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Andrzej ŚWIĄTONIOWSKI*, Ryszard GREGORCZYK*, Stanisław RABIARZ**

MODULAR MONITORING SYSTEM FOR PILGER MILL DRIVE

SUMMARY

Development of new steel types with improved fatigue parameters coupled with intensification of milling processes: raised productivity and deformation levels in the subsequent roll passes have led to increased loads which cause dynamic processes in the mill. As a result, the number of failures has vastly increased and the quality of rolled products has deteriorated. Attempts are made to overcome this problem by close monitoring of the working condition of the milling machine as well as the process parameters. The paper outlines a conceptual design of a monitoring system for the pilger mill drive for producing seamless tubes, showing the first stage of the project implementation.

Keywords: *pilger mill, dynamic load, monitoring*

SYSTEM MONITOROWANIA WALCARKI PIELGRZYMOWEJ DO WALCOWANIA RUR BEZ SZWU

Wprowadzenie nowych gatunków stali o podwyższonych własnościach wytrzymałościowych przy jednoczesnej intensyfikacji procesów walcowania – zwiększeniu prędkości i wartości odkształcenia w kolejnych przepustach – jest przyczyną wzrostu obciążeń i wystąpienia w układzie walcarki zjawisk o charakterze dynamicznym. W ich wyniku obserwuje się znaczący wzrost liczby awarii urządzeń oraz pogorszenie jakości wyrobów. Ograniczenia skutków tego typu zjawisk poszukuje się na drodze monitorowania stanu procesu i konstrukcji. W artykule przedstawiono koncepcję realizacji układu monitorowania pielgrzymowego walcowania rur bez szwu oraz pierwszy etap jego realizacji.

Słowa kluczowe: *walcarka pielgrzymowa, obciążenia dynamiczne, monitoring*

1. INTRODUCTION

The pilger hot rolling process involves the periodic forging of the tube walls with rollers of variable roll face profile (Fig. 1).

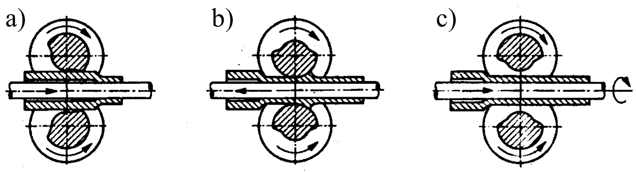


Fig. 1. Operating principle of a pilger mill (schematic diagram)
a) reduction of wall thickness; b) wall calibration;
c) tube release

Thus generated impact loads lead to vibrations within the mill structure. These negative phenomena seemed to enhance with intensification of rolling processes, introduction of new steel grades with improved fatigue parameters, when at the same time the productivity and deformation levels in the subsequent roll passes were raised. Another source of vibrations of the rolling machine are kinematic disturbances produced by excessively worn or damaged shaft bearings, gear mashing, couplings and other drive assemblies, responsible for fatigue damage or failure of machine components, at the same time leading to vast deterioration of the quality of rolled products, disqualifying them from the highly competitive market.

Despite major advancements made in the recent years, our knowledge about the nature of vibrations generated within rolling machines appears still insufficient to eliminate or, at least effectively limit, those negative impacts without jeopardising the mill productivity. That is why theoretical and experimental research is done to solve this problem and new solutions are sought by way of close monitoring of the plastic deformation processes and the working conditions of involved machines and assemblies.

The first diagnostic systems were introduced in the cold rolling plants producing steel sheets and belts, in the late 20th century, becoming now a standard component of the newly-constructed plants (Kazanecki 2003; Hardwick and Dunlop 1999).

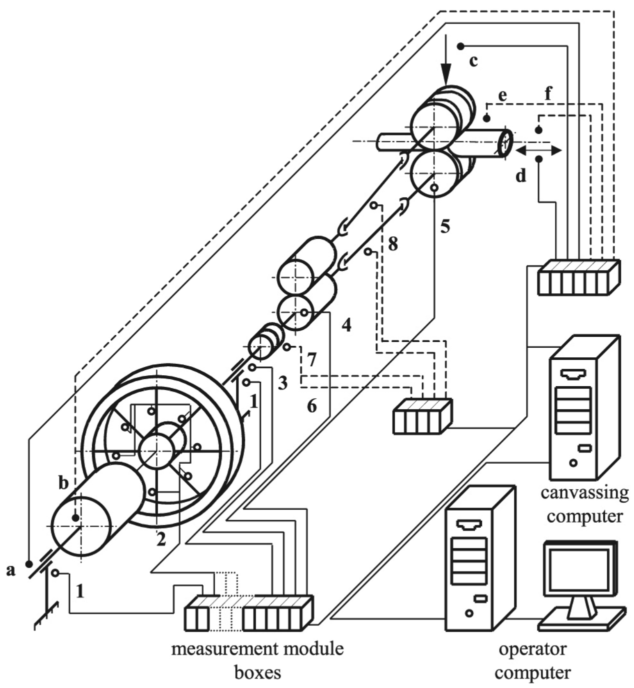
Monitoring systems were more difficult to implement in rolling mills producing long elements (rods, shaped profiles, buntons, rails) as well as tubes. The difficulties mostly arose due to a variety of process parameters and assortment of rolled products and due to the nature of involved dynamic processes.

