A METHOD OF DETERMINING THE PRIMARY RESISTANCE OF A BELT CONVEYOR WITH A PARTICULAR CONSIDERATION OF BELT PROPERTIES

Lech Gladysiewicz, Monika Hardygora and Witold Kawalec

INTRODUCTION

The designing of a modern belt conveyor should be supported by advanced calculation methods of its resistance to motion. Such methods have been developed upon the theoretical analysis of physical phenomena that occur in the running belt conveyor. Such analysis is based on the detailed investigations of the movement of a belt supported by idlers, followed by appropriate laboratory tests and in-situ measurements. The approach allows to identify the impact of all important parameters (like mechanical properties of a belt and transported bulk solid material) on the resistance to motion with the use of physical equations of loads, tensions, displacements and deformations rather than dimensionless, controlled by empirical constants, equations that were typical in earlier, simplified calculation methods. As a result, the theoretically proven and verified by test calculation methods of belt conveyor resistance to motion are capable to deal with any designed belt conveyor installation. Though conveyors usually consist of standard elements each new installation is unique and requires multivariate calculations of all considered, optional solutions to get the best one. This is the only approach that ensures the proper design of a conveyor since the entry phase of the project.

In the Institute of Mining Engineering of the Wroclaw University of Technology the investigations in the field of belt conveyors and belts have been carried out for many years [1, 2]. The original method of calculation of belt conveyor main resistance to motion has been developed on the basis of theoretical analysis of the energy dissipation processes in a conveyor belt and in the material load stream, as well as the analysis of the interaction between the belt and idlers. Based on the obtained algorithm, an “in-house” computer program has been written used both for consultancy purposes at the University and for operational use by several Polish open cast lignite mines and belt conveyor equipment suppliers [3]. Among others tasks the program has been employed for calculating the innovative projects like the first in Poland pipe-belt conveyor, the curvilinear conveyor in the lignite pit “Turow”, boosted conveyors with additional belt drive and an all-new modular belt conveyor with several compact drive units distributed along the bottom belt and drive transfer from bottom to top belt through the middle (drive transferring) roller of the upper idler (a solution for low underground transportation ramps) [4,5,6,7].

METHOD OF CALCULATIONS THE MAIN RESISTANCE TO MOTION

Main resistance force is prevailing for conveyors longer than 80m. Therefore the accuracy of calculations of the main resistance force components is vital for the proper choice of a belt conveyor design. Taking into account the energy dissipation processes in a conveyor belt and in the material load stream the following components of the main resistance force are considered:

– idler rotational resistance, W_k
– indentation resistance, W_e
– flexure resistance of a belt, W_b
– flexure resistance of bulk material, W_f
– sliding resistance of a belt on idlers, W_r

The components of the main resistance force are determined for each idler set for both upper and bottom strand.

The *idler rotational resistance* is calculated with the use of empirical formulae. However the scatter of results of direct laboratory tests of the rotational resistance performed for inspecting lots of rollers often reaches up to 100%. Therefore, average values are mere approximations of a rotational resistance of a single roller. However, when applied to the whole conveyor with some 250 to 360 rollers for each 100 m of its route, the average values provide sufficient approximation of the total rotational resistance force of all idlers. For a single roller this force can be calculated as a function of a belt speed $\boldsymbol{v_t}$ and an ambient temperature $\boldsymbol{T_C}$ (measured in °C) [3]:

$$W_{k1} = C_T \cdot (a_l + b_l \cdot v_t) \tag{1}$$

where the empirical, dimensionless factor $\boldsymbol{C_T}$ represents with satisfactory accuracy the influence of the ambient temperature $\boldsymbol{T_C}$ [3]:

$$C_T = \exp\left(0{,}405 - 0{,}025 \cdot T_C + 0{,}00026 \cdot T_C^2\right) \tag{2}$$

The idler rotational resistance is a sum of values of a rotational resistance of each roller (depending on the idler set layout). The coefficients of the rotational resistance $\boldsymbol{a_l}$ and $\boldsymbol{b_l}$ are calculated as mean values of test results made for a given type of a roller.

