This book is focused on the underground mining and provides basic introduction into this problem from the engineering point of view (i.e. coal ploughs, longwall shearers, conveyors, support systems, design, control, strength analyses, reliability assessments, deterministic and probabilistic approach, etc.).

Tato kniha se zaměřuje na hlubinné hornictví a poskytuje základní informace o této problematice z pohledu inženýra (tj. uhlenné pluhy, kombajny, dopravníky, výztužné systémy, konstrukce, řízení, pevnostní analýzy, posudky spolehlivosti, deterministický a pravděpodobnostní přístup atp.).

**KEYWORDS**

underground mining, technology, excavating machines (coal ploughs, longwall shearers), conveyors, support systems, design, control, strength analyses, reliability assessments (deterministic and probabilistic approach)

hlubinné hornictví, technologie, dobývací stroje (uhlenné pluhy, kombajny), dopravníky, výztužné systémy, konstrukce, řízení, pevnostní analýzy, posudky spolehlivosti (deterministický a pravděpodobnostní přístup)

**AUTHORS‘ NOTE**

This book was compiled by authors from the “Department of Production Machines and Design” as well as the “Department of Mechanics of Materials” (Faculty of Mechanical Engineering, VŠB - Technical University of Ostrava) and is focused on problems of mining industry.

The book is dedicated to students, academic staff, engineers and technicians and contains up-to-date information from the branch of mining.

This book provides information on mines and machinery in underground coal longwalls, in particular:

- Excavating machines (coal ploughs and longwall shearers)
- Armoured face conveyors
- Support systems in the coal faces (individual supports and powered supports)
- Example from strength analyses (stress calculation, statics, dynamics, analytical approaches, numerical methods - FEM, probabilistic reliability assessment – SBRA Method, etc.)

During the compilation of this book, current knowledge on machine equipment from the Czech, Slovak, German, Polish, Australian and US mines and companies have been used. Therefore, some documentation has been used with courtesy of producers mentioned in the text; such as OSTROJ OPAVA, a.s., BUCYRUS Lünen (i.e. former DBT Gmbh), OHR, s.r.o. Opava; FAMUR s.a. Katowice, EICKHOFF Hochum, RYFAMA a.s. Rybnik, Hornonitrianske Bane zamestnanecká a.s. (BME Nováky).

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INTRODUCTION

As civilizations began to use and depend on coal, metals, rock-oil, etc., the amount of sources found right at the surface was not enough anymore. Other ways of obtaining raw materials had to be developed. Hence, mining allows you to find and mine minerals, ores, and stones and other raw materials from resource nodes scattered throughout the world. It is one of the primary professions in our society.

Mining techniques are divided into two types:

- surface mining (this book is not focused on it),
- underground mining.

Underground mining is carried out when the rocks, minerals, or gemstones are located at a distance far beneath the ground to be extracted with surface mining. To facilitate the minerals to be taken out of the mine, the miners construct underground rooms to work in. The mining company selects the best feasible way to get the minerals extracted out. Most mining is carried out by using continuous mining that employs a continuous mining mechanism to cut the coal deposits from the walls. This means there is less of blasting and drilling and utilizes fewer miners down in the mines.

Coal is usually known as a fossil fuel formed as a result of the thriving ecosystem over the years where plant remains were hoarded by water and mud from the process of oxidation of the material and constant biodegradation.

The world of today (i.e. its level) could hardly exist without mining industry. The mining industry will continue to be an important support to the economy. Therefore, this book is dedicated to underground mining and is suitable for students, academics and experts in this branch.

1. LONGWALL SHEARERS

1.1 Longwall Mining

Longwall mining is a form of underground coal mining where a long wall of coal is mined in a single slice (typically 1 – 2 m thick). The longwall panel (i.e. the block of coal that is being mined) is typically 3 – 4 km long and 250 – 400 m wide. The basic idea of longwall mining was developed in England in the late 17th century. For more information see web page: http://en.wikipedia.org/wiki/Longwall_mining.

The coal is cut from the coalface by a machine called the longwall shearer (power loader), see Fig. 1.1. This machine can weigh 75 – 120 tonnes, and comprises of the main body, housing the electrical functions, tractive motive units to move the shearer along the coalface and pumping units (to power both hydraulic and water functions).

For longwall mining, a number of hydraulic powered supports called jacks, chocks or shields placed in a long line, side by side for up to hundreds of metres in length are used in order to support the roof of the coalface.

1.2 Types of Longwall Shearers

History of longwall shearsers:

1. At first, these were coal-cutting machines which used chain cutter sections to break coal; later on, these were combined mining machines which had cutter sections made by a combination of chain and hewing elements. These machines had a small output; they were faulty and their life-span was short.

2. Another stage of development was a longwall shearer with a cylindrical cutter section whose cutter bits fixed on a rotating drum cut coal from the pillar, the principal of which has been used to date.

Direct-on-face longwall shearsers can be divided according to several aspects, especially with regard to the mining technology and the machine's structural design:
a) As per the mining technology:
- single directional longwall shearers,
- bidirectional longwall shearers.

b) As per the number of cutter sections:
- single-drum longwall shearers,
- double-drum longwall shearers.

c) As per the travel method in the coal faces:
- longwall shearers with a drag chain,
- longwall shearers with a chainless travel.

d) As per the regulation type of travelling speed:
- hydraulic,
- electric.

1.3 Main Parts of Longwall Shearers

Longwall shearers (Fig. 1.2) must cut off coal in the face, travel along the face and load the loose coal onto an armoured face conveyor. Apart from these main functions, it must be equipped with a dust suppression system during cutting off the coal, prevention from the rock getting in the working area and barriers against coal shooting off the cutter sections.

Fig. 1.2 Longwall Shearer (see http://www.usmra.com/repository/category/mining/wallpaper).

1.4 Longwall Shearer Frame

The longwall shearer frame is made of a thick bed plate with large brackets for pivotally attached arms, travel wheels, crank blocks for the shearer’s travel on the conveyor, or the floor and also by brackets for the arm position control device. Individual control system blocks of the shearer are attached and interconnected on the frame, such as the cutter sections drive, travel drive, electrics and arm position control.

The cutter section drive is composed of electromotor and gear. The electromotor is placed on the frame, or in the arms.

When the electromotor is placed on the frame of the shearer, both cutter sections are usually driven by one electromotor. As both cutter sections are never in gear simultaneously (one always cuts off the pillar and another one loads the broken coal onto the conveyor), this application is advantageous mainly due to the output of one electromotor being alternately transmitted onto one and then onto the other one cutter section. Nevertheless, the disadvantage is its very long transmission part from the electromotor to the cutter section as well as the fact that the electromotor takes up a considerable part of the centre of the shearer.
and thus makes it longer. With this arrangement, the electromotor also usually drives the arm position control part.

Currently, the most applied arrangement is the placement of the cutter section electromotors into the arms of the shearer. This arrangement is demonstrated in Fig. 1.4. The advantage of such arrangement is mainly the fact that the shearer is shorter and during a breakdown it is possible to change the electromotor quickly as it is easily accessible. The transmission from the electromotor to the cutter section is very short and an epicyclic gear is often used. However, each arm must have its own electromotor and also other devices of the shearer are driven by their own electromotors.

1.5 Longwall Shearer Travel

The travel drive of the shearer varies according to combined cutter-loaders with a drag chain and ones with a chainless travel. In terms of the combined cutter-loaders with a drag chain, the drag chain gear is placed either on the combined cutter-loader, or in the return station in the gate. In both cases, the travelling speed of the shearer is controlled in dependence on the cutter section resistance, either electro-hydraulically or electrically.

Nowadays, the combined cutter-loaders with a drag chain are applied only in the edge seams where there is no armoured face conveyor. In other circumstances, the shearer with a chainless travel is used, as in this case the greatest danger of the previous type, which is swinging of the chain during travel of the combined cutter-loader, is eliminated.

The travel drive of the shearer (with a chainless travel) transmits the output from the electromotor onto the travel wheel which engages into the steady rests connected to the armoured face conveyor.

As the travelling speed must be proportional to the resistance during breaking the coal in the pillar, it is controlled. There are two possible control ways of the shearer travelling speed – hydraulic or electric.

Shearer travelling speed with hydraulic regulation (control), between the drive electromotor and the pin travel wheel, there is a closed hydraulic circuit with a regulating pump. Regulating pump is in dependence on the load changes (supplied amount) and thus also the running speed of the hydraulic motor, which drives the travel wheel. A basic example of the hydraulic regulation is in Fig. 1.5.

Shearer travelling speed (with its electric control) makes use of the speed variation in the drive electromotor by means of a thyristor frequency converter. Between the electromotor and the travel wheel there is an essential mechanical transmission. Therefore, the travel speed varies in dependence on the resistance during breaking and the coal in the pillar is managed by the electromotor speed variation, i.e. automatically by thyristor frequency converters, from where the electric power is transmitted to the electromotor via a cable. The thyristor frequency converters are usually positioned in the gate and the transmission of power is secured by a trailing cable. Figure 1.6 depicts a simplified block diagram of the shearer travelling speed electric regulation.

1.6 Chainless Travel of a Shearer

A chainless travel of a shearer calls for a special arrangement of the armoured face conveyor end sheets. The travel wheel is engaged in the fixed elements connected to the face conveyor. Nowadays, it is based on the means the travel track on the conveyor is made of. The following travel systems are mostly used:

- Eicotrack,
- Dynatrace,
- Dynaride.
Eicotrack System has a vertical travel wheel on the shearer, which is in a contact with the bars with transversal rests substituting rack bars. The bars are joined with end sheets of the face conveyor. The bars are of a half length compared to the length of a conveying through. This arrangement permits to overcome floor irregularities in a better way, and it eliminates potential errors in the idler roll pitch. An example of an operational design of the Eicotrack travel system is in Fig. 1.7.

In addition, the travel drive is equipped with a braking system which permits the shearer to stop and therefore prevents its uncontrolled movement, especially in the sloped working faces. The Eicotrack travel system secures a continuous travel of the combined cutter-loader along the pillar.

The other two systems (i.e. Dynatrack and Dynaride) use specially customized chains for the travel of the drive travel wheel, which differ very slightly with both systems. The chains are anchored on the adjustable sheet of the scraper conveyor. In contrast to the previous system, the chain does not form a solid element as it is in the case of fixed bars of the Eicotrack travel track. The chain is flexible; it permits an autobalance of the travel track and a good engagement of the travel wheel. Examples of the operational designs of both systems are in Fig. 1.8.

1.7 Arm Position Control

The device for an arm position control of the combined cutter-loader is composed of a hydraulic unit, usually placed on the combined cutter-loader frame, electric-controlled hydraulic distributors and hydraulic cylinders, controlling the individual arms. The position of each arm is controlled individually. The set arm position is kept by a hydraulic lock of the lifting cylinder. The hydraulic unit is driven either by a separate electromotor or by an electromotor of one of the cutter-loader main functions (cutter section or travel). A simplified block diagram of a combined cutter-loader arm position control is in Fig. 1.9.

Fig. 1.7 Eicotrack Chainless Haulage System (1 – bar, 2 – travel wheel, 3 – armoured face conveyor, 4 – shearer guide bar, 5 – conveyor end sheet).

Fig. 1.8 Chainless Haulage System (1 – chain, 2 – travel (chain) wheel, 3 – armoured face conveyor).

Fig. 1.9 Block Diagram of an Arm Position Control Device (1 – hydraulic unit, 2 – hydraulic distributor, 3 – hydraulic cylinders with a lock, P – pressure run, T – return run).

The arms with cutter sections are of a box construction. Inside the arms there are mainly transmissions for the transfer and reduction of the torsional moment from the electromotor to the cutter section. In this case, the electromotor is placed on the combined cutter-loader frame, the transmissions are distributed along the whole lengths of arms. Locating the cutter section electromotor on the arm means that the transmissions are shorter. They often make use of an epicyclic gear.

The arms are pivotally attached to the combined cutter-loader frame and according to needs their position is changed by hydraulic cylinders placed between the combined cutter-loader frame and the arm construction.

1.8 Cutter Sections

At the end of the arms there are cutter sections. The basic task of the cutter sections is to remove the rock from the coal pillar and load it onto the face belt conveyor under a minimum power consumption and acquiring the suitable grain-size. The cutter sections are composed of three fundamental parts, see Fig. 1.10:

- cutter section body,
- cutter holder,
- cutters (i.e. cutting tools).

The cutter section bodies are usually weldments. Generally, they are made of a cylinder which is on the circumference fitted with double or four gauge helical tracks. The
spiral angle is selected in a way an optimum feed of mine run arrives along the cutter section axis onto the face belt conveyor.

The cutter holders are welded onto the helical tracks. In the spiral internal cavity there are pipes for water supply to the sprinkling jets.

The water supply into the cutter edges area significantly reduces the dustiness during rock breaking, see Fig. 1.10. The allocation of cutter holders on the cutter section depends on the used cutter type and breaking characteristics of rocks in the pillar. The cutter section hubs serve for fastening the cutter-loader arm output shaft. They also allow for the transmission of the sprinkling water from the arm hollow shaft into the cutter section body.

Significant properties and parameters of the combined cutter-loader cutter sections are as follows:
- good efficiency of breaking at a low power consumption,
- ability to well load the broken coal onto the conveyor,
- low dustiness during breaking,
- low wear and long life-span.

Typical example of a cutter section is shown in Fig. 1.11.

The width of a cutter section corresponds to the combined cutter-loader attack on the pillar and the most often it is 630 or 800 mm. The cutter section diameter depends on the effective thickness of the seam in which the combined cutter-loader is going to work. The lower effective seam thickness is pertinent to the lower diameter. Concerning the combined cutter-loaders for low-coal seams, the cutter section diameter is usually 800 – 1000 mm; concerning the combined cutter-loaders for high-coal seams, it is up to 2500 mm.

Cutters are the most loaded parts of the combined cutter-loaders and their expenditure in the course of operation is quite high. An optimum operating mode of a cutter can be reached both by their number on the cutter section and by a functional layout of the tool planes and the individual cutter order of engagement. A distributed load on all the cutters during their work is important in order to secure an approximately their life-span. The number of cutters on the cutter section is influenced by the number of cutters in one tool plane and by the width of the cutter section. A range of cutters is illustrated in Fig. 1.12.

Fig. 1.10 Water Sprays on a Longwall Shearer to Reduce Respirable Dust (source http://www.cdc.gov/niosh/mining/aboutus/history.htm).

Fig. 1.11 Cutter Drum (1 – body, 2 – cutter holders, 3 – cutting tool).

