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REDUCTION OF THE LOAD ON THE MAIN DRIVE SHAFT OF BAND SAWS

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ABSTRACT

Analysis of the load on the main drive shaft of band saw is provided in three cases:

Bending couples resulting from the strain forces of the saw band and the belt transmission are absorbed by the shaft.

- The bending couple resulting from the force of strain of the saw bend is absorbed by the shell of the bearing unit and the bending couple caused by the strain force of the belt transmission is received by the shaft.
- Bending couples from the strain forces of the saw band and the belt transmission are received by the shell of the bearing unit.

Strain forces of the saw band and the belt transmission loading the main drive shaft are determined. Validating test strength calculations of the main band saw shaft are made.

Constructive solutions to reduce the bending in the console parts of the main drive shaft which will improve the work of the saw band and the V-belt transmission are proposed.

Key words: band saw, main drive shaft, belt transmission, strength calculations, construction

INTRODUCTION

The cutting mechanism of the band-saw machines (fig. 1) is composed of upper 1 and lower 3 feed wheels and saw band 2, main shaft 4, driven by an electromotor 5 and a belt transmission 6. The main shaft 4 is the most loaded of all the elements mentioned above. The load of the shaft is: turning couple M_{t2}, tangential force F_t, tension force of the saw band F_0, the weight of the lower feed wheel G_w, tension force of the belts F_R, weight of the driven V-belt pulley G_p.

The V-belt transmission is generally decreasing ($d_{p2} > d_{p1}$).

The rotation frequency of the main shaft n_2 is determined in dependence with the cutting speed of the band-saw υ according to the expression

$$n_2 = \frac{60\upsilon}{\pi D}, \qquad \textbf{(1)}$$

Where n_2 is the rotation frequency of the main shaft, min^{-1};

υ – the cutting speed of the band-saw, m.s^{-1};

D – the diameter of the feed wheels, m.

At $\upsilon = 40 \div 50$ m.s^{-1} and $D = 0,6 \div 1,3$ m, for n_2 we obtain the interval $n_2 = 700 \div 1400$ min^{-1}. These rotation frequencies of the main band-saw shaft can be achieved with asynchronous electro-motors with synchronic rotation frequencies $n_1 = 750$, 1000 и 1500 min^{-1}. In these cases the initial ratio of the belt transmission $i = \frac{n_1}{n_2}$ is obtained in the interval $i = 1 \div 2$. Consequently, we receive the values $d_{p2} = (1 \div 2) d_{p1}$ for the diameters of the V-belt transmission.

The power of the electro-motor P_1 is determined in accordance with the power of cutting (Gochev, 2005) and the efficiency ratio of the cutting mechanism.

The dimensions l_1, l_2 and l_3 are accepted in conformity with the construction.

Subject of the present work is making an analysis of the load on the main shaft provoked by the motion of the band-saw cutting mechanism and to propose solutions for its reduction.