The *indentation resistance* is determined upon the theoretical analysis of cyclic, viscoelastic deformations of a bottom cover and a carcass of a belt. These deformations are caused by radial forces on rollers. The formula (3) describes the indentation resistance of a single roller [3].

$$W_{ei} = 0{,}463 \cdot \psi_e \cdot \sqrt[3]{\frac{R_i^4}{D_K^2 \cdot l_{ki} \cdot c_e}} \tag{3}$$

where: $\boldsymbol{\psi_e}$ – damping factor of belt subjected to cyclic compression, $\boldsymbol{R_i}$ – normal reaction on a given roller (radial force) [N], $\boldsymbol{D_K}$ – roller diameter [m], $\boldsymbol{l_i}$ – length of the contact zone between the belt and the given roller [m], $\boldsymbol{c_e}$ - elementary transverse flexural rigidity [N/m^3].

The indentation resistance of the whole idler set is a sum of values of an indentation resistance of each roller.

Following Greune, Hager, Hintz [8, 9], the impact of an ambient temperature $\boldsymbol{T_C}$ on the indentation resistance on an idler set can be taken into account with the use of the empirical function:

$$W_e = W_{e20} \cdot \left(1{,}18 \cdot 10^{-4} \cdot T_C^2 - 0{,}0118 \cdot T_C + 1{,}189\right) \tag{4}$$

where: $\boldsymbol{W_{e20}}$ - indentation resistance of an idler set following (3) at the ambient temperature 20 °C [N], $\boldsymbol{T_C}$ - ambient temperature [°C].

The indentation resistance depends on belt parameters, roller diameter and radial force on a roller (roller load). The indentation resistance of the whole idler set requires the identification of reactions on each roller. In the developed method radial reactions on rollers are determined as components of belt and material load and reactions caused by forces resulting from horizontal or vertical curvature of a conveyor route. Also the unsymmetric idler load caused by belt misalignment (either on straight or curvilinear routes with horizontal curves) are taken into account. The indentation resistance following (3) heavily depends on belt parameters: damping factor $\boldsymbol{\psi_e}$ and elementary transverse flexural rigidity of a belt $\boldsymbol{c_e}$. Both have to be identified from tests with the use of a special test rig [10].

The force of *flexure resistance of a belt* $\boldsymbol{W_b}$ arises on each idler. This force performs a work of deformation of a belt on a distance $\boldsymbol{l_g}$ (the spacing between consecutive idlers) which balances the energy dissipation $\boldsymbol{E_{nb}}$ (see fig.1):

$$W_b \cdot l_g = \psi_b \cdot E_{nb} \tag{5}$$

where: $\boldsymbol{\psi_b}$ – damping factor of belt subjected to cyclical bending (different from $\boldsymbol{\psi_e}$); it is identified from tests on a special test rig [10].

Hence the flexure resistance of a belt is calculated from the equation [3]:

$$W_b = \psi_b \cdot \frac{E_{nb}}{l_g} \tag{6}$$

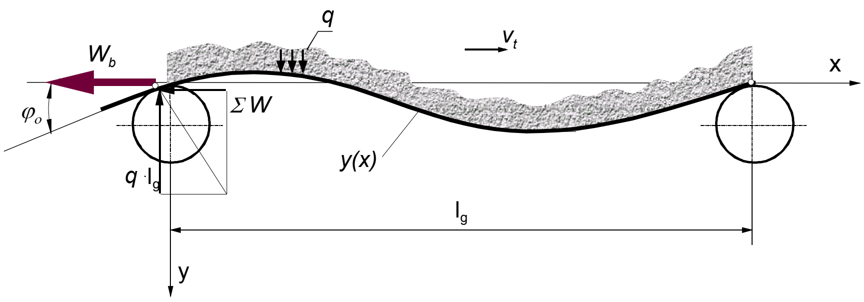


Fig. 1. Scheme of bending of a belt between two consecutive supporting idlers [3]

The strain energy of bending of a belt treated as a linear elastic beam supported by idlers with a constant flexural rigidity ***EJ*** over its length $\boldsymbol{l_g}$ is calculated from the integral [3]:

$$E_{nb} = \frac{q \cdot l_g}{2 \cdot EJ} \cdot \int_0^{l_g} M^2(x) \cdot \mathrm{d}\, x \qquad (7)$$

The strain energy of bending is calculated with regard to transverse flexural rigidity of a belt ***EJ*** resulting from idler troughing angle. If any misalignment of a belt occurs then the cross section of a belt on idlers is unsymmetrical which influences on the flexure resistance of a belt.