2. STRUCTURE OF THE MONITORING SYSTEM

The purpose of this study is to design and implement a system for monitoring technological processes and the working condition of a pilger rolling mill for producing seamless tubes with the diameter ϕ 273–508 mm.

The conceptual design is the result of many years' expertise and research into plastic flow of tube walls in rollgaps, process disturbances and its effects, i.e. emergency conditions in the mill performance (Kazanecki 2003; IMŻ 2007; Świątoniowski and Marczak 2007).

* Department of Machines for Technology and Environment Protection, AGH University of Science and Technology in Krakow
** PWSZ Krosno



Step I	Step II
Vibration signals	
1. accelerometers on the main shaft bearing housing 2. systems of strain gauges on the flywheel arms 3. systems of strain gauges on the main shaft 4. accelerometers on bearing housings in the pinion stand 5. accelerometers on bearing housings in working rolls	6. accelerometers on the main coupling's housing 7. system of strain gauges on the main coupling components 8. system of strain gauges on spindle shafts and heads
Variable parameters of the process	
a) main shaft rpm b) current in the motor c) roll separating force	d) tube blank feeding frequency e) tube blank temperature f) loading of the feeding apparatus

Fig. 2. Schematic diagram of the system for monitoring of the mill condition and of the pilger hot rolling process

That applies particularly to the flywheel, the main drive coupling and spindle transmitting the torque from the pinion stand to the rolls and finally the bearings. These negative processes occur as forces and torque quickly increase when the rolling process is nearly over (as a result of relatively long rolling time and temperature decrease of a tube wall with a small mass and large surface area) and due to disturbed performance of the feeding apparatus.

Engineering of a system for monitoring the mill condition and the pilger rolling processes for producing seamless tubes (Fig. 2) shall proceed in two steps.

Parameters monitored in the first stage (Świątoniowski and Marczak 2007) include:

– bending stress in most stressed cross-sections of the flywheel's all arms[1)],
– radial and axial run-out of the flywheel rim,
– torsion moment in the main drive shaft between the coupling and the wheel hub,
– spectra of bearing vibrations in the main shaft and working rolls.

Pilger processes are used in production of tubes ranging in diameter and tube thickness, made of various steel grades, hence the wide range of operating loads and their variability throughout the process (Kazanecki 2003). As a consequence, mere recording of stresses in particular rims and arms (Fig. 3), though useful in assessing the deterioration of the quality of its coupling to the wheel hub – increased clearance levels (Świątoniowski and Marczak 2007) – fails to reliably alert us to the upcoming critical condition[2)].

[1)] The flywheel in the rolling mill with the mass 55.9 t and inertia moment 988 700 kgm^2 comprises a segment-structured rim 9000 mm in diameter connected via 8 arms (each having two spokes) to an iron hub mounted on the main drive shaft. The spoke ends are fixed in the hub sockets and in the rim with tapered plugs and pressed with bolts, passing through the plugs.

[2)] It is worthwhile to mention, that when clearances appear in the spoke-hub connections, the elasticity characteristics of those nodes shall be nonlinear and, in the consequence, vibrations of the shaft-flywheel system shall be nonlinear, too, making machine operation still more dangerous.