Fig. 1.12 Shearers Cutting Tools (1 – cutter body, 2 – cutter tip, 3 – soldering, 4 – cylinder part for inserting into the holder, 5 – lock).
The cutters construction abides mechanical properties of the broken rocks, the method of its work on the cutter section and the shape of the cutter holder. Some basic requirements which the combined cutter-loader cutters must meet are as follows:

- high strength during coal mining as well as in breaking off adjoining rock which is difficult to break loose,
- minimum power consumption for breaking loose,
- low speed in blunting the cutter tip,
- easy mounting and demounting of a cutter during operation,
- low loss in cutters.

In the course of work of the combined cutter-loaders, all the electromotors, oil filling in the drive units and oil filling in the hydraulic unit to control arm positions are getting warmer. All these elements are cooled with water which is supplied into the combined cutter-loader by a flexible hose from the pipeline in the passage to the coalface. On the entry end of the combined cutter-loader, there is a filter that protects the interior of the combined cutter-loader coolers from being fouled.

The jackets of electromotors are customized for the flow of cooling water. In the oil filling of the hydraulic unit tanks there are coolers through which water flows and cools them down. The cooling is intensive; the amount of water must not exceed a manageable limit. The amount of water is partially regulated by sprinkling the cutter sections in the course of work of the combined cutter-loader. The warmed cooling water, which runs through the electromotors and coolers, is usually brought to the sprinkling jets on the cutter sections as you can see in Fig. 1.10. During the movement of a combined cutter-loader, the jets create a water screen in the place of breaking, which reduces the dustiness during breaking coal loose. The water supply is automatically stopped when a cutter section stops. Apart from its positive effect, the sprinkling has a negative effect on the increase of water in the coalface; therefore there is a tendency to limit the amount of water to a quantity that is indispensable. This is being achieved by an appropriate construction of the jets and observation of the supplied water pressure. Figures 1.10 and 1.13 illustrate some arrangements of cutter holders with jets.

1.9 Combined Cutter-Loader Equipment

The broken rock must be loaded onto a scraper conveyor that consequently transports it out of the coalface. The loading function is executed by a cutter section with a gathering shield as well as by gathering plates that are placed on the pillar side of the scraper conveyor. The cutter section moves and loads the rock onto the conveyor by helical lugs on its circumference. Therefore, the cutter sections of the right and left arm also have helixes with different windings (right-handed and left-handed) so that the rock is always moved towards the conveyor in the course of a combined cutter-loader passage in different directions.

The gathering shield of a cutter section can be formed by a board that can be tilted. Its dimensions depend on the cutter section, i.e. its diameter and length. With combined cutter-loaders for thicker seams, the gathering shield is of a semi-circular shape and it is turned into the gathering position by hydraulic cylinders.

The gathering plates are designated for loading the broken rock from the area between the pillar and the conveying troughs. They are fixed onto the through sides on the pillar side. An example of an arrangement of a gathering plate on a conveyor is shown in Fig. 1.14.

Fig. 1.14 Armoured Face Conveyor with a Toe Plate (1 – conveyor, 2 – toe plate, 3 – shearer travel track, 4 – shifting mechanism, 5 – conveyor end plate, 6 – coal pillar, 7 – broken coal).

The combined cutter-loader control system must ensure the following:

- a safe and comfortable control and handling of the machine in accordance with the combined cutter-loader’s operating conditions,
- the control elements must be placed on safe, easily-accessible places and the possibility of their accidental switch-on must be minimized,
- a simple, preferably wireless remote control, which is not limiting the operating crew’s working capacity,
- along with the remote control, there should also be a possibility to control it by the device on the combined cutter-loader,
- the combined cutter-loader must be equipped with an automatic signal transmission at a switch-on,
- the combined cutter-loader must be equipped with a device preventing it being switched on from two or more control stations. If necessary, functioning of others must always be excluded and only one control station is left in operation,
- all operational controls and emergency switch-off must be centralized into one place and distributed in such manner that it would be always free and safely...
2. COAL PLOUGHS

2.1 Evaluation of a Current State of Underground Coal Mining

In the 1990s, ploughing ranged among the most economical mining technologies of low seams in Europe. However, along with switching to high coal mining, this technology lost on its importance. As OKD, a.s. again begins to approach low seams where ploughing technology is suitable, this issue is becoming very topical again.

Following the political and economic changes in the countries of Eastern Europe and the Czech Republic, the developments in the last decade have had a significant impact on the underground coal mining processes in Europe. At the beginning of the 1990s, there were huge changes in traditional "coal" powers of Western Europe. Coal mining was stopped in Holland and Belgium; France and Germany noted its gradual considerable reduction. A sharp fall in mining has been drawn also with the most powerful coal producer in Western Europe, in the German Federal Republic (GFR). The target programme of German coal mining as of 2005 is extraction of 26 million tons, keeping ten underground mines and reducing state subsidies to a half in 2005. At the same time, as of 2020 it supposes a drop in mining down to 50% of stated consumption due to improvements of all follow-up processes making use of coal substance (power engineering technologies, coke technologies, chemical processes based on coal and environmental technologies).

Despite the above-mentioned developments in Western European mining as well as a drop in coal demand, the industrial activities cannot do without this substance. Future developments in coal supplies for the whole of Western Europe, the so-called strong "coal" powers will play a decisive role, see Tab. 2.1.

<table>
<thead>
<tr>
<th>Country</th>
<th>Production Capacity</th>
<th>Export</th>
<th>Consumption</th>
</tr>
</thead>
<tbody>
<tr>
<td>POLAND</td>
<td>in 2001 the production was 104 million tons, with the export of 23 million tons. Sudden changes can be expected along with an anticipated privatisation of coal mining and consequential clearing off debts of the mines in operation. The then level of state subsidies also played an import role before joining the EU.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>USA</td>
<td>with a high level of mining exceeding 1 billion tons, the export capacity is quite low and the consumption in the domestic market is consequently too high. With strong pressures on the mining costs per 1 ton, efficiency is upgraded, mainly of underground mining and penetration of European technologies.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>AUSTRALIA</td>
<td>in 2002 the producing capacity was approx. 246 million tons and the export from this annual production was a total of 186 million tons, divided into 86 million tons of energetic coal and 100 million tons of cokeable coal. Out of this share, the European market was supplied by 6 million tons of energetic coal and 21 million tons of cokeable coal. The above-mentioned figures make Australia the most important world-wide exporter of coal.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Tab. 2.1 Coal Production in Selected Countries.

Recently, the relevance of Chinese coal mining has also grown on importance. This is mainly due to an application of modern technologies of longwall working along the strike.
There is a significant decrease in coal production down to approx. 1000 million tons, along with an increase in concentration and performance parameters. The production is directed to the domestic market.

The main countries of the former USSR have not established themselves noticeably in the European market so far. Having successfully dealt with the issue of quality in relation to preparation technologies, performance and transport, it will be necessary to take this producer trying to penetrate the coal market into account.

The plough technology noted its gradual development in the 1950s. The credit for this must be given to the successful construction of puller ploughs REISSHAKEN HOBEL RH 1 to RH 4 of WESTFALIA LÜNEN company (Germany). These plough systems were introduced in the German mining; later on, they were bought in the Czech Republic, Poland, former USSR and gradually they were also manufactured in these countries. All the types, marked as puller ploughs, have been in a successful operation up to date, especially in the conditions of low seams with well ploughable coal. Their principal drawback was the fact that their application did not secure a constant depth of a cut; the cut was dependent on the thrust size on the pillar. During the passage of a plough body, the face belt conveyor is being pushed away from the pillar onto the caved side and, thus, its ventilation occurs. Therefore, this type of the puller plough was later superseded by ploughs with a constant cut depth, which are more efficient and can be used for harder coals.

### 2.2 Main Parts of New Ploughing System Constructions

Ploughing systems, see Fig. 2.1, comprises of the following parts:

- scraper conveyor,
- plough body mounted with a cutter system,
- plough drive,
- plough chain guide,
- plough chain,
- tilting device,
- plough system anchoring.

![Fig. 2.1 Coal Plough Systems.](image)

### 2.3 Scraper Conveyor

A scraper conveyor, see Fig. 2.2, represents the fundamental part of a ploughing system that ensures drawing of the broken coal and simultaneously it forms the skeleton of the plough chain guide. On the conveyor drive frame there is also a plough drive. The development in technologies of the last years has pointed out the necessity of heavy conveyors the construction of which eventuated in the GFR; the DBT has developed scraper conveyors of the PF range. These robust conveyors with an E section allow for the use of a pedestal frame thickness of 20 – 40 mm, with new types of the PF – 5 range over 50 mm. The through joint strength ranges from 2000 – 4000 kN.

![Fig. 2.2 Scraper Conveyor](image)

The drive frames of the MR 15 – 17 and MR 15 – 23 ranges support the installation of drive unit outputs of 200 – 500 kW; with the new MR 35 range it is up to 800 kW. As for the transmissions, the PL25 or KPL25 epicyclic gear cases are applied; referring to new conveyors, there are specially developed the DBT PL25/30 or KPL25/30 high-output epicyclic gear cases for the outputs of over 500 kW. Working on the development, an advantage of single-speed asynchronous motor application with the output up to 800 kW has been noticed.

As for the coalfaces with the production over 3500 t/day of gross output over 400 m long, gear cases were prepared, which apart from usual and currently used ones allow for additional functions without adding expensive parts:

- smooth start,
- good start of a conveyor when loaded,
- load equalization,
- overload protection,
- very slow run,
- conveyor chain tension.
With the dimensions of the gear case, little space was taken into consideration, especially if the gear case is set in the breaking. It is the gear case with which the Ruhrkohle Bergbau, a.s. complements the range of more powerful gear cases. Transmissions of 45 CST (CST stands for a controlled start transmission) are rated for a drive output of 800 kW and momentarily for an output torque of 450000 Nm.

The gear cases are constructed in a modular way and they comprise of two cases fastened together. In one case, there is an epicyclic gear case with a spur gearing or an epicyclic gear with an angle transmission with a special plate clutch, see Fig. 2.3. In the other case, there is a CST power supply unit with an oil pump which for cooling flows through the plate clutch. In this case, there is also a hydraulic control and measured values memory.

The plough chain guide, see Fig. 2.4, must ensure a definite guidance of a plough body, with a sufficient accommodation to the unevenness formed by troughs and bumps. Next, it must ensure a simple access to the plough chain, prevent blocking of chain channels and secure a high wear resistance on the guide tracks. The guide must also be adaptable with a scraper conveyor and its troughs of a combined cutter-loader design.

The plough body, see Fig. 2.5 and 2.6, must cut coal off the seam in a steady cut and at the same time load it onto the conveyor. It must be able to adapt to seams of various thickness and be led exactly on the plough guide. The body contour has been gradually developed into a feather (concave) shape as it best corresponds to the distribution of forces onto the individual cutter groups.
2.6 Plough System Tilting Device

The tilting device, see Fig. 2.7, serves as a guidance of the floor cutters and thus secures good following of the floor. It also allows for the use of bigger forces with shifting cylinders and therefore bigger thrust that makes deeper cuts with hard coals possible.

Fig. 2.7 Tilting Device.

2.7 Plough System Anchoring

High strengths of the conveyor chain and the plough thrust the conveyor and the plough guide together. The block plough anchoring close to the drive makes the plough system move towards the coalface, see Fig. 2.8. According to the state of the seam, the movement of the conveyor can gradually start and continue. The shifted conveyor influences ploughing a great deal. A laterally shifted conveyor causes breakdowns during operation on the main as well as auxiliary drives, but also in the coalface. In order to prevent this, the block anchoring is applied. The block anchoring supports are support props. As these must also be shifted – they do not function as a support – on each drive at least 2 block anchorings should be installed. With a sufficient number of block anchorings it is possible to shift the complete plough system by up to 45° upwards even in semisteep measures.

Fig. 2.8 Plough System Anchoring.

2.8 New Plough Drive Constructions for Ploughing in Difficult Conditions

In order to transmit the motor output onto the plough drive, S-15 LK plough transmissions of the DBT company used to be employed, see Fig. 2.9. It is a conical spur gear and an integrated plate clutch of a maximum output of 315 kW.

The integrated plate clutch is clipped in a standard way (by a spring) and it slips in case of an overload. The spring strength can be adjusted and monitored electronically. The S-15 LK transmission also monitors the position and the direction of the plough.

Fig. 2.9 S-15 LK Gear Case.
Requirements for an increase in plough outputs to 500 kW and next to 800 kW led to developments and installation of transmissions of a new generation of the P-30 UEL type, see Fig. 2.10. It contains an epicyclic gear (small space needed) equipped with an integrated plate clutch ensuring an overload protection. The working principle of this gear case is a constant load. The overload protection is represented by a multi-plate clutch that holds the ring gear at the exit end. In case of the overload, the clutch slips and the ring gear revolves. Both electromotors switch off when the ring turns.

When ploughing hard coals, the plough drives are required to:
- protect drive elements against overload,
- make use of installed driving power,
- allow a minimum number of switch-ons that protects the motors from thermal overload,
- define cut depths to secure balanced load on the drive elements.

An efficient utilization of the installed drive power is reached by output equalizing systems. There are passive systems that can only reduce the number of revolutions of the main motor drive (essential on both drives) and active systems that can both brake and accelerate the main motor (essential on one drive only).

### 2.9 Electrohydraulic Drive System with a PS 16 Transmission

The electrohydraulic drive system with a PS 16 epicyclic overlapping gear case of WESTFALIA BECORIT company (now a part of BUCYRUS company) was the first output equalizing system for ploughing hard coals, suitable for underground work.

Possible regulation of the load equalization by the active system calls for a special overload protection. The fundamental disadvantage of this system is the fact that it is very rigid. This results in the plough chain and the PS 16 gear case having a significantly higher maximum load than with an electrohydraulic system.

For an efficient combination of overload protection and output equalization, the WESTFALIA BECORIT company has developed for its field a proven system of overload protection called S15-/S-20 ÜL, an active gear case equalizing load, LA-15. In this case both functions are ideally combined. The anti-overload protection system of S15-/S-20 ÜL met almost all requirements, see Fig. 2.11. An exact, operation-safe measurement method in case of overload lies in the observation of torque moment directly on the chain spur wheel.

The hydraulic tension gear permits a fast and safe execution of the required chain initial tension. The system must allow an exact, reliable and safe taking up of the chain. At the same time, it keeps the chain in the initial tension by means of a locking device. After this chain operation, the chain is slack and the required switch to join both chain ends is possible. The initial chain tension value can be displayed on the pressure gauge.

### 2.10 Powered Supports

The plough system can be run with an individual hydraulic support (IHV) or advancing support. It can be said that following an introduction of higher-class ploughs, on average both coal production and output have doubled. The maximum production values were up to 800 t/day and the mined area was up to 300 m². The introduction of powered supports with an electrohydraulic control and a defined cut brought significant improvements, see Fig. 2.12. This technology increased the production and output by 3.5 – 4 fold.