The damping factor $\boldsymbol{\psi_b}$ has to be calculated with the use of supplementary, empirical formulas developed for textile and steel-cord belts.

The *flexure resistance of bulk material* $\boldsymbol{W_f}$ results from cyclic deformations of material stream. These deformations are strictly connected with the bending line of a belt between idlers and therefore in previous methods of calculations of main resistance to motion both flexure resistances: of a belt ($\boldsymbol{W_b}$) and of a bulk material ($\boldsymbol{W_f}$) used to be identified as a single component from an approximated formula. Assuming that transported bulk material is a kind of grainy medium, two zones of movement of a belt with the material can be identified (see fig.2):

– active zone, where the material stream causes deformations of a belt (bending and an increase of an actual throughing angle),
– passive zone, where deformations of belt cause deformations of a material stream.

The length of each zone equals half of the distance between idlers $\boldsymbol{l_g}$.

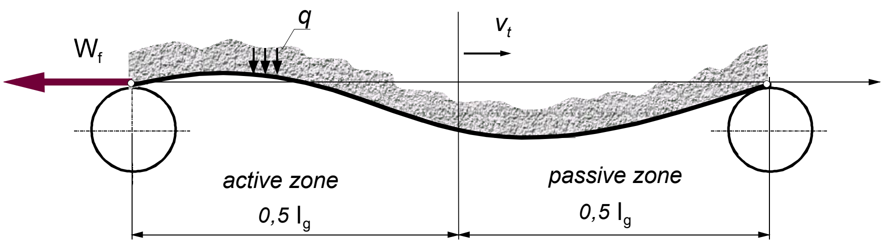


Fig. 2. Scheme for identification of the flexure resistance of bulk material [3]

Elementary forces that cause the deformations of a belt together with transported material stream are linear loads ***q*** uniformly distributed over the single distance between idlers. Within the passive zone the work of loads on the deformations ***y(x)*** equals:

$$E_{nf} = q \cdot \int_0^{0{,}5 \cdot l_g} y(x) \cdot \mathrm{d}\, x \qquad (8)$$

The bending lines of a belt ***y(x)*** assumed for calculations the flexure resistance of bulk material and the flexure resistance of a belt are the same. Assuming that streams of energy of both separated zones (fig.2) are proportional to pressure between the belt and the transported material, following the grainy medium mechanics with regard to energy dissipation the flexure resistance of bulk material is expressed [3]:

$$W_f = \psi_f \cdot \frac{E_{nf}}{l_g} \tag{9}$$

where $\boldsymbol{\psi_f}$ –damping factor of material is identified as a function of internal friction angle of material $\boldsymbol{\varphi_w}$ (in degrees or radians) [3]

$$\psi_f = 1 - \frac{\tan^2\left(\frac{\pi}{4} - \frac{\varphi_w}{2}\right)}{\tan^2\left(\frac{\pi}{4} + \frac{\varphi_w}{2}\right)} \tag{10}$$

The *sliding resistance of a belt on idlers* $\boldsymbol{W_r}$ consists of two components [2]. The first is caused by forward tilt of side rollers while the second is a result of belt misalignment and its transverse movements on idlers. The sliding resistance of a belt on idlers can be combined from friction forces on rollers which are calculated from their radial reactions (identified for indentation resistance) and kinetic friction factor between rollers and the belt.

The assessment of the impact of belt misalignment is more difficult as it depends on the random operating conditions of a conveyor like lateral and vertical tilt, lateral offset and horizontal skew of idlers or poor belt or bulk stream tracking. However, these parameters cannot be measured or predicted precisely so the belt misalignment can only be assessed.