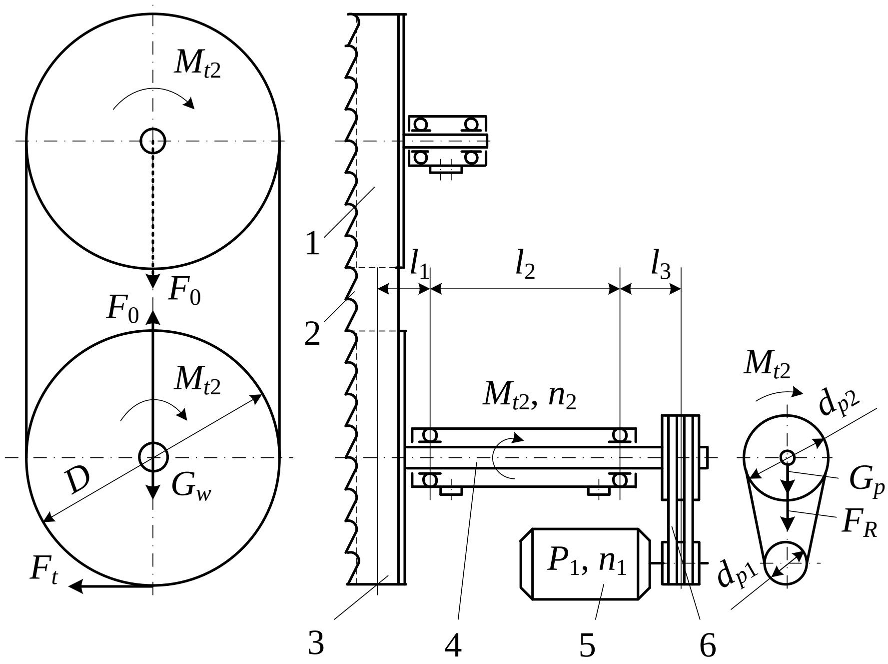

Figure 1: Cutting mechanism of band-saw machine

2. STRENGTH CALCULATION OF THE MAIN BAND-SAW SHAFT

The load on the principal band-saw shaft is complex. It is loaded by transversal forces that bend it and a torque which is twisting it (fig. 1). Three cases of shaft loading are considered: the bending moments resulting from the tension forces of the saw band and the V-belt transmission are absorbed by the shaft; the bending moment provoked by the tension force of the saw band is absorbed by the case of the bearing housing, and the bending moment of the tension force cased by the V-belt transmission is taken by the shaft; bending moments provoked by the tension forces of the band saw and the V-belt transmission are absorbed by the body of the bearing housing.

2.1. Bending moment resulting from the tension forces of the band saw and the V-belt transmission are absorbed by the shaft.

In this case the main shaft of the band saw is submitted to important bending stress in both console parts. The bending stress is provoked by the tension force of the saw band and on the other hand by the belt tension. Shaft bending aggravates the work of the saw band and the V-belt transmission. The calculation scheme of the diameters of the bending and twisting moments and the construction of the main band saw shaft are shown on fig. 2.

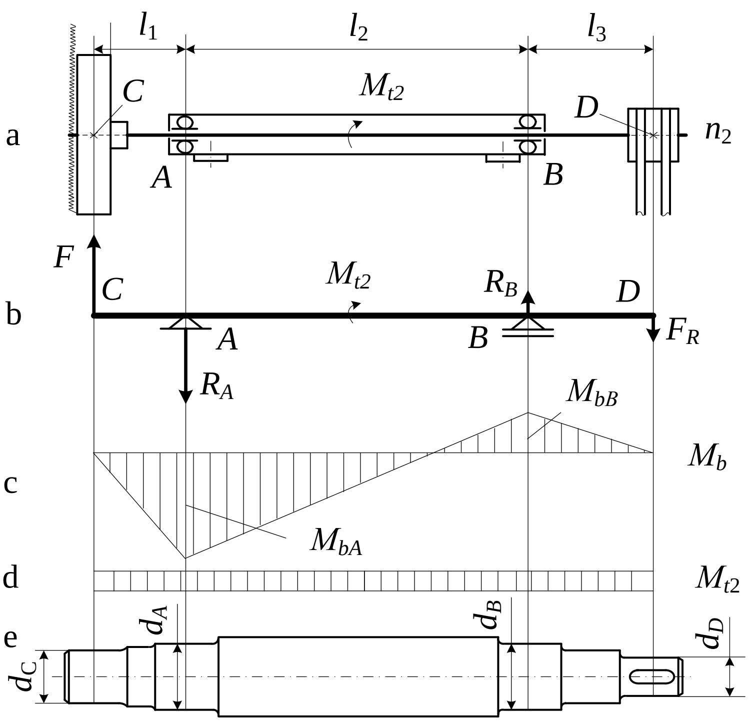

Figure 2: Calculation scheme and diagrams of bending and twisting moments and construction of the main band saw shaft.

The calculation and the construction of the shaft are developed in (Sokolovski and Deliiski, 2011). The load of the shaft is determined, the calculation scheme is realized, the diagrams of bending and twisting moments are shown and the diameters of the shaft are calculated.