The most applied powered supports for low seams and seams of medium thickness is a shield two-prop telescopic support with a stable upper bar, with the range of 0.5/1.5 or 0.7/2.
What is decisive for automated ploughing is a modern intelligent electrohydraulic control. Currently, a PM4 control is supplied along with the supports, see Fig. 2.13. For more information see also chapter 3.

2.11 Ploughing with a Defined Cut and Others

Recently, in the OKD. a.s. company, ploughing with the co-called defined cut has proved practical. This permitted an introduction of intelligent control systems of powered supports, including an optimization of all linked elements of the plough system. During ploughing with the defined cut, the regime breaks the coal pillar only under a set value. The value is usually 4 cm, but it can be higher. It is not desirable to fall under the value of 2 cm – the so-called shade ploughing. The control system allows for all the functions be visualized. Visualization can be installed in the central point in the mine – as a rule, in the road under the coalface or in the central point on the ground.

At present, the most modern plough systems are applied in the coal mines in Germany where black coal mining is concentrated under DSK (Deutsche Steinkohle) a.s. – German black coal. Their operational employment for coal mining in 1.5 – 1.7 m of seam thickness has shown:

- “seam thickness” parameter determining loads of average 1.6 m for both processes of ploughing and combined cutter-loader mining is identical,
- for average thickness there is one significant difference; with combined cutter-loader coalfaces the breaking-down is higher by 24% than with plough operations – this means that there is a considerably higher share of cliffs in the combined cutter-loader operations; there can be extra costs (unconsidered here) in the amount of EUR 8 per one ton of rock,
- even if the medium seam thickness is identical, the average daily output in plough operations is higher by 18%,
- average operational costs with plough operations keeping the same seam thickness and approximately same number of staff, irrespective of extra costs due to an increased share of rock, are 19% lower than with combined cutter-loader operations.

On the basis of the above-mentioned it can be supposed that a new plough technology could increase the output in the coalfaces even more and thus decrease the mining costs.
3. SUPPORT EQUIPMENT IN UNDERGROUND MINE COALFACES

3.1 Introduction to Support Equipment

The support equipment always forms part of the mining complex and its main tasks are as follows:

- Provide resistance against the forces of the upperwalls and regulate their subsidence,
- Create a safe working area both for the movement of staff through the coalface and for the flow of air current,
- Co-operate with the other machines in the mining complex – shift them, hold them in the desired position or push them onto the coal pillar,
- According to needs, move its own position in the coalface on its own.

Figure 3.1 shows a sectional view of an underground mine coalface as an example of machinery arrangement in the coalface.

3.2 Individual Support

For sheeting in the coalfaces using the individual supports, the support unit is two-particle:

- upper bar, carrying out the function of a bearing part,
- props, carrying out the function of a standing part.

With the individual supports, most of the operations are done by hand. In the course of handling an operation, both parts – upper bar and prop – can be separated and handled separately.

The individual supports used to have a very wide use in the coalfaces with various mine geological conditions; these days powered supports prevail and the individual supports are used only in the low seams. However, they are still much used in the powered support coalface entrances. These days, mainly hydraulic props with an external fill are applied.

Hydraulic props with an external fill are supplied from the central working fluid distribution, which passes through the whole flow. From the central distribution there are branches equipped with a filling connection at the end. The main hydraulic element of the prop is a valve block that has a feed check valve, safety valve and prop withdrawal valve. When mounting the hydraulic prop, the filling connection is fitted on the neck of the valve block and after the filling connection is open, the fluid flows under the prop piston and braces it. After mounting, the filling connection loosens from the valve block and is used for mounting other hydraulic props in the coalfaces. When straining the hydraulic prop over the nominal load capacity, the liquid flows through the safety valve out of the prop. During prop withdrawal, a prop withdrawal lever is used with which the prop withdrawal valve is opened and the fluid flows out of the prop. A tensioning spring helps a faster prop withdrawal. To see an example of the prop with an external fill go to Fig. 3.2. Fig. 3.3 displays a valve block.

Fig. 3.1 Sectional View of an Underground Mine Coalface (1 – pillar, 2 – roof, 3 – floor, 4 – fall of face, 5 – mining machine/shearer, 6 – armoured face conveyor, 7 – support).

According to the type of handling and control in the coalface, the support equipment falls into the following groups:

- individual support,
- powered support.

What is distinctive is the method of translocation in the coalface. The individual support is controlled and also moved from one position to another by hand, whereas all the activities of the powered support are robotized by hydraulic cylinders.

Fig. 3.2 Hydraulic Props with an External Fill (1 – prop withdrawal lever, 2 – head, 3 – valve block, 4 – handle, 5 – external column, 6 – internal column, 7 – tensioning spring).
Hydraulic props with an external fill are produced in integrated ranges with load-bearing capacities of 300 and 400 kN, and extended lengths from 500 to 4000 mm. An integrated range of props (i.e. an example of a hydraulic circuit arrangement) is displayed in Fig. 3.4.

A hydraulic circuit of props with an external fill comprises of:
- valve block (forms part of the prop),
- hydraulic unit,
- central pressure line,
- pressure line branches,
- filling connection.

The working fluid for the hydraulic props with an external fill is emulsion, which is a mixture of water and emulsiifier oil with concentration 1 – 3 %. The working fluid has the pressure from 20 to 32 MPa. The hose bore of the central pressure line is Js 20, and of the branches to the filling connections it is Js 13.

3.3 Coalface Powered Support

Much like the individual supports, the powered support in the underground mine coalface also forms part of the mining complex, which is usually made up by a mining machine, scraper conveyor and supports. The mining complex tends to have the best operational results, i.e. to break the most mine run applying the least manual work. The hinder of the process used to be the individual support, which consumed too much time and physical effort in order to shift it into a new position and thus create a new working area in the coalface.

In the 1950s, it was for the first time when a powered support was constructed in Great Britain, see Fig. 3.5. In the course of 55 years of its existence it has undergone a turbulent development, but it also showed that it has a positive impact on the labour productivity level in the underground mines. To be able to meet its mission in the mining complex, it must execute a range of functions that are expected in this case; in particular:

- Good protection of the working area for the working crew, all the way from the roof, fall and pillar.
- Deadening the forces of the converging overlying strata.
- Adjustment of the support height in the broken seam thickness along with a maximum height manoeuvrability.
- Following the end of a mining machine operation cycle, movement of the support into a new working position.
- Movement of a face belt conveyor and mining machine into a new working position.

All the methods of motion are carried out hydraulically by means of hydraulic cylinders located in the powered support construction, which are controlled by hydraulic elements, usually from one point. The support is composed of individual sections erected one next to another along the coalface.
3.4 Main Functions of the Powered Support

Support mounting: With this activity, the working fluid is supplied into the support hydraulic prop section until the upper bar leans against the roof and performs its chucking power. Then the supply of fluid is cut off. The chucking power size is usually 70/80% of the support nominal load capacity and also depends on the roof quality. With a weak overburden, the chucking power is lower so that the roof is not destroyed immediately during mounting the support. The chucking power depends on the supplied fluid pressure as well as on the support operator’s correct handling if there are no means for an automatic control of the support mounting.

Transfer of forces from the overburden: Following mounting, the overburden force affects the support. Along with the increasing forces in the hydraulic props, the working fluid pressure grows as well.

Reaching the nominal load capacity, the pressure in the props reaches the value for which the safety valve is set and the fluid starts flowing off the prop. The smaller volume of the working fluid, the shorter prop and support height – support “convergence” occurs. The downward travel of the support is for as long as the forces on the nominal load capacity level are performed. If there is a drop in force under this limit, the safety valve closes and the downward travel is stopped. The support is loaded with forces from the roof and fall in the course of the whole mining cycle in the coalface. After decoaling the mining machine increment of face advance, the support, mining machine and conveyor need to be shifted into new working positions.

Support shift: With modern powered supports the basic position in the coalface is a “step back” position, i.e. its distance from the scraper conveyor is by one lift of a shift mechanism hydraulic cylinder. This position enables the support of individual sections being shifted into a new working position directly after decoaling the increment of face advance by a mining machine. Before moving the section into a new position, its hydraulic props “withdraw”. It means that from the inside, a share of working fluid is released and thus their length is shortened and the upper bar releases itself from the roof. After the prop withdrawal, the support section can be moved into a new working position. To move a section there is a shift mechanism whose one end is fastened to the scraper conveyor and the second to the support frame. The shift mechanism is controlled by a hydraulic cylinder which changes its length as well as the distance between the shift mechanism fastenings on the conveyor and section frame. When moving the section towards the conveyor, the working fluid is supplied into the hydraulic cylinder in a way that its length shortens. In this way, the section is moved into a new position towards the conveyor. Using the hydraulic cylinder lift up, the section shift stops. In a new position the support section is consequently fastened. All the time when a section is being shifted, the neighbouring sections carry the overburden forces.

Shifting the scraper conveyor and the mining machine: Mounting the support section in a new working position, there is a stable support to shift the conveyor and the mining machine. The conveyor is shifted gradually along the coalface and the mining machine is usually moved in some coalface end position. There is a shift mechanism whose hydraulic cylinder prolongs itself by the intake of the working fluid and thus holds off the shift mechanism fastenings on the conveyor and frame, and shifts the conveyor away from the mounted support. In the new position, the conveyor is held hydraulically by the blocked hydraulic cylinders of the support.

3.5 Standing-Enclosing Supports

Standing-enclosing supports provide supports above the working area in the coalface as well as an enclosure preventing the broken rock from entering the working area. Furthermore, the basic parts of the supports, apart from a base frame, hydraulic props and upper bar, are also shields enclosing the working area from the fall on the face. In the past 30 years there has been a significant progress in the developments of standing-enclosing supports which have become a dominant type of supports for all worked seams.

In contrast with the previously applied supports, especially standing supports, the standing-enclosing support has the following advantages:

- better protection of the working area from the roof and fall,
- it is simple as for function and operation as it has a smaller number of hydraulically controlled elements,
- long upper bar with a great load capacity on the pillar edge,
- big height range given by the inclined position of the hydraulic prop as well as by a convenient arrangement of joining the frame and shield,
- better stability applicable mainly with supports for seams with a big longitudinal face gradient and supports for steeply inclined seams,
- it affects a relatively low specific surface pressure on the coalface floor and roof.

What has become important for the arrangement of standing-enclosing supports is joining the frame and shield; they are divided as follows:

- supports with a single-joint coupling of the frame and shield is in Fig. 3.7,
- supports with a four-joint coupling of the frame and shield is in Fig. 3.8.

Basic operating diagram of powered supports: The basic division of powered supports is given by their functional relationship to the overburden; the supports can be divided as follows:

- standing support,
- enclosing support,
- standing-enclosing support.

Schematic diagrams of powered supports are in Fig. 3.6.

Fig. 3.6 Schematic Diagrams of Powered Supports (a – standing, b – enclosing, c – standing-enclosing).
Fig. 3.7 Support with a Single-Joint Coupling (1 – frame, 2 – shield, 3 – hydraulic prop, 4 – telescopic arm, 5 – upper bar, 6 – joint of a shielded arm, 7 – upper bar pivot).

Support with a single-joint coupling: The hydraulic prop changes its length along with the changing worked seam thickness. Simultaneously, the gradient of shield changes; it moves around the coupling joint with the frame. Along with this movement, the upper bar pivot makes a circular movement with the centre in the coupling joint of the frame with the shield. As the distance of the upper bar pillar edge from the upper bar pivot is constant, along with an increasing height of supports the distance of the pillar edge from the pillar grows, which is unfavourable as for good protection of the working area. With such supports this drawback is usually offset in two ways:

- a telescopic arm located in the shield of the support which draws out in dependence on the seam thickness,
- a telescopic front part of the upper bar, which can achieve the same effect.

Another disadvantage of this support construction type is a quite small resistance, due to a low lean of the hydraulic prop into the shield and, thus, low force affecting the upper bar. On the contrary, its advantage is an extraordinarily simple construction and its great flexibility.

Support with a four-joint coupling: Currently, only standing-enclosing supports with a four-joint coupling of the frame and shield are constructed, see Fig. 3.8. The coupling of the frame and shield is carried out by means of drawbars of such length that, along with the changing height, the top shield pivot moves along the lemniscate, whose part is made use of in the supports, edgewise it is approximately parallel with the pillar. This ensures the fact that the front part of the upper bar keeps the same distance from the pillar in the course of the whole height of the supports for which it is contracted. The four-joint coupling of the frame and shield with the lemniscate course is shown in Fig. 3.9.

Coupling the frame and shield by means of drawbars, the supports gains a certain rigidity in the whole operating range and does not call for any special means to keep the distance of the upper bar pillar edge from the pillar, and the overall construction makes a safe working area in the coalface. An example of a powered support with a four-joint coupling of the frame and shield is shown in Fig. 3.8 and 3.10.


3.6 Resistance of the Powered Supports

Resistance is an important parameter of a powered support and it can be related to two parameters of the supported roof:

- support resistance by roof surface /kN×m^-2/,
- support resistance by coalface length unit /kN×m^-1/.

3.7 Required Support Resistance and Structural Design

In order to determine the required support resistance \( R \) for specific working conditions, there is a range of theories that differ in various countries with advanced mining industry. The reason is that it is not always possible to uniquely determine and apply all the influences that will affect the support in the coalface. Therefore, determining the required support resistance is usually approached empirically. In the Czech Republic, one of the most applied formula is given by prof. Zamarský (VŠB - Technical University of Ostrava, Czech Republic):

\[
R = \frac{10\, \rho \, w \, k_{w} \, k_{z} \, k_{r}}{k - 1} / \text{kN×m}^{-2}/
\]

where:
- \( w \) is worked seam thickness /m/,
- \( \rho \) is overlying rock density /t×m^-3/,
- \( k_{w} \) is coefficient of the fall lag /1/,
- \( k_{z} \) is coefficient of the broken out area disposal method,
- \( k_{r} \) is coefficient of the overlying strata self-supportedness,
- \( k \) is factor of swell in the failure cushion.

The structural design of powered supports corresponds to the operational conditions in the countries in question. There are significant differences, especially with supports designed for:

- low seams,
- seams of medium thickness and high coal.

Illustrations of such powered supports are in Figures 3.12 and 3.13.
Apart from a steel frame which is customized for the support performance, the power hydraulics and the support hydraulic circuit are decisive.

3.8 Main Powered Support - Power Hydraulics

Among the power hydraulics there are:
- hydraulic props,
- shift mechanism,
- hydraulic cylinders to control the upper bar position,
- hydraulic cylinders of side panels,
- support frame stability system,
- support frame lift.

**The hydraulic props**, see Fig. 3.14, exert chucking power of the support and consequently transfer the forces affecting the support from the overburden. The hydraulic prop design depends on the size of the desired travel in proportion to its minimum height. With small proportion, one *single-telescopic* prop is enough, which is simpler; bigger proportion calls for a *double-telescopic* one. A block diagram of both prop types is in Fig. 3.15.