A practical solution can be found with the use of fuzzy - not exact – numbers representing all mentioned errors of proper alignment of a conveyor route and belt tracking [11]. Since the fuzzy set theory was presented, it has been widely used when neither the exact values nor the random distribution of given parameters are known. Fuzzy numbers are examples of fuzzy sets. The most common are triangular and trapezoidal fuzzy numbers. The shape of the membership function (triangular or trapezoidal) of fuzzy numbers of the specific parameters have been modelled following a series of “in-situ” observations and measurements to provide proper representation of the real values.

Finally, the sliding resistance of a belt on a given idler caused by belt misalignment can be calculated with the use of actual exact numbers – values that are individually generated from fuzzy numbers (fuzzy sets) with regard to their membership functions. The summarised results from a series of such calculations made for consecutive idlers for the whole conveyor is a simulation of an actual total sliding resistance force. The obtained mean value has been found as a realistic assessment of the sliding resistance force.

Taking into account all main resistances to motion components helps an engineer to analyse the belt tension. The impact of belt tension on the value of the main resistance to motion becomes more important for longer conveyors where the belt tension changes within a wide range (see fig. 3).

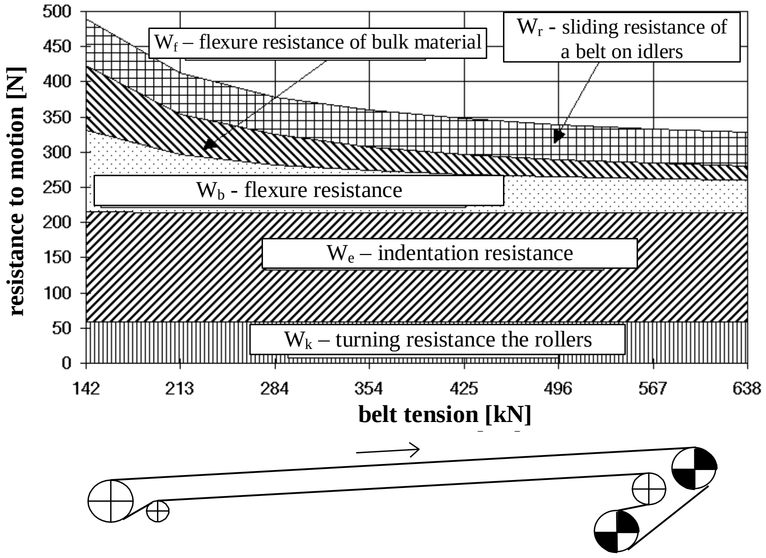


Fig. 3 Main resistances to motion along the top belt of a typical belt conveyor from a lignite open pit: transported material: overburden (density ρ=1600kg/m^3), length L=1100 m; lift H=10 m, belt width: B=2,25 m, speed: v_t=5.24 m/s, steel cord belt (strength: 3150 kN/m), idler (garland type, well maintained) spacing on carrying side l_g=1,0 m, troughing angle λ=45°, ambient temperature T_C=0ºC, two drive pulleys at the head station; 4 drive units rated 1000 kW each [3]

During steady operating conditions of the investigated conveyor (fig.3) the belt tension incremental changes from 142 kN at the tail pulley up to 638 kN at the head. The idler rotational resistance and the indentation resistance are constant while flexure resistance of a belt and bulk material and sliding resistance of a belt on idlers depend on the belt tension. Garland type idlers which are commonly used and their effective tilt angle is a result of interaction between idlers and belt. The presented method employs the actual resistance to motion forces to calculate the tilt angle in order to identify the sliding resistance of a belt on idlers.

The comparison of measured and calculated operating main drive power of the declined belt conveyor in a Polish open pit lignite mine is presented on figure 4. This conveyor had to be driven for a capacity up to 1000 m^3/h and retarded for greater capacity as shown on the chart. The calculations of the effective tilt angle of garland type idlers have allowed to compute the sliding resistance of a belt on idlers precisely. The “own method” results have matched the measurements figures within satisfactory accuracy especially for bigger actual capacity where the standard DIN 22101 method (coefficient ***f*** proven for standard, horizontal belt conveyor) has given poorer required drive power assessment.