The resulting transverse force F (in N) which is active in point C (fig. 2b) is determined by the expression:

$$F = F_0 + F_t - G_w, \qquad \textbf{(2)}$$

Where F_0 is the force of tension of the saw band, N;

F_t – the peripheral force, N;

G_w – the weight of the lower wheel in N.

The force provoked by the tension of the belts F_R (in N) of the vertical V-belt transmission which is active in pt. D (fig. 2б) is (Filipov, 1967)

$$F_R = 3F_t \qquad \textbf{(3)}$$

Bending moments (fig. 3) are:

$$M_{b\,C} = 0;\ M_{b\,A} = F.l_1;\ M_{b\,B} = (F_R + G_{ш}).l_3;$$
$$M_{b\,D} = 0. \qquad \textbf{(4)}$$

The twisting moment M_{t2} (in N.m), transmitted by the shaft (fig. 2d) is determined according to the formula (Sokolovski and Deliiski 2011)

$$M_{t2} = \frac{9554.P_1.\eta.d_{p2}}{n_1.d_{p1}}, \qquad \textbf{(5)}$$

Where P_1 is the power of the electromotor, kW;

η – efficiency ratio of the driving mechanism ;

n_1 – rotation frequency of the electromotor, min^{-1};

d_{p1} и d_{p2} – diameters of the driving and the driven pulley, m.

The diameters of the shaft at the limit journals in pt.C and pt.D (fig. 3e) are determined according to the expression (Sokolovski, 2007).

$$d_C = d_D \geq 1{,}1.\sqrt[3]{\frac{M_{t2}}{0{,}2\left[\tau_t^{\,'}\right]}} \qquad \textbf{(6)}$$

Where d_C, d_D are the diameters of the shaft in pt. *C* and in pt. *D*, where the feed wheel and the driven pulley are fixed, [m];

$[\tau_t^{'}]$ – reduced admissible twisting stress, [N.m^{-2}].

The diameters of the shaft in the supports pt. *A* and pt. *B* (fig. 3e) are elaborated with equivalent dimensions and thus with the dimension of the bigger one, i.e. $d_A = d_B$. In this case the gudgeon A (fig. 3c) is the one which is more loaded.

$$d_A = d_B \geq \sqrt[3]{\frac{M_{eA}}{0{,}1[\sigma_b]}}, \qquad (7)$$

Where M_{eA} is the equivalent moment in the support A, N.m,

$$M_{eA} = \sqrt{M_{bA}^2 + M_{t2}^2}\,; \qquad (8)$$

$[\sigma_b]$ – admissible bending stress, [N.m^{-2}].

The design of the shaft is of a step-like form beginning with the representation of the limit journals in sections C and D. The diameter of the journal in section C should be as important as possible. The shoulders between the journals in sections C and A should be a small number and of a small height.

The design of a part of the cutting mechanism (the main shaft with the bearings, the lower feed wheel and the driven belt pulley) is given on fig. 3.

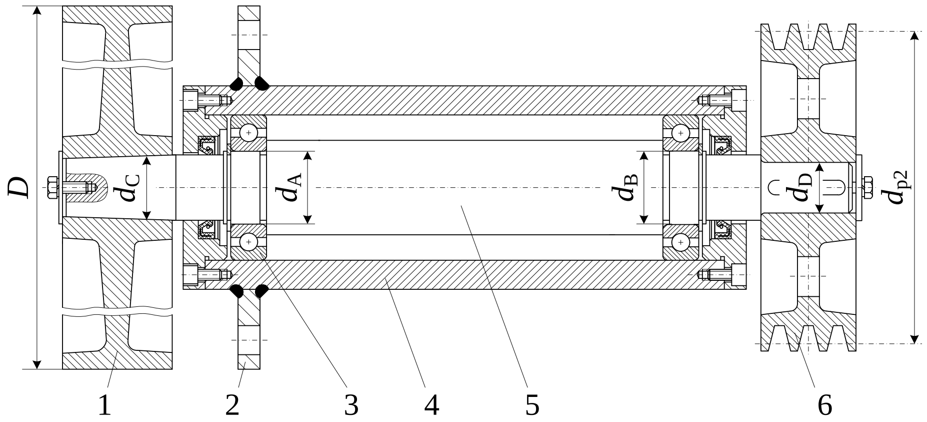

Figure 3: Design of the main band saw shaft submitted to bending by the tension forces of the band and the belts and submitted also to twisting