**Single-telescopic props**: Their hydraulic lift is given by the difference between the end positions of piston. The prop draws out when the working fluid is supplied under the piston through the underfeed. Simultaneously, from above the piston the fluid is pushed out of the prop through the top feed. During prop ramming the fluid is fed through the top feed and pushed out through the underfeed.
Double-telescopic props: The lift is determined by the sum of the central and internal column lifts. The pressure chamber of the external and central column is separated by a back (so-called foot) valve. This is due to the fact that with the same load capacity of the internal and central columns, in both pressure chambers there are different pressures given also by various piston diameters ($d_1$ and $d_2$). The prop draws out when the fluid is fed under the piston through the underfeed. At the same time, the fluid is pushed out of the prop via the top feed. First, the central column usually draws out and consequently, the fluid flows through the foot valve into the pressure chamber of the second stage and draws out the internal column. The foot valve closes after the fluid supply is stopped and thus also the pressure chamber of the second stage is closed. During prop ramming the fluid is fed into the top feed which is joined with both chambers above the central and internal column pistons. First, the central column rams. Contacting the bottom, the guard opens the foot valve and the internal column rams. During prop column ramming the fluid is pushed out of the prop via the underfeed which must be well rated as there is the same increased amount of fluid as it was in the case of a single-telescopic prop.

The shift mechanism is an important part of the powered support; with all the supports it has the following basic functions:
- shifting the support sections,
- shifting the conveyor and the mining machine,
- keeping the conveyor in the shifted position.

In addition, in the plough systems:
- generation of a stable thrust of the conveyor onto the pillar.

Hydraulic cylinders to control the upper bar position, the so-called angular cylinders, are located between the shield and the upper bar. It moves the upper bar around the joint in the coupling of the upper bar with the shield.

3.9 Powered Support Hydraulic Circuit

The hydraulic circuit can be divided into two parts:
- feed circuit,
- support control circuit.

The feed hydraulic circuit serves to supply the pressure liquid to the powered support sections and comprises of:
- hydraulic unit,
- system of hoses, fittings and cocks for the distribution of the pressure liquid in the coalface.

The hydraulic unit is usually composed of the following main parts:
- tank with mixing equipment,
- pump,
- cut-off valve,
- accumulator,
- filters (suction and refuse),
- safety valve.

Fig. 3.16 displays a hydraulic unit.
fitted with a 0 – circle is inserted into the fittings with a smooth, precise aperture and secured with an attachment clip. Such coupling of hoses permits fast demounting and mounting of the coupling and very high tightness.

![Fig. 3.17 Hose End.](image1)

Apart from hoses in the feed circuit but also in the support section circuits there are coupling fittings, such as direct couplings, T-pieces, knee-joints, cross couplings, taps, etc. They are all adjusted for a threadless coupling, and are displayed in Fig. 3.18.

![Fig. 3.18 Hydraulic Circuit Fittings.](image2)

The section of control circuit of the powered support has a great deal of functions and also it must protect the individual performance elements and the construction from being overloaded. The basic circuit elements in the support section are as follows:

- hydraulic distributor,
- hydraulic locks (non-return and two-way),
- safety valves,
- pressure indicators,
- special hydraulic elements,
- hoses and fittings.

**Types of powered support control:** The control must comply with all the required functions in the support along with respecting the operator’s safety. It means that the support section control must be carried out from a safe area, from at least the neighbouring section. The following types of control are known:

- full-flow control from the neighbouring section,
- impulse control from the neighbouring section,
- electrohydraulic control.
APPENDIX 1

SIMULATION-BASED RELIABILITY ASSESSMENT AND PERFORMANCE-BASED DESIGN APPLIED FOR BEAMS

This appendix (i.e. article: FRYDRÝŠEK, K.: Simulation-Based Reliability Assessment and Performance-Based Design Applied for Beams, In: Book and CD-ROM of 3rd International Conference “From Scientific Computing to Computational Engineering” - 3rd IC-SCCE, pp. 1-8, Athens, 2008, Greece) is focused on the solution of simple beams continually supported by elastic (Winkler’s) foundation. The foundation contains longitudinal changes. For the calculation of displacements, stresses and dimensions are derived some analytical and numerical procedures based on probabilistic approaches (Simulation-Based Reliability Assessment (SBRA) method, Monte Carlo Simulation Method, Performance-Based Design (PBD), AntHill software). Probabilistic approach includes influences of variability of loads, shape and material of the beam, and variability of modulus of the foundation (described by bounded histograms). Probabilistic approach is used for the reliability expertise, calculation of safety and calculation of dimensions of the beams. PBD is also based on the performance requirements which are usually defined as a synthesis of functionality, all-in cost, safety etc. Performance requirements can be expressed as an acceptable level of damage, which is defined by acceptable probability of possible failure. The acceptable level of damage is related to the yield limit.

Keywords: beams, elastic foundation, numerical simulations, Monte Carlo simulations, probability, Simulation-Based Reliability Assessment, Performance-Based Design

Abstract. This paper is focused on the solution of simple beams continually supported by elastic (Winkler’s) foundation. The foundation contains longitudinal changes. For the calculation of displacements, stresses and dimensions are derived and applied some analytical and numerical procedures based on probabilistic approaches (Simulation-Based Reliability Assessment (SBRA) method, Monte Carlo Simulation Method, Performance-Based Design (PBD), AntHill software). Probabilistic approach includes influences on the variability of loads, shape and material of the beam, and variability of modulus of the foundation (described by bounded histograms). Probabilistic approach is used for the reliability expertise, calculation of safety and calculation of dimensions of the beams. PBD is also based on the performance requirements which are usually defined as a synthesis of functionality, all-in cost, safety etc. Performance requirements can be expressed as an acceptable level of damage, which is defined by acceptable probability of possible failure. The acceptable level of damage is related to the yield limit.

1 INTRODUCTION TO THE BEAMS ON ELASTIC FOUNDATION

The basic analysis of bending of beams on an elastic foundation is developed on the assumption that the strains are small and the reaction forces \( q_0 = q_0(x) \) \( \text{Nm}^{-1} \) in the foundation are proportional at every point to the deflection \( v = x(x) \) \( \text{m} \) of the beam at that point, etc. (first proposed by E. Winkler, Prague 1867)\(^{[1],[2],[3]}\), see Fig.1. External loads on the beam also evoke bending moment \( M_0 = \text{Nm} \), axial (normal) force \( N = \text{Nm} \) and shearing force \( T = \text{Nm} \), see Fig.1.

Figure 1. Element of a Beam on Elastic Foundation

The general problem is described by the following ordinary differential equation:

\[
\begin{align*}
\frac{d^4v}{dx^4} + \frac{N_0}{EJ} \frac{d^2v}{dx^2} + \frac{K}{EJ} \frac{d^2v}{dx^2} &+ \frac{q}{EJ} \frac{d^2v}{dx^2} = 0 \quad (1)
\end{align*}
\]

where: \( E/\text{Pa} \) is modulus of elasticity in tension of the beam, \( N_0/\text{Nm} \) is distributed load (intensity of force), \( m/\text{Nm} \) is distributed couple (intensity of moment), \( m/\text{Nm} \), \( b/\text{m} \) is depth of the beam and \( \gamma_{\text{c}} \text{deg}^{-1} \) is transversal temperature increasing in the beam. For more information about the derivation of eq. (1), see reference\(^{[2]}\).

In the most situations, the influences of normal force, shearing force and temperature can be neglected (or the beam is not exposed to them). Hence, from eq. (1) follows:

\[
\begin{align*}
\frac{d^4v}{dx^4} + \frac{q}{EJ} \frac{d^2v}{dx^2} &+ \frac{K}{EJ} \frac{d^2v}{dx^2} = 0 \quad (2)
\end{align*}
\]

From the Winkler’s theory\(^{[1],[2],[3]}\), it is evident that:

\[
\begin{align*}
q_0 = \gamma \varphi(x) = \gamma x \varphi(x) \quad (3)
\end{align*}
\]

where functions: \( \varphi(x) = \text{stiffness of the foundation and } K(x) = N_{\text{em}} \) is modulus of the foundation which can be expressed as functions of variable \( x = \text{m} \) (i.e. longitudinal changes in the foundation) and \( b = \text{m} \) is width of the beam. Hence, eq. (2) can be written in the following form:

\[
\frac{d^4v}{dx^4} + bK(x) \frac{d^2v}{dx^2} + \frac{q}{EJ} \frac{d^2v}{dx^2} = 0 \quad (4)
\]

2 FIRST EXAMPLE (BEAM ON ELASTIC FOUNDATION)

Let us consider a straight beam on elastic foundation with longitudinal changes, see Fig.2. The beam of length \( L = \text{m} \), with free ends is exposed to one vertical force \( F = \text{N} \), i.e. other loads \( q \) and \( m \) are zero. Modulus of the foundation is given by linear function:

\[
K(x) = K_0 + K_1 \frac{x}{L} + K_2 x = K_3 + K_4 x
\]

Figure 2. Solved Example of the Beam on Elastic Foundation

Hence, in this case, the differential eq. (4) can be written in the following form:

\[
\frac{d^4v}{dx^4} + \left( \frac{K_0 + K_1}{EJ} \right) \frac{d^2v}{dx} + \frac{q}{EJ} \frac{d^2v}{dx} = 0 \quad (5)
\]

The approximate solution can be found in the form of polynomial of 6-th order:

\[
\begin{align*}
\gamma = \varphi(x) = \gamma x \sum_{i=0}^{5} b_i x^i
\end{align*}
\]

where: \( b_i = \text{m} \), \( b_i = \text{m} \) are unknown constants. Equation (6) must satisfy the four basic boundary conditions and force equation of equilibrium. The last two equations can be derived via variational principles or via satisfaction of differential equation (5) at chosen points. For more details about it see\(^{[2],[3]}\). Hence, the results
are written in Tab.1. The derived results (i.e. Tab.1) fit very well for short beams. Higher approximation must be used for longer beams, i.e. function: \( v = \sum_{i=1}^{n} b_i x^i \), where \( n \geq 7 \).

Let us consider probabilistic approach\(^{[3],[4],[5],[6],[8]}\) (i.e. all inputs are given by bounded (truncated) histograms which truly include the real variability of all inputs), which is the modern and new trend in the solution of mechanical systems. Probability analysis includes influences of variability of “I” shape (\( b = 0.09 \pm 9 \times 10^{-3} \text{ m}, \quad h = 0.2 \pm 2 \times 10^{-3} \text{ m}, \quad J_{xx} = 2.16 \times 10^{-5} \pm 6.5 \times 10^{-5} \text{ m}^4 \)), material: (\( E = 1.8 \times 10^{10} \pm 9 \times 10^{9} \text{ Pa} \), yield stress \( R_y = 162.36 \pm 18973 \text{ MPa} \), load (\( F = 157324 \pm 257324 \text{ N} \)) and also the variability of modulus of the foundation: (\( K_L = 1.125 \times 10^8 \pm 3.375 \times 10^9 \text{ N/m} \), \( K_s = 1.125 \times 10^8 \pm 3.375 \times 10^9 \text{ N/m}^3 \)), for example see Fig.3 to 6 (i.e. inputs for AntHill software, Simulation-Based Reliability Assessment (SBRA) Method\(^{[3],[4]}\)). Length of the beam \( L = 0.9 \text{ m} \).

The derived results (i.e. Tab.1) fit very well for short beams. Higher approximation must be used for longer beams, i.e. function: \( v = b_0 x^0 + b_1 x^1 + \ldots + b_n x^n \), where \( n \geq 7 \).

### Table 1. Solved Example 1 (Analytical Results of the Beam on Elastic Foundation).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( k )</td>
<td>640800 \text{ B/} E_{xx} = 180(41k \text{ B/L} + 94k_L \text{ L} + 38k_J \text{ E}_{xx} \text{ L} + (3k_2 + 2k_1) \text{ A}/L)</td>
</tr>
<tr>
<td>( A )</td>
<td>560E_{xx}[9k_0(2k_1 + k_2) + 3k_3] \text{ A}/L ]</td>
</tr>
<tr>
<td>( C )</td>
<td>( k \text{ A}/L = \frac{640800 \text{ B/} E_{xx} = 180(41k \text{ B/L} + 94k_L \text{ L} + 38k_J \text{ E}<em>{xx} \text{ L} + (3k_2 + 2k_1) \text{ A}/L)}{640800 \text{ B/} E</em>{xx} = 180(41k \text{ B/L} + 94k_L \text{ L} + 38k_J \text{ E}_{xx} \text{ L} + (3k_2 + 2k_1) \text{ A}/L) ]</td>
</tr>
<tr>
<td>( k )</td>
<td>( 640800 \text{ B/} E_{xx} = 180(41k \text{ B/L} + 94k_L \text{ L} + 38k_J \text{ E}_{xx} \text{ L} + (3k_2 + 2k_1) \text{ A}/L) ]</td>
</tr>
<tr>
<td>( \text{A}/L )</td>
<td>( 560E_{xx}[9k_0(2k_1 + k_2) + 3k_3] \text{ A}/L ]</td>
</tr>
<tr>
<td>( \text{B}/E_{xx} )</td>
<td>( 640800 \text{ B/} E_{xx} = 180(41k \text{ B/L} + 94k_L \text{ L} + 38k_J \text{ E}_{xx} \text{ L} + (3k_2 + 2k_1) \text{ A}/L) ]</td>
</tr>
</tbody>
</table>

### Figure 3. Histogram of Input Parameter \( E/\text{Pa} \).

### Figure 4. Histogram of Input Parameter \( R_y/\text{MPa} \).

### Figure 5. Histogram of Input Parameter \( F/\text{N} \).

### Figure 6. Histogram of Input Parameter \( K_s/\text{Nm}^{-1} \).

### Figure 7. 2D Histogram and its Sections for Output Parameter \( v = v(x) \).

### Figure 8. 2D Histogram and its Sections for Parameter \( k = k(x) \).

### Figure 9. 2D Histogram and its Section for Output Parameter \( \sigma = \sigma(x) \).

### Figure 10. Histograms of Output Parameters:

- a) \( v_{\max} = v(x = L = 0.9 \text{ m}) \), b) \( \sigma_{\max} = \sigma(x = 0.63 \text{ m}) \)


In the following example the SBRA method (Simulation-Based Reliability Assessment, direct Monte-Carlo method, AntHill software) is used.

A shaft of unknown circular diameter $D$ (see Fig.15) is exposed to bending moment $M_3 = 581515.3 \pm 602298.6 \text{Nm}$, normal (axial) force $N = 157493.7 \pm 217727.8 \text{N}$ and torque $M_4 = 367485.4 \pm 599164.3 \text{Nm}$, which are given by bounded histograms, see Fig.16 to 18. Yield stress of material is $R_y = 338.3 \pm 99.3 \text{MPa}$, see bounded histogram in Fig.19.