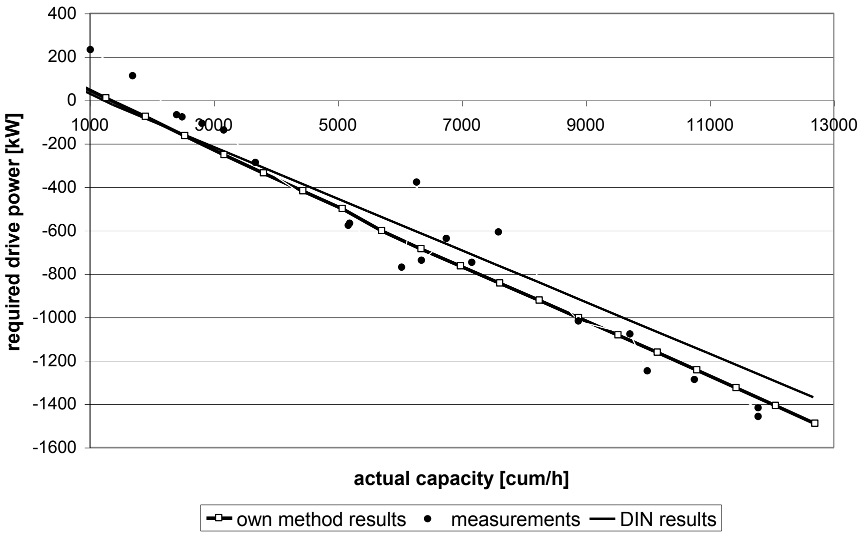


Fig. 4 Measured and calculated operating main drive power of the declined belt conveyor from a lignite open pit: transported material: overburden, angle of decline δ=-7,9°, belt width: B=2,25 m, speed: v_t=5.24 m/s, steel cord belt (strength: 3150 kN/m), idler (garland type, well maintained) spacing on carrying side l_g=1,2 m, troughing angle λ=45°, two drive pulleys at the tail station; 4 drive units rated 1000 kW each [3]

THE POSSIBILITIES OF DECREASING THE MAIN RESISTANCE OF A BELT CONVEYOR

The exact methods of calculating the main resistance force allow to detailed investigation of the influence of selected parameters of belt conveyors in order to identify the most influential factors and analyse possible improvements to obtain energy-saving solutions for belt conveyors. Assuming that combined idler rotational resistance and the indentation resistance share up to 70% of the total resistance force the biggest energy savings are expected there.

The rotational resistance is measured on a special test rig that has been built in a laboratory of the Institute of Mining Engineering [11, 12]. On the rig the axle of a roller is driven while the tube is fixed with a lever equipped with force measuring gauge to identify the rotational resistance of the roller. However, the quality of idlers requires maintaining the rotational resistance over the roller lifetime within acceptable limits. The changes of the rotational resistance are recorded with the use of a special fatigue test rig, where equivalent radial loadings of idlers (causing the real idler wear and loading) are set upon the statistical analysis of random loading on the conveyors. The difficult operating conditions (typical for underground mines) are also simulated by regulated moisture and dustiness in the test chamber. The rotational resistance is then measured in given intervals of the simulated lifetime of idlers (usually between 3 and 5 years) to provide the comprehensive characteristic of an idler.

The durability characteristics of pre-selected idlers supplied by 4 independent manufacturers (each meets the assumed level of turning resistance of up to 2 N per single roll at the beginning of its use as required by DIN standards) is shown on figure 5.