The lower feed wheel 1 is positioned in a console in the left end of the shaft 5 and is imbedded on the journal with diameter d_C. The shaft 5 is fixed on two radial ball bearings 3 disposed in a common bearing case 4 on the journals with diameters d_A and d_B. The bearing case 4 is composed of a thick-sectioned bush which by means of plate 2 and using a bolt joint is fixed to the body of the band saw. The driven belt pulley is disposed in the right end of the shaft 5 on the journal with diameter d_D. The joint between the shaft and the V-belt pulley is of a key-type.

After the realized analysis it was established that the maximum load of the shaft is in its section at the bearing next to the feed wheel. In this place the bending moment which is provoked by the tension force of the saw band is active.

2.2. The bending moment provoked by the tension force of the saw band is absorbed by the case of the bearing body and the bending moment by the tension force of the belt transmission is absorbed by the shaft.

According to the analysis of the diagram of the bending moment (fig. 2c) it is established that the bending moment provoked by the tension force of the belts (support B) is considerably smaller than the bending moment by the tension force of the saw band (support A). That is why in this case a medium version is possible according to which the shaft will absorb the bending moment provoked by the tension of the belts (fig. 4) whereas the bending moment resulting from the tension force of the saw band will be absorbed by the bearing body fixed firmly to the body of the band saw machine.

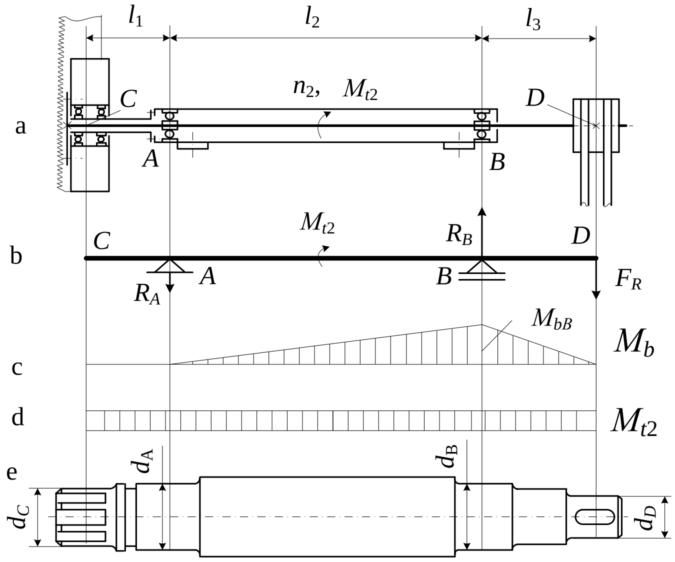

Figure 4: Calculation scheme and diagrams of the bending moment and torque and the construction of the main band saw shaft

In this case the shaft is loaded by the bending moment provoked by the tension force of the belts M_{bB}, determined according to the formula (4) and by the turning couple M_{t2}, determined according to the formula (5).

The diameters of the shaft d_C in pt.*C*, where the feed wheel is fixed and d_D in pt.*D*, where the driven belt pulley (fig.4e) is fixed are submitted only to twisting provoked by the turning couple M_{t2} (fig. 4d) and are determined by means of formula (6).

The diameters of the shaft are determined depending on the more loaded support in supports pt. A and pt. B (fig. 4e) where the antifriction bearings are fixed. In the present case the support B (fig. 4c) is more loaded. The diameters of the shaft d_A и d_B in the supports *A* and *B* are determined according to the formula (Sokolovski, 2007):

$$d_A = d_B \geq \sqrt[3]{\frac{M_{eB}}{0{,}1[\sigma_b]}}, \qquad (9)$$

Where M_{eB} is the equivalent moment in the support *B*, N.m,

$$M_{eB} = \sqrt{M_{bB}^2 + M_{t2}^2}. \qquad (10)$$

The construction of a part of the cutting mechanism (the main shaft with the bearings, the lower feed wheel and the driven belt pulley) is given on fig. 5. The lower feed wheel 1 is fixed on two radial ball bearings mounted on the flange 2. The flange 2 is set firmly on the bearing case 4 where by means of the plate 3 and bolts it is fixed to the body of the band saw. The driven V-belt pulley 6 is fitted on the right end of the shaft 5. The coupling between the shaft 5 and the feed wheel 1 is made with the help of a slot joint and the coupling between the shaft and the belt pulley 6 is realized by a key joint.