Calculate value of the diameter $D$ which is given by normal truncated distribution $\pm 1\%$ (i.e. $D_{\text{min}}^{+}$) with the accuracy of $0.1 \text{mm}$. The acceptable level of a damage is $P_{\text{ACC}} = 0.0005 = 0.05\%$ (standard reliability level) and is related to yield stress. In other words, $0.05\%$ of all loading states can result in yielding.

According to the theory of small deformations\(^1\), the following can be written:

\[
\tau = \frac{M}{\pi D^3/16}
\]

and

\[
\sigma = \frac{N}{\pi D^2/32} + \frac{4}{\pi D^2} \left( \frac{N + 8R_y}{D} \right)
\]

where $\sigma$ [MPa] is maximal normal stress and $\tau$ [MPa] is maximal shear stress. Hence, for equivalent von Mises stress $\sigma_{\text{eq}}$ [MPa] the following can be written:

\[
\sigma_{\text{eq}} = \sqrt{\sigma^2 + 3\tau^2} = \frac{4}{\pi D^2} \left( \frac{N + 8R_y}{D} \right) + \frac{48.8M_4}{D^3}.
\]

Factor of safety (i.e. probability of situations when $R_y < \sigma_{\text{eq}}$) is defined as:

\[
FS = P(R_y - \sigma_{\text{eq}} < 0),
\]

where operator $^\sim$ $P$ $^\sim$ means probability.

Hence, when $FS \geq 0$, it is evident that yield limit is not reached (i.e. in the shaft there are not any plastic deformations). The goal is to calculate diameter $D$ which satisfy the following condition:

\[
FS \leq P_{\text{ACC}}.
\]
Nevertheless, it is necessary to apply iteration methods (because from the eq. (8) it is not possible to express directly the unknown parameter D). Hence, iteration loop with the application of the secant method can be used, see Fig. 20. For selected initial conditions (diameters): \( D_0 = 30 \pm 0.3 \) mm and \( D_1 = 48 \pm 0.48 \) mm (bounded normal distributions \( \pm 1\% \)) it is possible to calculate (via the SBRA method for \( 10^6 \) Monte Carlo simulations) the values of \( F_S_0 = 0.869929 \) and \( F_S_1 = 0.000032 \).

Next applications of the bisection method and secant method (bounded to interval \( D_0, D_1 \)). Plane (2D) histogram of function \( F_S \) is presented in Fig. 24. It is evident that the required diameter \( D_f \) is presented in Fig. 24. 

Hence, \( F_S \geq F_{S, \text{Accept}} \) and \( F_S \leq F_{S, \text{Reject}} \). It is evident that the required diameter \( D_f \) must be in interval: \( D \in \{ D_0, D_1 \} \). From Fig. 20, a new approximation of the diameter \( D_f \) by secant method can be derived: \( D_f = f(\text{Accept}) = D_0 (F_S - F_{S, \text{Reject}}) + D_1 (F_S - F_{S, \text{Accept}}) = 47.9 \pm 0.48 \) mm. And from the results of the AntiHill software the following is calculated: \( F_S = 0.000035 \leq F_{S, \text{Accept}} \). Next approximation of \( D_f \) can also be calculated by a bisection method, see Fig. 21. Hence: \( D_f = \frac{D_0 + D_1}{2} = 38.95 \pm 0.39 \) mm and \( F_S = 0.048731 \geq F_{S, \text{Accept}} \). It is evident that \( D \in \{ D_0, D_1 \} \). Next applications of bisection method and secant method give the values of: \( D_0, D_1, \ldots, D_6 \). i.e. \( D \in \{ D_0, D_1 \} \). (iteration loop as a function \( D = f(F_S) \)).

The whole iterative procedures and Monte Carlo simulations can be speed-up by the application of parallel computers. However, on the present, it is impossible to solve large problems of mechanics by the PBD. The reason for this is the low rate of present-day computers.

The work has been supported by the Czech project FRVŠ 534/2008 F1b.

**REFERENCES**


APPENDIX 2

STOCHASTIC SOLUTION AND EVALUATION OF THE ORE DISINTEGRATION PROCESS

5 Solution (SBRA Method in Combination with FEM)

The results (acquired by the SBRA method in combination with FEM) were subsequently statistically evaluated, as shown.

Tab. 1: Parallel Computers Used in this Study (Date: August-September 2008)

<table>
<thead>
<tr>
<th>Computer name</th>
<th>No. of CPUs</th>
<th>No. of MC simulations</th>
<th>Wall time (hours)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Alfa</td>
<td>16</td>
<td>312</td>
<td>70.395</td>
</tr>
<tr>
<td>Opteron</td>
<td>20</td>
<td>28</td>
<td>54.6</td>
</tr>
<tr>
<td>Quad</td>
<td>4</td>
<td>86</td>
<td>69.015</td>
</tr>
<tr>
<td>Pci8932d</td>
<td>4</td>
<td>74</td>
<td>68.82</td>
</tr>
</tbody>
</table>

6 Results and Stochastic Evaluation

Figure 8 to 12 show the equivalent stresses (i.e. Stres distributions) at some selected time of the solution calculated for one of the chosen 500 Monte Carlo simulations (i.e. for one situation when the material of the ore is described by values $R_1 = 12.5$ MPa, $R_2 = 13.5$ MPa, $E = 20000$ MPa, and $\mu = 0.2$).

The movement of the bit and also the subsequent disintegration of the ore caused by the cutting are shown.
7 Comparison between Experimental Measurements and Stochastic Simulations

The calculated maximum forces (i.e. SBEA-FEM solutions, see Figure 17 and 18) can be compared with the experimental measurements, see also references [4], [5], and [6]. The evaluation of measurement shows that the maximum value of maximum force in $R_{max}^\text{SBEA-FEM}$ is $2288 \text{ N}$.

Hence, the relative error (calculated for the median value $R_{v}^\text{SBEA-FEM}$, $R_{v}^\text{medial-}\text{MED}$) is $5988 \text{ N}$.

The error of $4.02\%$ is acceptable. However, the experimental results also have large variability due to the anisotropic and stochastic properties of the material and due to the large variability of the reaction forces.

8 Proposition of Fully Probabilistic Assessment

Reliability function $R_F$, see chapter 2, can be defined by equation:

$$R_F = R_{\text{MEDIAL - MED}} - R_{\text{MAX - SBEA-FEM}}$$

where $R_{\text{MAX - SBEA-FEM}}$ is the allowable reaction force in the cutting bit, which can be acquired from the real capacity of the whole cutter-loader system in the mine.

If situations when $R_F \leq 0$ occur then the cutter-loader system is overloaded. Else if $R_F > 0$, then safe situations of loading occur. Hence, fully probabilistic assessment can be calculated by comparing of probabilities:

$$P(R_F \leq 0) = P_{\text{ALLOWABLE}}$$

where $P_{\text{ALLOWABLE}}$ is the acceptable probability of overloading of the cutting-loader system. This overloading sometimes really occurs in the mine. Value of $P_{\text{ALLOWABLE}}$ can be given by chosen performance requirements of the client (i.e. operator), see reference [1], [2], [3], and [4].

9 Conclusions

This paper combines the SBEA simulation-based Reliability Assessment method and FEM as a notable tool for simulating the hard rock (ore) disintegration process.

All basic factors have been explored in plane strain formulation, material nonlinearities, and the methodology for describing the finite elements during the ore disintegration process, application of parallel computers.

The use of FE deactivation during the ore disintegration process (i.e. to avoid the crack) is a modern and innovative way of solving problems of this type.

Because the real material of the ore (i.e. yield limit, fracture limit, Young’s modulus, Poisson’s ratio, etc.) is extremely variable, stochastic theory and probability theory were applied (i.e. application of the SBEA Method).

The error of the SBEA-FEM results (i.e. in comparison with the experiments) is acceptable. Hence, SBEA Method and FEM can be a useful tool for simulating real conditions of the ore disintegration process.

The SBEA Method, which is based on Monte Carlo simulations, can include all stochastic (real numbers) inputs and all results are also of stochastic quantities. However, for better application of the SBEA method (for simulating this large problem of mechanics), it is necessary to use superfast parallel computers. Instead of 360 Monte Carlo simulations (wall time cca 70.4 hours, as presented in this article), it is necessary to calculate $> 10^5$ simulations (wall time cca 35 days or more).

Our department will be able to make these calculations when new and faster parallel computers become available.

Hence, the fully probabilistic assessment was proposed according to the acceptable probability of overloading of the whole cutter-loader system.

All the results presented here were applied for optimising and redesigning the cutting bit (excavation tool), see Figures 1 and 10 and references [4] and [5].
APPENDIX 3

STRESSES AND DEFLECTIONS OF RAIL AND ANCHOR PIN IN THE DEPENDENCE ON THE SLOPE ANGLE OF THE RAILWAY IN MINES


Stresses and Deflections of Rail and Anchor Pin in the Dependence on the Slope Angle of the Railway in Mines

Napětí a průhyby v kolejnici a svorníku v závislosti na úhlu sklonu dráhy v dolech

Karel FRYDRÝŠEK 1, Jiří FRIES 2

Abstract: This paper is focused on calculations of stresses and deflections in the rail and anchoring pin for railways in mines. The calculations, according to the dependence on the slope angle of railway, fits well for the theory of beams on elastic foundation (analytical methods in combination with FEM, emergency situation, hard stop of train). Results can be used for the design of rails and anchoring pins of rail-sections in mines.

Abstrakt: Článek je zaměřen na výpočty napětí a průhybů v kolejnici a kotvícím svorníku pro důlní dráhy. Výpočty, prováděné v závislosti na úhlu sklonu dráhy, jsou platné pro teorii nosníků na pružném podkladu (analytické metody v kombinaci s MKP, havarijní situace, brzdění do zastavení vlaku). Výsledky lze použít pro návrh kolejnic a uzavírací seky v dolech.

Keywords: elastic foundation, beams, rail, anchor pin, mining transport, dynamic effects, deflections, stresses, construction, analytical methods, FEM

Klíčová slova: pružný podklad, nosníky, kolejnice, svorník, důlní doprava, dynamické efekty, průhyby, napětí, konstrukce, analytické metody, MKP

Introduction

Mining transport is one of the most important tasks for each mining company, see Fig.1. Therefore, this paper is focused on the design of rail-sections and its anchoring produced by Ferrit company – Czech Republic, see Fig.1 and 2. The goal is to calculate displacements and stresses in anchoring pins and rails according to the theory of beams on elastic foundation and spring supports. These calculations are performed for the anchoring pin with diameter

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$D_s = 25 \text{ mm}$ in the dependence on the slope angle $\alpha$ of the railway and velocity of train, see Fig.1, 2 and Tab.1. Hence, hard stop of the train (emergency situation, train rides to descent, dynamic effects) is solved by combinations of analytical methods [1], [2], [3] and FEM (Ansys software) [4], see Fig.3. The combination of both methods is suitable because it offers the best way of solution. Foot-wall is approximated by elastic subsoil (i.e. Winkler elastic foundation) for rail-section and anchoring pin, see Tab.2 where $N$ is normal force, $EJ_{rt}$ is bending stiffness, $k_e$ is foundation stiffness of anchoring pin, $F_t$ is tensile force and $F_z$ is bending force, $v$ is displacement, $M_e$ is bending moment and $T$ is shearing force.
Some results are presented in Fig. 4, 5 (total displacements and stresses in the rail) and Fig. 6 (deflections and stresses in the anchoring pin). For more information about solved problems see references [3] and [5].
APPENDIX 4

COMPUTER SIMULATION OF DYNAMIC LOAD ACTION ON THE HYDRAULIC LEG

This appendix is focused on the computer simulation of a dynamic load action on the hydraulic leg (i.e. application of the Finite Element Method and Finite Volume Method).

GONDEK Horst¹, SZWEDA Stanisław², MAZUREK Krzysztof³

ABSTRACT: The rock burst hazard forces a necessity of adaptation of a design of the powered roof support’s hydraulic leg which has to absorb increased dynamic loads. It is required to manufacture and to test the hydraulic leg prototypes on a stand test and it is very expensive. Thus, it is necessary to develop a computer simulation method, which would enable to reduce designing costs by reducing the number of the stand tests. The process of creation of the calculation task as well as the results of numeric calculation are presented.

Key words: hydraulic legs, computer simulation, longwall, dynamic load action

1. Introduction

Operational safety of the powered roof support depends mainly on reliability of hydraulic legs. Thus, the process of legs designing and especially their protection against results of the load caused by rock mass dynamic action, is of great importance as regards operational safety of roof support. The hydraulic legs are the parts of roof support that most frequently can be damaged in a result of rock bursts. From the accident files including 63 rock burst cases (reported in the Polish part of Upper Silesian Coal Basin within 1978-1995), in which roof support was damaged, it results that hydraulic legs damage made 54% of all recorded damages of roof supports [3]. Exemplary image of powered roof support, caused by a bump of 10^7 J energy in a vicinity of longwall face in the depth of 780 m is shown in Fig.1.

According to the PN-EN-1804-2-2004 Standard the hydraulic legs undergo compulsory testing on test rigs for static and dynamic loads [10]. The tests are very expensive and that is why it is reasonable to replace the stand tests of a new prototype, or at least a part of tests, by numerical testing made on hydraulic leg models, especially in

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case of designing a new model of the leg or new device protecting the leg against overload. At present the finite elements method (FEM) is the most reliable method for modelling of quickly changeable dynamic processes and strongly nonlinear processes occurring in solids, mechanical systems or fluids. Due to the computer simulation, it is possible to carry out much more virtual tests, which enables to understand more precisely the mechanism and to design the energy absorption devices in a better way. [2]

Computer simulation of a dynamic load impact on the hydraulic leg requires a solution of many problems associated with using the FEM method to model the complex mechanical systems. The following belongs to the most important problems:

– modelling the load, transferred by the metal parts of the leg and by working medium,
– interaction of moving metal parts and fluid including the changeable contact surface between working medium and cylinder walls,
– modelling an initial setting of the leg.

The proper solution of the above mentioned problems is particularly important when modelling the devices protecting the leg against dynamic overload

– especially gas accumulators.

Process of numerical analysis preparation to model the hydraulic leg as well as results of calculations, made by MSC.Software package, will be discussed below.

2. Preparation of Numerical Analysis

The preparation of numerical analysis, which aims at modelling the dynamic impact on hydraulic leg, can be divided into the following stages:

– creation of the geometric model,
– partition of the geometric model into finite elements and volumes,
– defining the material properties,
– determination of load and boundary conditions,
– attributing the physical properties to finite elements,
– preparation and generation of the input file.