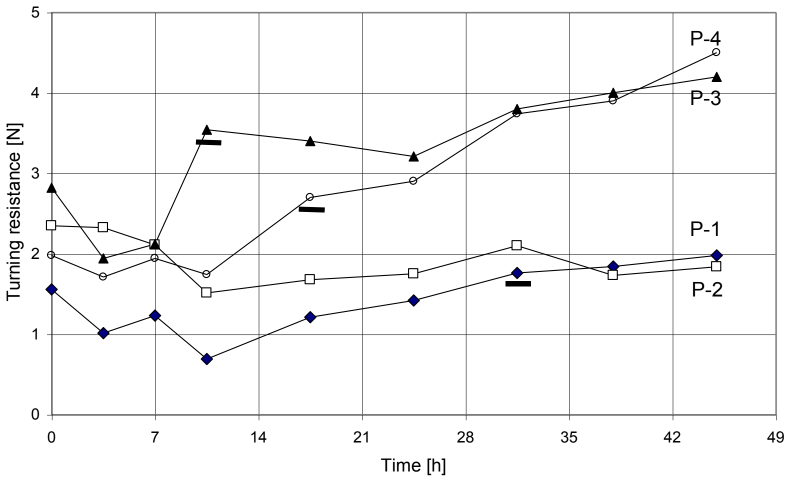


Fig. 5. Measured changes of rotational resistance of selected rolls (P1-P4) over their expected lifetime; underlined points represent the breaking points of a preliminary value of rotational resistance, 7 hours of testing represent 1 year [12]

Idlers P-1 and P-2 maintain the lowest turning resistance but the P-2 breaks its preliminary value of rotational resistance in 7 years while the P-1 more than 2 years earlier (which points on its shorter expected durability). The rotational resistance of idlers P-3 and P-4 rises after mere 20-30 months despite the lack of other symptoms of their excessive wear.

The obtained results of performed tests have proven that the rotational resistance of newly installed idlers does not guarantee maintaining the low level of the total resistance to motion of a conveyor and the special durability tests of idlers are necessary for safe long-term use a conveyor.

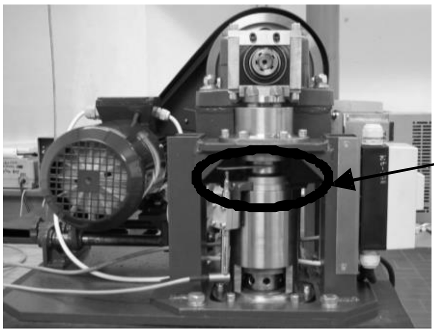
The investigation of idlers has proved that the reduction of the rotational resistance can be obtained through introduction of suitable changes of working tolerances, improvement of setting the bearings and optimised grease choice. The greatest effect, however, has been obtained after the replacement of traditional the standard labyrinth seal with the innovative ferromagnetic seal [13, 14].

The *indentation resistance* depends mostly on conveyor belt properties. The proper choice of material chosen for rubber bottom covers of belt can significantly decrease the indentation resistance. It is important to maintain the optimal parameters values within the wide ambient temperature range of -25ºC to +30ºC (figures valid for Poland).

Following (3) the indentation resistance depends on two parameters of a belt: damping factor of belt subjected to cyclic compression $\boldsymbol{\psi_e}$ and the elementary transverse flexural rigidity $\boldsymbol{c_e}$. The prevailing properties of rubber mix of bottom cover rather than of a belt carcass. The properties of various rubber mixes have been investigated on a special test rig in a climatic chamber where the operating conditions of a belt have been simulated by cyclic pressing (see fig. 6). The strength parameters of the tested specimens are presented in table 1.

Table 1. The strength parameters of the tested specimens of bottom covers rubber [15]

rubber mix symbol	unit	A	B	C	D	E
Tensile strength	MPa	20,3	16,7	20,7	18,3	24,3
Ultimate elongation	mm	680	421	643	475	655
Abrasion resistance	mm^3	107	120	96	105	95
Hardness	ºSh A	45	54	48	55	50



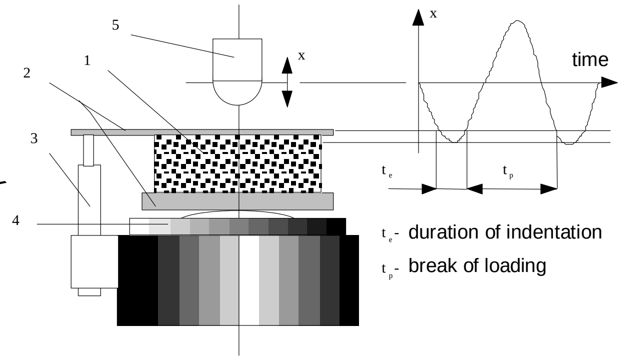


Fig.6. Test rig for examining rubber specimen under cyclic loading [15]