In this construction the shaft 5 is of a smaller diameter compared to the traditional one described in the precedent chapter. The boss of the lower feed wheel 1 is increased, the number of the bearings is doubled and it is necessary to make a supplementary flange 2. The design of the driven belt pulley 6 is not changed.

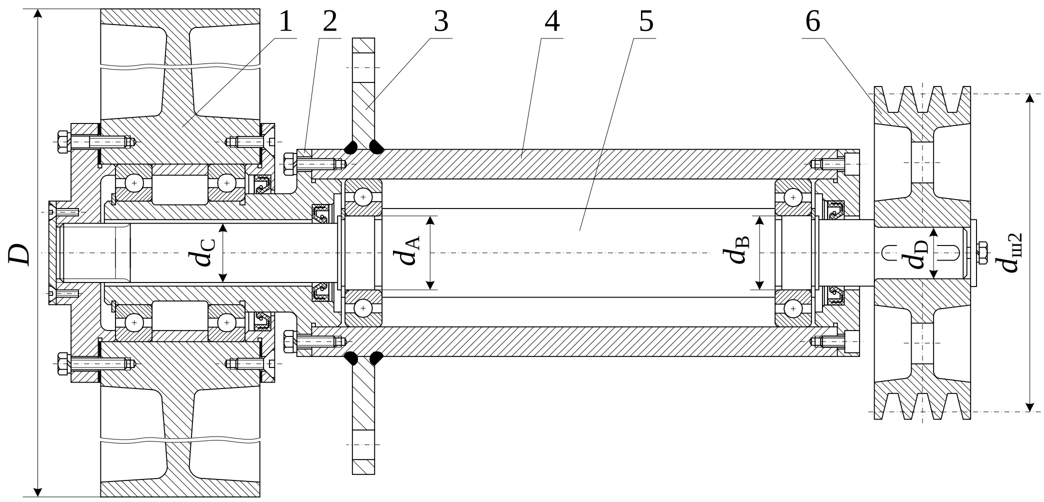

Figure 5: Construction of the main band saw shaft submitted to bending by the tension force of the belts and to twisting

2.3. Bending moments provoked by the tension forces of the saw band and of the belt transmission are absorbed by the case of the bearing body.

In order to decrease the bending a scheme in which the bending to be absorbed by the bearing body or by the body of the band saw (fig. 6) is proposed. More often the bearing cases are fixed to the body of the band saw by means of a bolt joint. On the entire length the shaft is loaded only by torsion of the turning couple M_{t2} (fig. 6c), determined with the help of formula (5) and its minimum diameter is specified according to the equation (6). The design of the shaft is of a shoulder-like form shown on fig. 6d.