Geometric model of the hydraulic leg, discussed in the paper, was built on the basis of technical documentation of the Ø200 hydraulic leg used in the FAZOS-12/28-Oz powered roof support. Geometric model of the hydraulic leg was developed with the help of CAD (Autodesk Inventor) software which enables to create parameterized spatial models.

Parameters of the geometric model of the hydraulic leg, after loading it to the MSC.Patran (preprocessor) software, enable to create the model made of finite elements and volumes. Partition of the geometric model into finite elements and volumes has been shown in Fig. 2.

In the hydraulic legs, an external load is transferred by steel tubes and by liquid, closed in an annular compartment. Simulation of the system operation, in case of a dynamic load, requires a recreation of both types of bodies in a numerical model.

Finite elements method (FEM) was used to recreate the steel subassemblies (upper prop and lower prop). The subassemblies were divided into uniform finite elements using the solid elements of CHEXA type and the material model of the following properties: Young modulus $E=2E+11$ Pa, density $\rho=7850$ kg/m$^3$, Poisson coefficient $\nu=0.3$ and yield point $R_s=7.8E+8$ Pa, was attributed to them.

Flow of liquid was described by Euler’s method (that is a defined part of the space in which the movement and state of material inside can be tracked). The space is divided into cells – finite volumes, and in a rough approximation we can assume that they are motionless. Continuous medium, i.e. liquid, passes through the cell walls. The software calculates the following state variables: pressure, mass, momentum, internal energy of material in the cells as well as forces acting on material in the cells and resulting changes of momentum [1]. The liquid filling the annular chamber of leg was modelled using the elements of the CHEXA type. The following properties of material (liquid) were attributed to those elements:

- density $\rho=1000$ kg/m$^3$ as well as modulus of volume elasticity $a_1=1.65E+9$ Pa.

In some way, you can say that the algorithms analyzing the finite elements and finite volumes operate independently. The forces transferred by the finite elements do not act directly on the material in cells of the Eulerian space. Coupling of the solid body model with liquid model is possible due to the ALE Coupling option. The method consists in implementation of an additional object (coupling surface, called “the skins”, modelled by surface finite elements of CQUAD4 type), which in both cases uses the mechanism of entering the boundary conditions. Coupling surface enables a transfer of the pressure which acts on the finite elements nodes due to the liquid neighborhood. The surface is also a boundary for the liquid, and its relocation causes a change in liquid boundary position and forces its flow. Due to that fact, it is possible to recreate the liquid flow in a leg’s cylinder, the walls of which can relocate or deform [1]. Fig. 3 shows the method for defining the coupling surface between solid bodies and liquid.

When modelling a dynamic load action on the hydraulic leg, installed in the support, which was set in the workings, we have to include not only the external load, but also the initial load of the leg. Diagram of the load and support of the discussed mechanical system is presented in Fig. 4.

To obtain the most disadvantageous variant of the leg’s load, it was assumed that initially there is a nominal static pressure in annular chamber equal to 35 MPa, i.e. the pressure at which the leg’s lowering starts due to an action of operational valve. In the discussed problem, the nominal pressure of the working medium was included to a dynamical analysis by defining the initial liquid density $\rho_0$. EOSPOL balance equation, which is a polynomial density function, is used to calculate the $\rho_0$ value. In case of (liquid) compression, it has the following form [6-9]:

$$p = a_1 \rho m + P_0$$

(1)

where:

- $P_0$ – pressure, Pa,
- $\rho$ – material density, kg/m$^3$,
- $\rho_0$ – wanted material density, kg/m$^3$,
- $a_1$ – volume elasticity modulus, Pa.
medium will increase from 0 to 35 MPa and will stabilize on this level. To protect upper prop against coming out from the cylinder, due to increasing pressure of the working medium, the additional surface (a limiting wall against which the upper prop is leaning – Fig. 4) was modelled. Additionally, not to allow for reciprocal penetration of upper prop components and wall, the proper type of contact should be defined (e.g. Master-Slave Surface).

External load of the hydraulic leg was defined in the form of forcing the leg’s head movement (Fig. 4). Time process of dynamic load was selected in order to achieve dynamic overload – pressure increase in annular chamber equals 1.7 times of the nominal pressure, after stabilization of the working medium pressure, on the level of the nominal pressure $p_n = 35$ MPa. Time of dynamic load increase and drop, for the time process given in Fig. 5, was accepted based on time processes of dynamic load realized on a test stand by ignition of explosives [11].

The final stage of analysis preparation included a generation of the input file, based on the following parameters of analysis:
- time of analysis (0.085 s),
- starting time interval (1E-7 s),
- minimal time interval (1E-8 s),
- file of results (pressure, dislocation, deformation, stress).

The generated input file, which included information about geometry, loads, boundary conditions and parameters of analysis, was loaded to MSC.Dytran (solver). MSC.Dytran calculation process verifies in the first step the following:
- geometrical correctness of the system,
- correctness of boundary conditions,
- correctness of defined contacts between the neighboring elements.

In the next step, the file is analyzed numerically. The software has to read the required values every 0.0001 s and to save them in proper ARC or THS results files.

Analyzed system was built from 54757 finite elements (flat and solid) and resulting 50113 nodes. Numerical analysis of the leg’s dynamic load phenomenon, lasting 85 ms, was carried out in MSC.Dytran within 17 hours.

### 3. Results of Numerical Calculations

Time processes of liquid pressure in the leg caused by a dynamic load were recorded in selected measurement points marked with P.0 – P.5 symbols in Fig. 6.

Fig. 6 Arrangement of measurement points on the model of hydraulic leg [5]

Fig. 7 Time processes of pressure in the model of liquid [5]

Fig. 8 Time processes of upper prop yielding and reduced stress in the element situated on the cylinder surface [5]

Fig. 9 shows a distribution of the reduced stress in steel subassemblies of hydraulic leg (upper prop, lower prop) which occurs in the result of dynamic load and working pressure of liquid in selected time intervals (from 0.02 s to 0.027 s).

Fig. 9 Distribution of reduced stress in steel subassemblies of hydraulic leg, time interval: a) 0.02 s – beginning of dynamic force action, b) 0.024 s – maximal dynamic force, c) 0.027 s – pressure drop to the initial value [5]
4. Analysis of Interaction between Steel Components, Liquid and Gas on an Example of Gas Accumulator Installed in the Hydraulic Leg

Application of gas accumulator is one of the methods for protecting the hydraulic leg against results of dynamic action of a rock mass. That device is particularly suitable to be used in conditions of a dynamic load characterized by a very short time of load increase, due to its small inertia and little possibility of rigidity changes in a wide range. Principle of operation of such a device consists in a significant compression of gas closed in the accumulator during the dynamic load action on the leg and return back to the initial dimension by accumulator and the leg when the dynamic load stops.

The hydraulic leg, equipped with gas accumulator (designed by KOMAG) (A), installed in the upper prop (B) of the leg, consists of cylinder (1) and piston (2) (equipped with sealing and guiding rings), installed inside the upper prop (B) (Fig. 11). In the leg’s head (8) there is a gas valve (4), which supplies accumulator gas chamber (3) through channels and a pipe (7). Liquid (5) is in a space confined by the piston and cylinder walls (9) and the leg’s foot (10).

A simplified geometrical model of the hydraulic leg, equipped with gas accumulator, was created to show possibilities of operation of three technical media (steel, liquid, gas) together. Geometrical model of the hydraulic leg, presented in Fig. 12, consists of cylinder (1) and upper prop (2) protected by a limiting wall against coming out (3). In the cylinder (1) there is a hydraulic liquid (4) which presses a piston (6) of accumulator filled with gas (5). Both liquid (4) and gas (5) are under suitable pressure.

Numerical analysis of the dynamic load of the leg with the gas accumulator, containing information about geometry, load cases, boundary conditions and parameters of analysis, was prepared similarly to the numerical analysis discussed in the previous chapter of this paper. Additionally, properties of gas, which fills the chamber, were defined. The gas was modeled using the elements of the CHEXA type. The following material properties were attributed to those elements (which characterize the ideal gas): density $\rho = 1.2887 \text{ kg/m}^3$, material constant $\gamma = 1.4$, internal energy in a temperature $293 \text{ K}$ $e = 1.94 \times 10^5 \text{ J/kg}$.

Assumed working pressures of liquid (35 MPa) and gas (40 MPa) in leg’s annular chambers and gas accumulator were entered to the numerical analysis by defining the initial densities of both media. In case of the gas accumulator, the E O S G A M balance equation, having the polynomial density function, is used to calculate the initial gas density $\rho_0$: \[ p = (\gamma - 1) \rho_0 e \] where: $p$ – pressure, Pa, $\rho_0$ – wanted material density, kg/m$^3$, $\gamma$ – material constant, $e$ – internal energy, J/kg.

Time processes of momentary pressure in Eulerian parts in the leg loaded dynamically, were recorded in selected “measurement points” marked in Fig. 13. Cells of liquid were marked with red color: P1 – measurement of pressure under piston and P2 – in lower prop’s bottom, and cells of gas were marked with blue color: P3 – measurement of pressure under accumulator’s piston and P4 – in accumulator’s bottom.

Calculated time processes of the medium’s working pressure in an annular compartment of the leg loaded dynamically, are presented in Fig. 14.

Calculated time processes of gas pressure in gas accumulator chamber loaded dynamically, are presented in Fig. 15.

In Fig. 16, pressure distribution in liquid and gas models – resulting from a dynamic load exerted on the initially set hydraulic leg, is presented in the selected time intervals (from 0.01 s to 0.0175 s).
Maps of reduced stresses in steel components of leg and gas accumulator in selected time intervals (from 0.01 to 0.018 s) are given in Fig. 17. Deformation of models and dislocation of gas accumulator piston were considered.

Map of plastic strains at the moment (0.015 s) of maximal load exerted on the leg, by a dynamic force, is presented in Fig. 18a. During the analysis in time interval from 0 s to 17 ms, there were no plastic strains in the leg.

Fig. 18. Map of plastic strains in leg: a) zero plastic strains at the moment of maximal dynamic load, b) plastic strains resulting from accumulator’s piston impact on protecting flange after gas decompression [6]

5. Summary

Problems that condition application of finite elements method for modelling the action of dynamic load on hydraulic leg were the subject of this paper.

Modelling of interaction between three different media – steel components, liquid and gas is the main problem. In the FEM method, each of these media is described in a different way: steel leg’s components by finite elements of Lagrange type, and fluid by finite elements of Euler type. Mutual interaction of those media were modelled by entering boundary conditions on coupling surface – the so called “skins”, made by surface finite elements (Lagrange elements).

The other problem is how to include an initial static setting of the leg to the dynamic load calculations. Initial static load was modelled by assuming the initial density of liquid, which depends on initial static pressure and physical properties of media. Besides, the leg dislocation, in the direction of its longitudinal axis, was one-sidedly limited assuming a rigid resistance plane as well as assuming a proper type of its contact with leg’s head. In an analysis of dynamic load, it was assumed that the time indispensable to reach the accepted leg’s static load is 20 ms. Besides, during the analysis of dynamic load simulation – pressure time processes in selected cells, reduced stresses and dislocations in pointed finite elements and leg’s nods – it was found that there is a necessity to smooth the obtained time processes using a progressive mean coefficient. Other method for eliminating imperfections of initial leg’s static load modelling process, would consist in a preparation of special procedures for users and thorough interference in the structure of input file of the MSC.Dytran calculation software.
6. Bibliography


APPENDIX 5

DETERMINISTIC AND PROBABILISTIC STRENGTH ANALYSIS OF THE CHAIN COUPLING LINKS

This appendix is focused on the deterministic and probabilistic strength analysis of chain coupling links in the mines.
APPENDIX 6

PRESSURE DISTRIBUTION IN THE CONTACT BETWEEN MINING SUPPORTS AND FOOT-WALL AS A PROBLEM OF 3D BODY ON ELASTIC FOUNDATION

This appendix is focused on the pressure distribution calculations in the contact between mining supports and foot-wall as a problem of 3D body on elastic foundation.

APPENDIX 7

MEASURING THE PULLING FORCE IN THE CHAIN OF THE TRAVEL GEAR OF THE DYNARIDE SYSTEM IN THE SHEARER EICKHOFF SL 500

This appendix deals with safety problems of chains of travel gears of the Dynaride system in shearsers. Measurements of the pulling force, including methods for measurement and evaluation, are described here. At the end, the determination of the safety coefficient \( \beta \) and its keeping are discussed. Authors: Horst Gondek (VŠB – Technical University of Ostrava, Czech Republic), Daniela Marasová (Košice University of Technology, Slovakia) and Lubomír Schellong (Lazy Mine, o.z., Ostrava-Karviná Mines, a.s. Orlová-Lazy, Czech Republic).

1. Introduction

On the basis of the decision of the Czech Mining Office on trial operation, requirements were laid down for determining the travel gear safety with the Dynaride system used in the shearer Eickhoff SL 500. For this reason a methodology for measurement was developed and measurements of the pulling power of the travel gear of the above-mentioned shearer Eickhoff SL 500 were taken directly under operational conditions.

The goal of these measurements was to determine the maximum magnitudes of forces produced in the chain of the RHINORIDE-Kette 38/42 type in various operation modes of the shearer and to determine its safety.

2. Measurement Methodology

On the chain with dimensions, as given in Table 1, tensometers were placed in order to measure a relative elongation that corresponded to the pulling force.

Tab. 1 Layout of Tensometers

<table>
<thead>
<tr>
<th>Sensor</th>
<th>Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>20 m before the shearer</td>
</tr>
<tr>
<td>2</td>
<td>6 m before the shearer</td>
</tr>
<tr>
<td>3</td>
<td>6 m behind the shearer</td>
</tr>
</tbody>
</table>

Measurements were made under the following conditions:

<table>
<thead>
<tr>
<th></th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Initial stressing force at chain tensioning, ( v = 0 \text{ m/s} )</td>
</tr>
<tr>
<td>2</td>
<td>Initial stressing force at full loading, ( v = 0 \text{ m/s} )</td>
</tr>
<tr>
<td>3</td>
<td>Backward non-load running of the shearer, ( v = 8 \text{ m/s} )</td>
</tr>
<tr>
<td>4</td>
<td>Forward run of the shearer into the pillar, ( v = 8 \text{ m/s} )</td>
</tr>
<tr>
<td>5</td>
<td>Run of the shearer into the pillar forward and backward (coal on the conveyor), ( v = 8 \text{ m/s} )</td>
</tr>
<tr>
<td>6</td>
<td>Backward run of the shearer (coal on the conveyor), ( v = 16 \text{ m/s} )</td>
</tr>
<tr>
<td>7</td>
<td>Forward run of the shearer without mining (coal on the conveyor), ( v = 8 \text{ m/s} )</td>
</tr>
</tbody>
</table>
With reference to the chain design and adjustment, the location of the tensometer (strain gauges) on the inner side of the horizontal chain link was chosen (Fig. 2).