Rubber specimens 1 (in fig.6) in the form of 6, 10 and 12 mm thick disks with a diameter of 46 mm have been used. They have been placed between two metal plates (2). The bottom plate is supported by a force strain gauge (4) to measure the pressing force ***σ(t)*** of the punch (5). The deformations ***ε(t)*** of the specimen are measured by an induction displacement gauge (3). The cyclic pressing of the belt rubber cover is simulated by the to-and-fro motion of the punch touching the plate (2) in its lower dead position only for a short time $\boldsymbol{t_e}$. (see chart of loading on fig.6). The hysteresis loop of rubber ***σ = f(ε)*** has been recorded to be further transformed to the damping factor $\boldsymbol{\psi_e}$ and the modulus of elasticity $\boldsymbol{E_e}$ of tested rubber covers subjected to cyclic compression. Then the elementary transverse flexural rigidity $\boldsymbol{c_e}$ is identified.

Basing upon the laboratory test results of parameters of various rubber mixes, the expected indentation resistance force of belt conveyors from open pit lignite mines has been calculated (see fig.7).

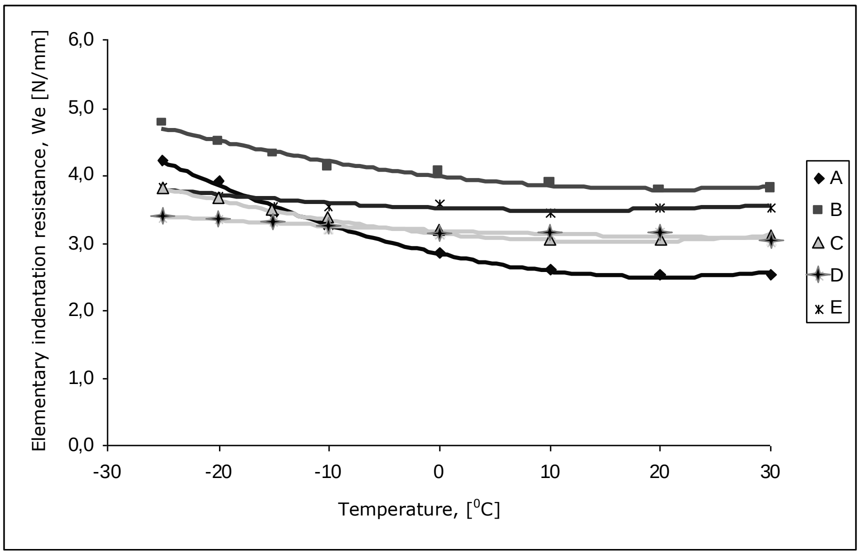


Fig.7. The indentation resistance $\boldsymbol{w_e}$ of bottom belt covers (10 mm thick) as a function of ambient temperature (loading 1 kN, frequency of cyclic pressing 1 Hz) [15]

Tests have shown the strong dependency of parameters of rubber on an ambient temperature. Both the damping factor and the modulus of elasticity of rubber increase by up to 50% when the temperature lowers. The winter temperatures cause the increase of indentation resistance force and, consequently, the total main resistance force. The rubber mix ***A*** has been chosen for a prototype energy-efficient belt because it maintains the smallest elementary indentation resistance with regard to the distribution of an ambient temperature over a year in Poland.

Another solution of avoiding the increase of indentation resistance is to use thinner and abrasion-resistant bottom covers.