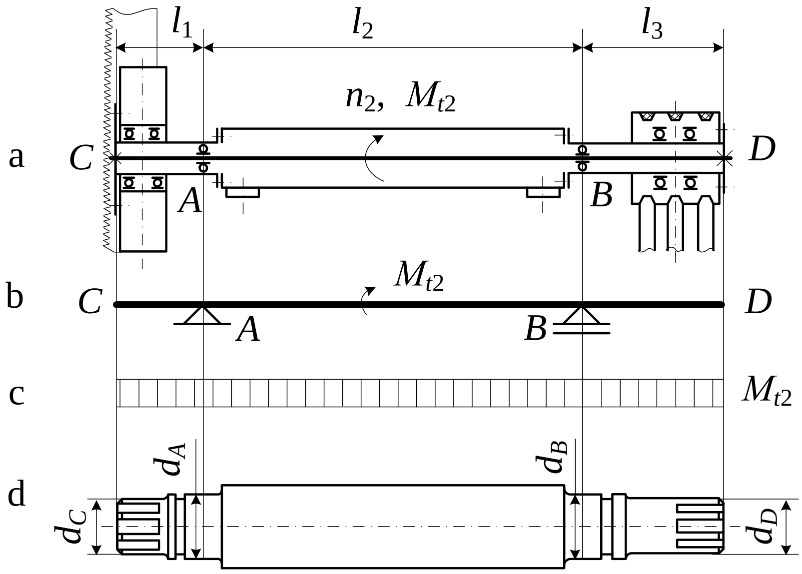

Figure 6: Calculation scheme and diagrams of the torque and the construction of the main band saw shaft

The construction of a part of the cutting mechanism (the main shaft with the bearings, the lower feed wheel and the driven belt pulley) is given on fig. 7. At this solution the bearing fixing of the lower feed wheel 1 and the driven belt pulley 6 are separated from the bearing fixing of the shaft 5. The bearing knot of the lower feed wheel is consisted of two radial ball bearings, the flange 2 which is mounted to the case of the bearing 4 with bolts and the low feed wheel 1. The bearing body 4 consists of a thick-section tube which through the welded plate 3 is connected to the body of the band saw by means of a bolt joint.

The bearing knot of the driven belt pulley 7 is consisted of two radial ball bearings, the flange 6 which is fixed to the bearing body 4 with bolts and the pulley 7. The coupling of the shaft 5 with the feed wheel 1 and with the belt pulley 7 is made by a slot joint. The shaft 5 is fixed on two radial ball bearings disposed in the bearing body 4.

CONCLUSION

The realized strength calculations of the main band saw shaft allow making an analysis of the critical sections of the shaft. The results of the analysis give reasons to seek possibilities for decreasing the load of the main shaft of band saw machines. Two new variants of disposing the bearings knots of the shaft are proposed. They could be used at the development of a new band saw design which will contribute to the improvement of the operation of the band saw cutting mechanism.

The implementation of the proposed solutions will lead to the increase of the reliability of the band saw machines.

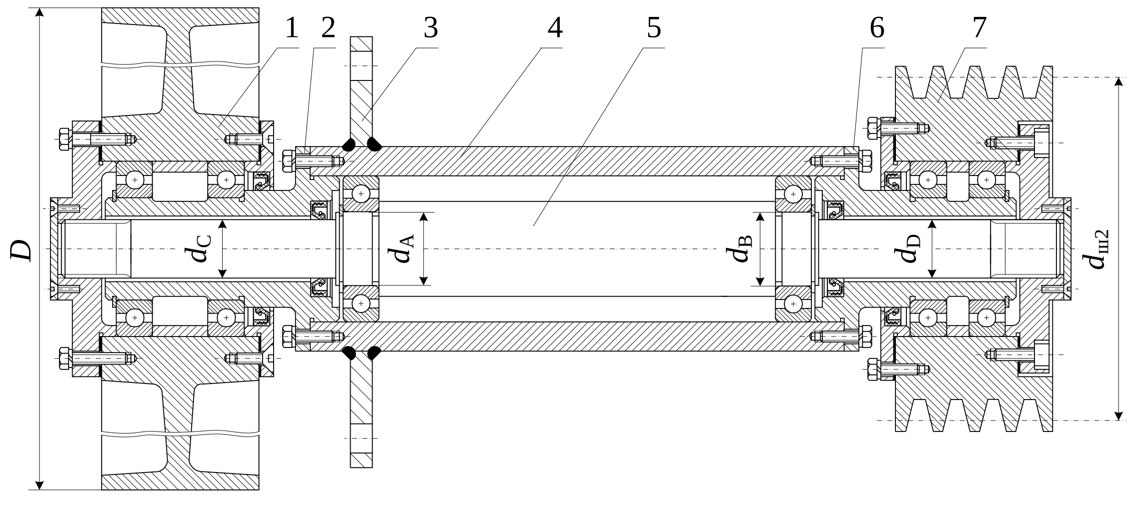

Figure 7: Construction of the main band saw shaft submitted to torsion

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