**3. Load Force – Relative Elongation Relation**

As for the place of the tensometer location, the relative elongation shows that it is produced not only by tensile stress but also by the bend of the link, which, from the point of view of elasticity and strength, manifests itself as a thick curved bar; the location of the tensometer is very unfavourable. This was the reason for carrying out the study of stress conditions on the link by the finite element method. The link was loaded with the force \( F_V = 500 \text{ kN} \); the force in the point of action of sequential links had the distribution of the cosinusoidal form (Fig. 3).

The resultant value of the relative elongation in the points of sensor placement (see Fig. 4) is \( \varepsilon_V = 3.421\% \). Based on this result, a ratio of the load force to a relative elongation may be determined, \( k_V = 146.16 \text{ kN/\%} \).
4. Measuring Voltage – Relative Elongation Relation

For the used tensometric amplifiers, the conversion from the measured voltage $U_m$ [V] into the relative elongation $\varepsilon$ [%] is determined according to the following relation:

$$\varepsilon = U_m \cdot \frac{C \cdot 4}{k \cdot n \cdot A}.$$

Where: $C$ is the calibration value of the amplifier used [mV/V]; $k$ is the tensometer constant, $k = 2.02$; $n$ is the number of active tensometers, $n = 1$; $A$ is the value of measuring voltage at switching on the calibration switch [V].

In the set of the amplifiers used for measurement, the resultant values are as follows:

<table>
<thead>
<tr>
<th>Point of measurement</th>
<th>C [mV/V]</th>
<th>k</th>
<th>n</th>
<th>A [V]</th>
<th>$\varepsilon$ [%/V]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.2</td>
<td>2.02</td>
<td>1</td>
<td>0.380</td>
<td>1.070</td>
</tr>
<tr>
<td>2</td>
<td>0.2</td>
<td>2.02</td>
<td>1</td>
<td>0.498</td>
<td>0.747</td>
</tr>
<tr>
<td>3</td>
<td>0.1</td>
<td>2.02</td>
<td>1</td>
<td>0.400</td>
<td>0.495</td>
</tr>
</tbody>
</table>

5. Measuring Modes and Measurement

Measurements were taken in the following modes:

<table>
<thead>
<tr>
<th>Record</th>
<th>Mode</th>
<th>Graph</th>
</tr>
</thead>
<tbody>
<tr>
<td>RET_03</td>
<td>Initial stressing force at chain tensioning, $v = 0$ m/min</td>
<td>A</td>
</tr>
<tr>
<td>RET_05</td>
<td>Initial stressing force at full chain loading, $v = 0$ m/min</td>
<td>B</td>
</tr>
<tr>
<td>RET_06</td>
<td>Backward non-load running of the shearer, $v = 8$ m/min</td>
<td>C</td>
</tr>
<tr>
<td>RET_07</td>
<td>Forward run of the shearer into the pillar, $v = 8$ m/min</td>
<td>D</td>
</tr>
<tr>
<td>RET_08</td>
<td>Run of the shearer into the pillar forward and backward (coal on the conveyor), $v = 8$ m/min</td>
<td>E</td>
</tr>
<tr>
<td>RET_09</td>
<td>Backward run of the shearer (coal on the conveyor), $v = 16$ m/min</td>
<td>F</td>
</tr>
<tr>
<td>RET_10</td>
<td>Forward run of the shearer without mining (coal on the conveyor), $v = 8$ m/min</td>
<td>G</td>
</tr>
</tbody>
</table>

Zero values (values of the non-load chain) were determined according to the values that had been measured at maximum loading (for sensor 3 behind the shearer) and for the backward run (for sensors 1 and 2 before the shearer). The result values and record curves of measurements are given below in the following graphs:
Backward non-load running, \( v = 8 \text{ m/min} \)

Graph C

Forward run of the shearer with mining, \( v = 8 \text{ m/min} \)

Graph D

Forward run with mining and backward run (coal on the conveyor), \( v = 8 \text{ m/min} \)

Graph E

Backward run (coal on the conveyor), \( v = 16 \text{ m/min} \)

Graph F
3. CONCLUSION

The measurements of the pulling force in the chain of the travel gear with the Dynaride system in the shearer Eickhoff SL 500 have confirmed the theoretical pulling force of the machine.

Moreover, the real measurements have also confirmed the co-acting of various influences which - in the course of measuring in situ - affect the maximum attainable pulling force. The above mentioned influences include the following items:

- locking the mining system (the pulling force does not act in the shearer axis)
- the influence of the chain guidance with reference to the conveyor curvature
- the non-homogeneity of the coal pillar does not enable any absolute measurement at the zero velocity
- resistances to the proper guidance of the mining system.

With regard to the above-mentioned enumeration of passive resistances, the obtained results of the measurement correspond to the data from the producer and may be taken into account when calculating the safety of the Dynaride pulling chain.

Based on the confronted data from producers of the Dynaride chains, it is evident that with the existing known and guaranteed technologies, safety calculations may be based on the guaranteed value of the chain strength of 1900 kN. At proper testing of the pulling chain, the values of 2076, 2051 and 1992 kN were acquired. As for these values, the certificate was issued by the chain producer.

In view of the measured values, one may state that the safety was secured in case of the run of the shearer when mining with the velocity of 8 m/min, when the pulling force of 680 kN was measured.

Thus, the safety coefficient $\beta$ is as follows:

$$\beta = \frac{1900}{680} = 2.79$$

which is the value that is admissible for the chain of the travel gear with the Dynaride system in the shearer Eickhoff SL 500.
APPENDIX 8

OBSERVATION ON HYDRAULIC POWERED ROOF SUPPORT LOCATED IN FACES WITH EARTHQUAKE

This appendix is focused on the analysis of information obtained in the course of the earthquake phenomena and their effects investigation in the OKR (mainly in faces, indicating the importance of the hydraulic powered roof supports). It deals with some knowledge of basic support bends designed for working in such conditions. Authors: Horst Gondek (VŠB-Technical University of Ostrava, Czech Republic), Arnošt Ševčík (VŠB-Technical University of Ostrava, Czech Republic).

Introduction:

Technical literature [1] mentions the results of earthquake phenomena analysis in the OKR (Ostrava-Karviná District) during the period of 1989-2000, which provides the following significant facts related to hydraulic powered roof supports:

- In the monitored period, 72 earthquakes were recorded, 34% of which was with the seismic energy of \(10^7\) J, with 24% approx. \(10^3\) J and with 6% - \(10^1\) J, the top was 2.3 \(10^1\) J. Other earthquakes were of a lower energy.
- Much stronger earthquakes were in faces than in opencasts. This is brought about especially by greater disturbance to the working coherence in the faces.
- Earthquakes most frequently occur in faces with a firm top, and mainly in seams with thickness of over 4 - 6m.
- In addition, an examination of length of the damaged face following an earthquake is also interesting. With energy of \(10^1\) J it was 50 ÷ 120 m, with energy of \(10^2\) J 50 ÷ 100 m, and with energy of \(10^3\) J mainly 40 ÷ 60 m.
- The above-mentioned facts are of interest but they are not sufficient for a unique determination of the way the earthquake’s load transfers to the supports and how it distributes along the length of the damaged face section.

As the analyses imply, it is possible to conclude that the earthquake occurrence with the energy of and over \(10^3\) J is significant and therefore it is necessary to pay relevant attention to the supports designed for work in such conditions, and the supports section should bear the energy of at least \(10^3\) J without any serious effects.

Since 1993, in the OKR – Lazy Mine, there has been a complex in the seams with thickness of 6.0m, which has gone through many earthquakes and thus on the basis of its operation it is possible to summarize some observation to do with the WS1.7 supports resistance, whose main technical parameters are as follows:

| Supports vertical range | 2.8 ÷ 6.0 m |
| Section span in the face | 1.5m |
| Section interval | 0.8m |
| Supports resistance | 739 ÷ 1042 kN.m⁻² |

as for the supports of individual parts.

The supports have a classical arrangement suitable for mining of high coal seams, and their main parts are as follows:

- one-piece foundation frame,
- follow-up bars of the lemniscatic mechanism,
- massive shield,
- 2-piece head piece draft with an extension and pillar support,
- two hydraulic props with two telescopes,
- angular cylinders of the main head piece,
- control cylinders of other head piece parts,
- ram,
- hydraulically powered head piece and shield lining.

With regard to the well-designed parts of the supports steel frame (according to EN - \(\sigma_{\text{def}} = \frac{R_{\text{f}}}{1.5}\)) in the course of the operation there have been no significant failures as for the steel parts, even if the support has been in operation for over 10 years, and from the faces more than 5.5 million tons have been mined out.

A greater attention should be given to the protection of hydraulic elements of the supports from the pressure bumps effects. Among the elements there are:

- angular cylinder to control the main head piece
- hydraulic coupling

Head piece angular cylinder

There are two of these cylinders in the supports; they are double-acting with diameters of 140/80mm, with a stroke of 400mm and they are protected by double-sided safety valves. The cylinder must be sustained from an overload, both with supports head piece control (mainly clamping of props) and during a dynamic load. In both cases, the speed of engaging and disengaging the cylinder depends on the kinematic system structure (shield – head piece – prop – angular cylinder) and the speed of head piece movement. For the given system in the WS 1.7 supports, what is suitable are the safety valves with a flow rate of Q = 200 dm³/min¹. Valves with a lower flow rate would have an adverse effect on the system operation; especially during the dynamic load as the interior volume of liquid is small and thus with no elasticity. There is a considerably better situation with hydraulic props.

Hydraulic props with pulse valves

The supports have two hydraulic props of the following parameters:

- prop power at \(p = 45\) MPa 3180 kN
- 1st degree piston diameter 300 mm
- 2nd degree piston diameter 225 mm
- 2nd degree stroke 1574 mm
- maximum space under pistons 175 dm³

The hydraulic prop has a considerably large interior volume, which is very advantageous during the dynamic load. In addition, the working height of the supports ranges during operation from 4.5 to 5.5 m, i.e. in the range of 75 ± 91% of the maximum height. The prop was secured by a safety valve with a flow rate of 1250 dm³/min⁻¹. It appeared that mainly due to a great resistance of the supports as well as extraordinary elasticity of the hydraulic prop there were only rare breakdowns of the valves. However, an increase in a pulse-valve flow rate had been considered and in 2002 the MPA Dortmund test room tested a prop with a flow rate valve of 5500dm³/min⁻¹.

The hydraulic prop was clamped into a test frame (for the diagram see Figure 1) in a way the bottom pillar was released by 790 mm and the top pillar by 610mm, which represents about 44% of the total range of the lift. The prop clamping pressure was 28 MPa, which is
87% of the maximum clamping pressure. The prop was gradually loaded with a load drop of 20,000 kg and energy of 100, 200 and 300 kNm. In the course of the test, pressure was monitored depending on the time and its course is described in Figures 2, 3 and 4. The safety valve with a maximum flow rate of 5500 dm$^3$/min was set for the pressure of $p_e=45$ MPa.

![Figure 1 Test Frame Diagram](image1)

*Figure 1* Test Frame Diagram
1 – test frame, 2 – load, 3 – hydraulic prop, 4 – pulse valve, H – load height

![Figure 2 E = 100 kNm](image2)

*Figure 2* $E = 100$ kNm

![Figure 3 E = 200 kNm](image3)

*Figure 3* $E = 200$ kNm

In order to evaluate the system function (hydraulic prop – safety valve) some values determined by measurements have been used as well as values calculated on the basis of the acquired values. They are as follows:

- $p_u$ clamping pressure of the prop (28 MPa)
- $p_e$ nominal pressure of the safety valve (45 MPa)
- $p_1$ maximum pressure in the prop during the test
- $p_2$ residual pressure after the test and deduction of the static pressure (2.3 MPa)
- V liquid loss volume during the test
- $E$ load potential energy (kNm)

Values are stated in Table 1.

<table>
<thead>
<tr>
<th>$E$ (kNm)</th>
<th>$p_1$ (MPa)</th>
<th>$p_2$ (MPa)</th>
<th>$p_1/p_2$</th>
<th>V (dm$^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>57.2</td>
<td>21.5</td>
<td>2.7</td>
<td>0.46</td>
</tr>
<tr>
<td>200</td>
<td>72.2</td>
<td>12.9</td>
<td>1.61</td>
<td>1.07</td>
</tr>
<tr>
<td>300</td>
<td>88.2</td>
<td>2.4</td>
<td>1.96</td>
<td>1.81</td>
</tr>
</tbody>
</table>

If significant values arising from the tests with three different energies are evaluated, it is necessary to point out an important fact that not even with the energy of $3.10^5$ J (300 kNm) the pressure did not exceed twice the nominal pressure and after the tests in the prop there was still residual pressure. The values of liquid loss volume are relevant to the amount the pulse safety valve released from the prop during the test. This amount was also sufficient for eliminating the given dynamic load.

**Conclusion:**

The standing operation of the WS1.7 supports as well as the carried out tests of the pulse valve hydraulic prop both lead us to some new observations in connection with mechanized supports for faces with earthquake, e.g.:

- The steel construction needs to be rated with $1.5 \div 1.7$-fold degree of safety to the slide of the applied material.
- In order to protect the angular cylinders to control the head piece, it is necessary to select safety valves with a sufficient flow rate, which shall protect them both
During dynamic load and during the supports control, mainly during clamping the hydraulic props.

- With the hydraulic prop it is necessary to take into account 2.0 degree of safety to the slide of the applied material.
- The hydraulic props need to be protected by pulse valves which during the dynamic load with the energy of (2.5 ÷ 3) \(10^5\) J do not create maximum pressure in the prop, which would be of a value of over twice the nominal pressure.
- The pulse valves must have a rapid reaction on the opening as well as shutting, an ability to release as largest amount of liquid as possible out of the prop in a short time. Though, the amount of liquid released out of the prop must correspond to the energy value affecting the prop.
- In case the amount of the released liquid was too high, in the operation it would lead to a loss of contact of the supports with the roof.
- Depending on the seam thickness, the mechanized supports must have a sufficient resistance and good elasticity of the hydraulic props.

**Bibliography**


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**APPENDIX 9 Optimisation of Belt Conveying Transport Routes**

This appendix is focused on the optimisation of belt conveying transport routes. Authors: Horst Gondek, Stanislav Budirský, Jan Šamárek (Czech Republic).

### 1. Introduction

Defects on conveyor belts and hereto-related downtimes have a significant impact on transport economics in mining enterprises. The defect rate of a transport route depends on the number of conveyor belts involved and, therefore, since the likeness of defects to happen decreases with the decreasing number of conveyor belts involved, there is a continuous, logical concern in the extension of individual conveyor belts in the route. Decrease in the number of conveyor belts involved also implies the reduction of discharge points i.e. the spots where the belt is subject to damage most frequently.