CONCLUSIONS

1. The presented method of analytical identification of components of main resistance to motion of a belt conveyor due to taking into account all significant parameters allows to make a multivariate analysis in order to decrease the driving force of a belt conveyor.
2. The proper idlers selection together with careful choice of conveyor belt parameters are the key factors to obtain the energy-saving solutions.
3. The presented method of calculations the main resistance to motion of a belt conveyor consists of set of algorithms developed not only for calculations the specific main resistance components but also several required supplementary parameters (the bending line of a belt between idlers, flexural rigidity of a belt, reactions on idlers, etc.) which have been implemented into a "in-house" software program.

REFERENCES

1. Gladysiewicz L. Gollasch J., Vefahren zur Ermitteln der Reibungswiderstände an Gurtförderern, Hebezeuge und Fördermittel 1991, Heft 12.

2. Gladysiewicz L. Gollasch J., Vefahren zur Ermitteln der Anteile des Walkwiderstandes eines Gurtförderers, Hebezeuge und Fördermittel 1992 ,Heft 1.

3. Gladysiewicz, L., Belt conveyors – theory and calculations. Wroclaw University of Technology, (in Polish)

4. Gladysiewicz L., Hardygóra M., Kawalec W., Belt conveying in the Polish mining industry - selected topics, 2004, vol. 24 nr 4/5, s 236-242, Bulk Solids Handl.

5. Gladysiewicz, L., Kawalec, W., Object modelling of a belt conveyor. Mine Planning and Equipment Selection 2000. Athens

6. Gladysiewicz, L., Kawalec, W., Energy-saving solutions for belt conveying. Mine Planning and Equipment Selection 2002, Prague-Bouzov

7. Gladysiewicz, L., Kawalec, W., Non-uniform drive distribution on a conveyor with belt booster drive. Transport Przemyslowy, 3(17)/2004, p.6-10 (in Polish)

8. Greune A., Energiesparende Auslegung von Gurtförderanlagen Diss, Universität Hannover 1989 (not published)

9. Hager M., Hintz A., The Energy-Saving Design of Belts for Long Conveyor Systems, Bulk solids handling.4/93, Trans Tech Publikations. Clausthal-Zellefeld, Germany, ISSN 0173-9980

10. Gladysiewicz, L., Methode zur Ermittlung der Hauptwiderstaende des Gurtbandfoerders unter besonderer Beruecksichtigung der Gurteigenschaften. (Thesis), Bergakademie Freiberg, (in German)

11. Gladysiewicz L., Kawalec W., Modelling chosen operating conditions of a belt conveyor, Proceedings "Mine Planning & Equipment Selection", New Delhi

12. Gladysiewicz, L., Krol, R., Comparative quality analysis of idlers used in underground mines. Transport Przemyslowy, 1(23)/2006, p.8-13 (in Polish)

13. Gawlinski M., Gladysiewicz L., New generation of seals for bearings nodes in conveyor idlers, 2004, vol. 24 nr 4/5, s. 244-246, Bulk Solids Handl.

14. Gladysiewicz L., Krol R., Modernisierung der Tragrollen zur Verringerung des Tragrollen-laufwiderstands, Neue Trends in der Anlagenentwicklung. 10. Fachtagung Schuettgutfoerdertechnik 2005. Magdeburg, 28. und 29.O9.2005

15. Bajda M., Hardygora M., Effect of temperature on the properties of a conveyor belt cover, Proceedings “Mine Planning & Equipment Selection”, 2006, Torino

AUTHORS’ CV

Lech Gladysiewicz, Professor at the Wroclaw University of Technology (Poland), Dean of Faculty of Geoengineering, Mining and Geology, mechanical engineer, a specialist in the field of belt conveyor calculations and transportation systems equipment selection, author and co-author of some 120 published papers.

Monika Hardygora, presenter of the paper, Professor at the Wroclaw University of Technology (Poland), Vice President of the University, mining engineer, a specialist in the field of belt conveyor technology, especially conveyor belts, manager of the certified belt conveyor laboratory; author and co-author of some 160 published papers.

Witold Kawalec, PhD, mathematician and mining engineer, lecturer and research worker in the Institute of Mining Engineering at the Wroclaw University of Technology (Poland), main fields of interest: mathematical modeling of systems in mining with special focus on computations of belt conveyors and applications of computer methods in mining engineering; author and co-author of some 60 papers.