight (each conveyor takes half of the log weight) m and the gravitational acceleration g = 9.81 m·s^{-2} acts:

$$F_g = \frac{m}{2} \cdot g \tag{3}$$

and the value of the corresponding friction force F_T according to Coulomb at the static friction coefficient of the kinematic pair (steel – polyethylene) f_0 = 0.2 is (4):

$$F_T = F_g \cdot f_0 \, . \tag{4}$$

The determination of the tensile forces $F_1 = F_2$ (N) of the conveyors was carried out based on the relationship (5) using the determined constant chain operating coefficient $\vartheta = 1.4$:

$$F_{1,2} = F_T \cdot \vartheta \, . \tag{5}$$

Each of the conveyors will be equipped with two traction chains, i.e. $i_r = 2$. Then, each chain has a traction force F_r given by (6):

$$F_r = \frac{F_{1,2}}{i_r} \, . \tag{6}$$

To tension both conveyors, mechanical chain tensioning will be used using a screw with a tensioning force F_N = 1,000 N. Hence, the tensioning force in one chain is (7):

$$F_{N1} = \frac{F_N}{i_r} \, . \tag{7}$$

A single-row roller chain DIN ISO 8187 type 10 B-1 was selected from the chain catalogue (Tab. 1, Fig. 4).

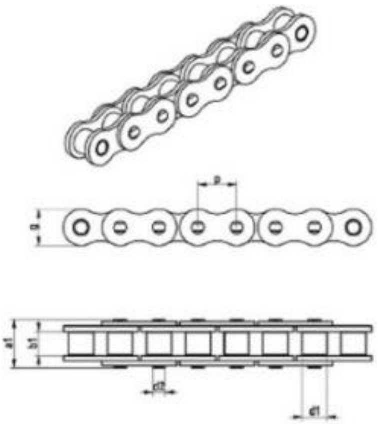

Fig. 4. **Single row roller chain DIN ISO 8187 10B-1**

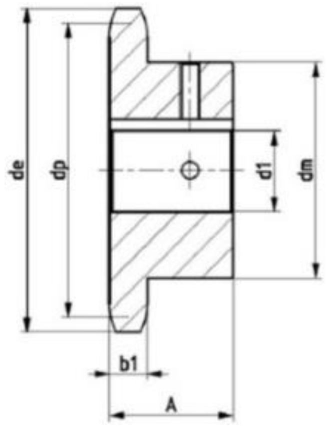

Fig. 5. **Sprocket wheel, the BEA 10 B-1 system**

Table 1

Parameters of the chain/sprocket wheel couple

Parameters of the chain 10B-1		Parameters of the sprocket wheel BEA 10 B-1	
Designation	**Value**	**Designation**	**Unit**
Pitch, p	15.875 mm	Number of teeth	14
Width, b_{1min}	9.65 mm	Diameter d_1	depends on shaft diameter
Max. roller diameter, d_{1max}	10.16 mm	Diameter d_e	78.2 mm
Max. pin diameter, d_{2max}	5.08 mm	Diameter d_p	71.34 mm
Max. plate height, g_{max}	14.73 mm	Hub diameter d_m	52 mm
Max. chain width, a_{1max}	19.6 mm	Wheel width A	30 mm
Number of rows	1	Width across the tooth b_1	9.1 mm
Chain breaking strength	27,400 N	wheel material	steel C45
Chain unit weight	0.91 kg·m^{-1}	–	–