Different requirements exist for conveyor belts arranged behind the under-face mechanism and those in the main route. The variance of requirements posed to the conveyor belts in the route requires individual procedures to be followed during projecting the conveyor belts while maximum unification of their elements and other machine units is utilised. The mine designer should be allowed to design the whole transport route in such a way so that it reflects optimally the input parameters such as the traffic capacity, route- and site configuration and/or simultaneous loading. One of the sources, which you should use while doing so is the Directions for use, which, however, provide only a static information about the conveyor belt. Therefore, an offer should be made to designers concerning an additional tool which provides information about the measures which should be adopted when selecting and locating assembly parts of a conveyor belt if even a single input parameter is changed. In this paper the assembly parts include the belt, the driving station and the tension unit.

This “dynamic” tool is represented by a mathematical model of calculation of conveyor belt parameters in the Excel spreadsheet program. Developments in the programming potential reduce the time required for evaluation of individual options of transport route configuration. Thus, the customer-company designer can make a real idea of possible options of the transport route configuration which will be completed into the final form of individual conveyor belts by the manufacturing-plant designer.

A second benefit of the program is a rapid, multivariate calculation of newly configured conveyor belts, which may be required due to advancement of the mining works causing changes in the tilt, length and capacity of conveyor belts.

### 2. Getting Acquainted with the Program

Operational length of a conveyor belt depends on a multitude of input parameters. Gathering of these parameters is a time-consuming task. The parameters can be subdivided into those being dependent on the machine and belt properties and those being dependent on the operational location. As you can see in the Table 1 showing input calculation parameters according to ČSN ISO 5048, since this is a unified machine, many of these parameters can be entered into the program as fixed ones. These areas are locked and they cannot be modified, nor the predictive ability of the program for a particular machine can be affected.

In the first column of the Table 1, you can see the extent of input variables, which are fixed in this way.
Tab. 1

<table>
<thead>
<tr>
<th>Modification by users not accessible</th>
<th>Modification by users not accessible</th>
<th>Modification by users allowed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency of driving unit for driven conveyor belts</td>
<td>Transport capacity</td>
<td>Conveyor belt length</td>
</tr>
<tr>
<td>Efficiency of driving unit for braked conveyor belts</td>
<td>Conveyor belt speed</td>
<td>Tilt angle of the conveyor belt in direction of movement</td>
</tr>
<tr>
<td>Weight of rotating parts of rollers on the upper branch support 1</td>
<td>Velocity component of transported mass in belt movement direction</td>
<td>Total coefficient of friction</td>
</tr>
<tr>
<td>Weight of rotating parts of rollers on the lower branch support 1</td>
<td>Conveyor belt density</td>
<td>Total coefficient of friction for braked conveyor belts</td>
</tr>
<tr>
<td>Spacing between supports in the upper branch</td>
<td>Allowed tensile stress of the belt</td>
<td>Location of driving unit</td>
</tr>
<tr>
<td>Spacing between supports in the lower branch</td>
<td>Relative elongation at allowed stress</td>
<td>Location of tension unit</td>
</tr>
<tr>
<td>Coefficient of friction between transported mass and the side part</td>
<td>Number of electric motors installed</td>
<td>Position of tension unit relative to the driving unit</td>
</tr>
<tr>
<td>Coefficient of friction between transported mass and the belt</td>
<td>Number of brakes</td>
<td>Tilt weight of rock</td>
</tr>
<tr>
<td>Clear width of side conveyance</td>
<td>Selection of tension strength for passive tension unit</td>
<td></td>
</tr>
<tr>
<td>Length of side conveyance</td>
<td>Selection of tension strength for active tension unit</td>
<td></td>
</tr>
<tr>
<td>Number of belt cleaners</td>
<td>Conveyor belt start time</td>
<td>Conveyor belt braking time</td>
</tr>
<tr>
<td>Resistance due to belt bending on drums and resistance in drum bearings</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Conveyor belt width</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Driving drum diameter</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of driving drums</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coefficient of friction between the drum and the belt (soiled side of the belt)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coefficient of friction between the drum and the belt (clean side of the belt)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angle of contact on the driving drum 1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angle of contact on the driving drum 2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Static protection against belt creep on drum</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Electric motor speed</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Braking torque on one brake</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Moment of inertia of a typical drum</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Moment of inertia of electric motor</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Moment of inertia of clutch</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Moment of inertia of gear-unit</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Maximum allowed belt sagging</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Starting coefficient</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Since the program solution conforms to the ČSN ISO 5048 standard, which does not involve conveyor belts with irregular route profiles, further simplification can be achieved. However, the program can also be used for calculation of these conveyor belts, but in such a case interventions in the locked area are required to be made and designers of the conveyor belt manufacturer, i.e. the designers of this program, should be consulted.

The program is written as a spreadsheet file, in which individual sheets represent individual types of conveyor belts. The designer will select a conveyor belt sheet to comply as much as possible with the required transport capacity. Then the designer enters mandatory input parameters, which define the location of engagement and the present-day or future operational environment of the conveyor belt. These parameters also include the belt parameters. The program offers the most frequently used types of belt to be selected. Subsequently, the program gives the following information:

- calculation of conveyor belt parameters for the required capacity and acquisition of various control parameters such as the power requirement, coefficient of conveyor belt security and minimum value of belt prestress for various types of tension units;
- immediate change of parameters for the required capacity according to the location of driving units and depending on the number and type of driving drums, starting method, type of tension unit, etc.;
- using these information the program enables planning of renovation works, investment, expected purchase of the mechanical parts as well as the electrical units parts such as cable length, number of transformers, etc.;
- the program allows the mine operating staff to conceive the requirements for a new machine.

To monitor an immediate change without the necessity to move the cursor around the table the designer may copy the cell containing the result, which is the most important for him, to free cells located below the area of input data entry. By unlocking the input data area defined in the machine / belt design and by their modification you can use the mathematical model described above to make a solution of conveyor belts with a variable tilt, more than one loading point and variable loading with mine run all along their length. Being beyond the scope of a normal user these issues are not dealt with in the present paper any more.

3. Example of a Driving Unit Location and its Bearing on the Conveyor Belt Tension

A part of the conveyor belt calculation for the inclined transport is shown in Tab. 2. In this arrangement of the conveyor belt the tension unit is located at the point of minimum tension in the belt, i.e. behind the discharge drum. It should be noted that this is the true tension unit capable of maintaining the pre-set tension in the belt and not the belt charger, which, in domestic mines, is incorrectly labelled as tension unit. The driving station is located in front of the conveyor belt return drum (see the diagram in Figure 1).
Tab. 2 Tension unit located near the discharge station and driving unit located near the return station

<table>
<thead>
<tr>
<th>No.</th>
<th>Name</th>
<th>Sign</th>
<th>Calculated value</th>
<th>Unit</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Transport capacity</td>
<td>Q</td>
<td>1100</td>
<td>t.h⁻¹</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Conveyor belt length</td>
<td>L</td>
<td>420</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Belt speed</td>
<td>v</td>
<td>2.5</td>
<td>m.s⁻¹</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Tilt angle of conveyor belt in direction of movement</td>
<td>δ</td>
<td>12</td>
<td>°</td>
<td>Inclined transport</td>
</tr>
<tr>
<td>67</td>
<td>Recommended installed power</td>
<td>P</td>
<td>294</td>
<td>kW</td>
<td></td>
</tr>
<tr>
<td>118</td>
<td>Select total tension on the tension unit with non-stationary drum NZₐ</td>
<td>NZₐ</td>
<td>24500</td>
<td>N</td>
<td></td>
</tr>
<tr>
<td>156</td>
<td>Anti-creep security on the driving drum during steady-state condition</td>
<td>kₑ</td>
<td>4.16</td>
<td></td>
<td>≥ 1.3</td>
</tr>
<tr>
<td>157</td>
<td>Anti-creep security on the driving drum during start</td>
<td>kᵣ</td>
<td>9.72</td>
<td></td>
<td>≥ 1.3</td>
</tr>
<tr>
<td>158</td>
<td>Anti-creep security on the driving drum during braking</td>
<td>kᵦ</td>
<td>3.02</td>
<td></td>
<td>≥ 1.3</td>
</tr>
</tbody>
</table>

The calculation program will show us how this favourable situation (see coefficients kₑ, kᵣ, kᵦ) turns for the worse due to relocation of the tension unit. Table 3 shows a part of the calculation of the same conveyor belt with the tension unit located before the driving unit near the return drum (see the diagram in Fig. 2). As we can see in the Table 3, a mere relocation of the tension unit while maintaining the same tension strength results in a drop in the belt-creep coefficients on the driving drum below the allowed values for steady-state and braking. If the conveyor belt were designed in this way, a belt creep would occur on the driving drums in the steady-state condition and in the transitional state, i.e., during braking, which would result in substantial reduction of the driving system service life. The designer has the opportunity to select the driving system and locate the individual units in such a way as to ensure its correct function and increase the service life especially of the belt and the driving system. Thus, it is possible to cut down significantly the mining engineer’s unceasing complaints of the wear of the belt and its joints and gearbox defects.

Tab. 3 Tension unit and driving unit located near the return station

<table>
<thead>
<tr>
<th>No.</th>
<th>Name</th>
<th>Sign</th>
<th>Calculated value</th>
<th>Unit</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Transport capacity</td>
<td>Q</td>
<td>1100</td>
<td>t.h⁻¹</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Conveyor belt length</td>
<td>L</td>
<td>420</td>
<td>m</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Belt speed</td>
<td>v</td>
<td>2.5</td>
<td>m.s⁻¹</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>Tilt angle of conveyor belt in direction of movement</td>
<td>δ</td>
<td>12</td>
<td>°</td>
<td>Inclined transport</td>
</tr>
<tr>
<td>67</td>
<td>Recommended installed power</td>
<td>P</td>
<td>294</td>
<td>kW</td>
<td></td>
</tr>
<tr>
<td>118</td>
<td>Select total tension on the tension unit with non-stationary drum NZₐ</td>
<td>NZₐ</td>
<td>24500</td>
<td>N</td>
<td></td>
</tr>
<tr>
<td>156</td>
<td>Anti-creep security on the driving drum during steady-state condition</td>
<td>kₑ</td>
<td>1.10</td>
<td></td>
<td>≥ 1.3</td>
</tr>
<tr>
<td>157</td>
<td>Anti-creep security on the driving drum during start</td>
<td>kᵣ</td>
<td>2.18</td>
<td></td>
<td>≥ 1.3</td>
</tr>
<tr>
<td>158</td>
<td>Anti-creep security on the driving drum during braking</td>
<td>kᵦ</td>
<td>0.87</td>
<td></td>
<td>≥ 1.3</td>
</tr>
</tbody>
</table>
4. Examples of Options for the Method of Starting to Required Installed Power

During a conveyor belt start a dynamic force is added to the force, which is necessary for its steady-state run. The dynamic force is induced by the acceleration of rotating masses of the motor, drums and rollers in the route, and the advance masses of the belt and the rock. The dynamic force is therefore directly proportional to the reduced weight of the conveyor belt elements and to the acceleration of this weight. Table 4 shows a part of the calculation program, which indicates belt pulls during an uncontrolled conveyor belt start (no start time defined).

Tab. 4

<table>
<thead>
<tr>
<th>No.</th>
<th>Name</th>
<th>Sign</th>
<th>Calculated value</th>
<th>Unit</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>85</td>
<td>Actual acceleration of the conveyor belt mass</td>
<td>$a_0$</td>
<td>0.51</td>
<td>m.s$^{-2}$</td>
<td>satisfied</td>
</tr>
<tr>
<td>86</td>
<td>Actual start time</td>
<td>$t_a$</td>
<td>4.9</td>
<td>s</td>
<td>....</td>
</tr>
<tr>
<td>87</td>
<td>Selection of conveyor belt start time</td>
<td>$t_b$</td>
<td>4.9</td>
<td>s</td>
<td>selected</td>
</tr>
</tbody>
</table>

123 T1 - start | $T_1$ | 5000 | N | Lower branch |
124 T2 - start | $T_2$ | 12094 | N | Lower branch |
125 T3 - start | $T_3$ | 13143 | N | Upper branch |
126 T4 - start | $T_4$ | 46104 | N | Upper branch |
157 Anti-creep security of the driving drum during start | $k_d$ | 1.05 |          |          |

Table 5 shows the same part of the program for the same conveyor belt with electrically controlled start (start time defined). By the comparison of the two results you can immediately see the three benefits of the controlled start:

<table>
<thead>
<tr>
<th>No.</th>
<th>Name</th>
<th>Sign</th>
<th>Calculated value</th>
<th>Unit</th>
<th>Note</th>
</tr>
</thead>
<tbody>
<tr>
<td>87</td>
<td>Selection of conveyor belt start time</td>
<td>$t_b$</td>
<td>18</td>
<td>s</td>
<td></td>
</tr>
</tbody>
</table>
123 T1 - start | $T_1$ | 5000 | N | Lower branch |
124 T2 - start | $T_2$ | 9860 | N | Lower branch |
125 T3 - start | $T_3$ | 10888 | N | Upper branch |
126 T4 - start | $T_4$ | 34787 | N | Upper branch |
157 Anti-creep security of the driving drum during start | $k_d$ | 1.44 |          |          |

As you can see in the Table, the forces which the tension unit of the a) type exerts permanently on the belt, belt junctions and the conveyor belt structure are much higher than those exerted by the active tension unit of the b) type with the non-stationary drum. For the sake of better representation this result is shown graphically in the figure 3. In this way the designer can use this program to make an independent view on the required tension unit for the conveyor belt in question.

Conclusion

The above described procedure and examples demonstrate that even a simple mathematical model of the conveyor belt may contribute significantly to the optimisation of transport routes.
The designer of the unified-machine user is allowed to analyse factors of loading, simulate various operating conditions of the conveyor belt and select the most economical solution. It is no exception that mining companies purchase conveyor belts in a pell-mell way. However, where the use in the critical transport routes, in the so-called expressways, is expected, it is useful to make an optimisation using the above described program.

Thus, designers may avoid the typical signs of incorrectly configured conveyor belts in the route by using:

- too low or too high electric motor outputs;
- impractical conveyor belt starts;
- incorrect selection of belt parameters;
- uselessly high input pull values in the belt, etc.

By using this program both the manufacturer designers and the user designers may attain more economical transport routes.

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Ostroj Opava, a.s. literature (brochures), Czech Republic
BUCYRUS company literature (brochures), Germany
Hornonitrianske Bane zamestnanecká, a.s. (BME Nováky) literature (brochures), Slovakia.