Due to the limited dimensions of the machine, it is necessary to choose a sprocket with a pitch diameter of less than 75 mm. We choose the BEA 10 B-1 system sprocket (Fig. 5) with a pitch diameter

d_p = 71.34 mm from the Haberkorn company. The sprocket parameters are given in Tab. 1. The hydraulic motor that was on the previous input belt conveyor will be used to drive the chain conveyor. The hydraulic motor has a max. speed n_{max} = 284 rpm, a torque M_{HM} = 285 N·m and a working pressure p_p = 110 bar. The flow rate of hydraulic oil from the hydraulic pump is less than the maximum flow rate on the hydraulic motor, at which the max. speed is $n_{\max}$ = 284 rpm. By measuring the speed, the actual speed of the hydraulic motor was determined to be n_{kut} = 68 rpm. Calculation of the circumference o of the pitch circle is according to equation (8):

$$o = \pi \cdot d_p \,. \tag{8}$$

Then, the speed in m·s^{-1} of the chain conveyor is (9):

$$v = \frac{n_{skut}}{60} \cdot o \,. \tag{9}$$

The required torque M_k of conveyor No. 1 is (10):

$$M_k = F_1 \cdot \frac{d_p}{2} \,. \tag{10}$$

Hence, the transmitted power P is (11):

$$P = \frac{\left(F_1 + F_2\right)}{\eta_c} \cdot v \tag{11}$$

with the considered mechanical efficiency of one shaft supported on rolling bearings η_1 = 0.92 [14] and with the number of four shafts we consider the total mechanical efficiency η_c = 0.72. It is not necessary to consider the inclination of the conveyor, since it will operate in a horizontal position. The calculation of the chain length L_1 (14) for the conveyor No. 1 is made based on the knowledge of the number of links x_1 necessary to bridge the minimum axial distance of shafts No. 1 and 2 $a_{1\min}$ = 941 mm (12):

$$x_1 = \frac{2 \cdot a_{1\min}}{p} + z \,. \tag{12}$$

The actual number of links will be rounded to the nearest even number. Then, the actual axial distance a_1 is (13):

$$a_1 = \frac{\left(x_1 - z\right)}{2} \cdot p \,, \tag{13}$$

$$L_1 = x_1 \cdot p \,. \tag{14}$$

The calculation of the chain length for the conveyor No. 2 is identical. The number of chain links x_3 connecting conveyors No. 1 and No. 2 can be determined (15) with the known maximum axial distance of the shafts $a_{3\max}$ = 194 mm:

$$x_3 = \frac{2 \cdot a_{3\max}}{p} + z \,. \tag{15}$$

The actual number of links is chosen to the nearest even number from which it is subsequently possible to determine the actual axial distance a_3 with which the conveyor is constructed (16):

$$a_3 = \frac{\left(x_3 - z\right)}{2} \cdot p \tag{16}$$

and then, the length of the chain L_3 is (17):

$$L_3 = x_3 \cdot p \,. \tag{17}$$

The calculation of the number of chain links between the conveyor shaft and the hydraulic motor can be determined (18) with a known minimum axial distance of the shafts $a_{4\min}$ = 308 mm:

$$x_4 = \frac{2 \cdot a_{4\min}}{p} + z \,. \tag{18}$$

The actual number of links is chosen to the nearest even number from which it is subsequently possible to determine the actual axial distance with which the conveyor is constructed (19):

$$a_4 = \frac{(x_4 - z)}{2} \cdot p \tag{19}$$

and then, the length of the chain L_4 is (20):

$$L_4 = x_4 \cdot p \,. \tag{20}$$

The derived mathematical model, i.e. eqs. 1 to 20, is not created with the purpose of improving the current state of the given issue in the field of science. However, the results from these physical laws described by the equations are important from the point of view of compiling a structural design specifically adapted to the needs of the investigated (optimized) machine. The novelty of the applied research in question lies in the presentation of the beginning of the optimization process of a commercially available and manufactured machine. The overall design is immediately included to the production process, which significantly improves the current difficult process of material processing.

Results and discussion

This section presents the results of the analytical model (eqs. 1 to 20) in a clear tabular form (Tab. 2). Using this model, chains were selected that will be used in the construction of a real conveyor prototype implemented in the semi-automation process of a working machine.

Table 2

Results of solving the constructed mathematical model

Equation	**Volume, m^3**	**Result**
(1)	Maximum volume of a tree trunk, V	0.39 m^3
(2)	Maximum weight of a tree trunk, m	386.1 kg
(3)	Gravity force of a trunk per conveyor, F_g	1,893.82 N
(4)	Friction force between a steel chain and a polyethylene line, F_T	378.76 N
(5)	Tensile forces in chains of individual conveyors, F_1, F_2	530.26 N
(6)	Tensile force per conveyor chain, F_r	265.13 N
(7)	Tension force of each conveyor drive chain, F_{N1}	500 N
(8)	Circumference of the pitch circle of the selected sprocket, o	0.224 m
(9)	Actual speed of the chain conveyor, v	0.25 m·s^{-1}
(10)	Equilibrium torque at conveyor load, M_k	18.91 Nm
(11)	Power transmitted by the chain conveyor, P	368.24 W
(12)	Number of links of the conveyor chain No. 1, No. 2, x_1	134 pcs
(13)	Axial distance of the sprockets of the conveyor No. 1, No. 2, a_1	952.5 mm
(14)	Total length of the chains of the conveyor No. 1, No. 2, L_1	2,127.25 mm
(15)	Number of links of the chain connecting conveyors No. 1 and No. 2, x_3	38 pcs
(16)	Axial distance between the shaft No. 1 of the conveyor No. 1 and the shaft No. 3 of the conveyor No. 2, a_3	190.5 mm
(17)	Length of chain connecting conveyors No. 1 and No. 2, L_3	603.25 mm
(18)	Number of links of the chain connecting shaft No. 3 of the conveyor No. 2 with the shaft of the hydraulic motor, x_4	54 pcs
(19)	Axial distance between the axis of the shaft No. 3 of the conveyor No. 2 and the axis of the hydraulic motor, a_4	317.5 mm
(20)	Length of the chain connecting shaft No. 3 of the conveyor No. 2 with the shaft of the hydraulic motor, L_4	857.25 mm

The determined values allowed the design of the necessary chains, respecting the geometric and force conditions of the machine during its work. A documentary image of the detail of the chain and sprocket used is shown in Fig. 6.

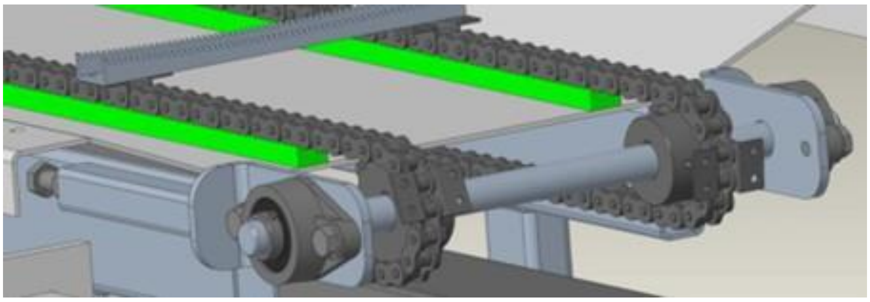

Fig. 6. **Conceptual design of a part of the chain conveyor assembly**

Conclusions

The main objective of this work was to create a part of the system of analytical equations, with the help of which it will be possible to achieve the structural design of a single-purpose chain conveyor. This can be considered fulfilled, because by comparing the value of the transmitted load and the value of the strength at the time of chain rupture, the selected chain is suitable. In further research, the authors will focus on the analytical and numerical dimensioning of all four necessary shafts and on the compilation of an accurate 3D CAD model of the proposed structure.

Acknowledgements

This research was created and supported within the following projects: KEGA 031ŽU-4/2023 and KEGA 024ŽU-4/2024. "Funded by the EU NextGenerationEU through the Recovery and Resilience Plan for Slovakia under the project No. 09I03-03-V01-00131.

Author contributions

Conceptualization, M.B.; methodology, M.B. and J.D.; software, A.L.; validation, S.S. and J.D; formal analysis, M.B and A.L..; investigation, M.B., J.D., A.L. and S.S.; data curation, J.D., A.L.; writing – original draft preparation, M.B.; writing – review and editing, J.D. and S.S.; visualization, A.L., S.S.; project administration, J.D.; funding acquisition, M.B. All authors have read and agreed to the published version of the manuscript.

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