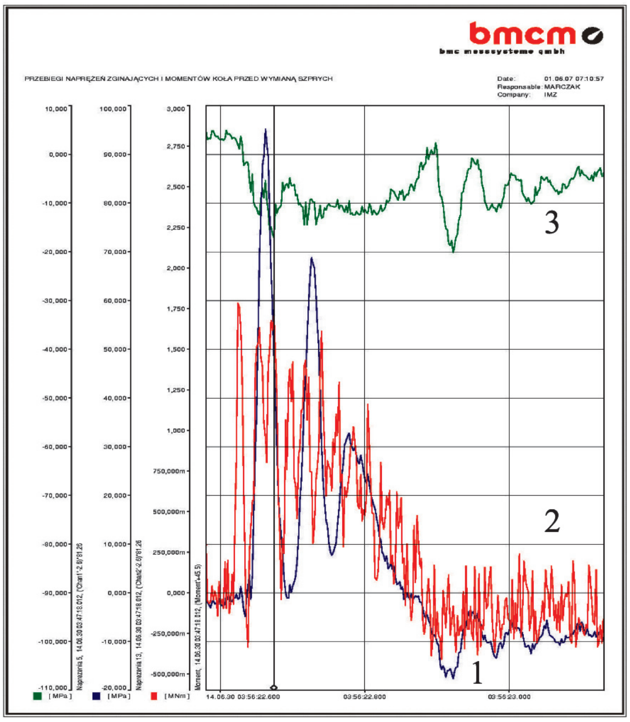


Fig. 3. Oscillograms of bending stress in the spokes 5 and 13 (1, 2) of the flywheel arms and torsion moment in the main shaft (3) of the pilger mill (IMŻ 2007)

A much better picture is obtained when we analyse how each individual flywheel arms should contribute to the overall load transmission (Fig. 4). This proportion is expressed as the quotient of effective bending moment of the arms in its most stressed cross-section to such value of the total moment of inertia forces applied to the wheel rim whilst its speed is varied (i.e. when accumulated energy is imparted) during the subsequent forging cycles.

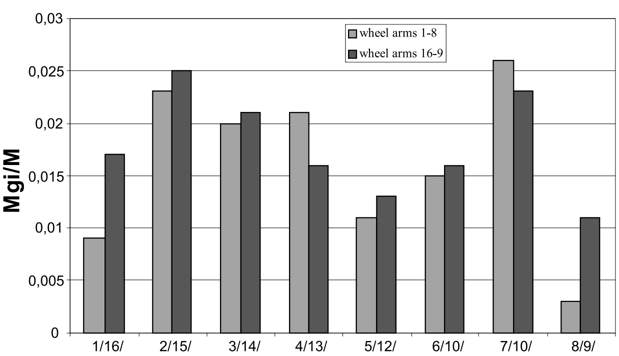


Fig. 4. Contributions of individual wheel arms to the transmission of inertia force moment. Condition after one year of service

It is readily apparent that after one- year service (see Fig. 4) of the rolling mill following its repair, the distribution of loads carried by individual flywheel arms are non-uniform. Stress amplitudes in spokes on the motor-end are lower on the average that in the neighbouring pair, located on the mill stand end. That implies that the flywheel system does not display the properties of a rigid solid, instead it behaves like a feeble frame, easily deformable under the applied loads. The resultant overloading of wheel arms on the mill stand end leads to their breaking, which further enhances the difference between the loads transmitted onto particular wheel sections.

In accordance with the procedures currently in force (Radkowski 2002), the system for monitoring the bearing condition in a roll mill (Fig. 2), both in the drive shaft (I step of system implementation) and the rollers in the stand (designed for step II) uses acceleration signals for vibration measurements. Piezoelectric vibration sensors fixed to the bearing housing enable the measurements in three directions in the rectangular coordinate system – horizontal H, vertical V, axial A.

Analogue signals from vibration sensors are converted into digital signals in equipment module boxes. That is why those module boxes are equipped with AD-DC converters and an additional control module enabling data transmission from all modules via the network interface Ethernet. The sampling frequency during measurements is 500 Hz.

The precision class of the measuring instruments is chosen as follows: 0.5–1.0 in vibration measurements and 0.2–0.5 in measurements of process variables. This selection implies a certain trade-off between reliability of diagnostic signals and costs of construction and expansion of the monitoring system.

In the most general case, the working condition of bearing is assessed in accordance with the technical standard PN-ISO-10816-1. As regards roll bearings, its recommendations cannot be directly implemented as these bearings are designed as large-sized slide bearings. Besides, the standard does not cover roll mill machines.

It has to be emphasised that early detection of damages in of key importance when diagnosing roll bearings. In this case the application of the effective amplitude of vibrations Vrms, recommended by the standard PN-ISO-10816-1 (Tab. 1) plays only a supportive role.

Among operating parameters registered during the roll mill's idle run and under loading during the forging duty cycle (Tab. 1), those found reliable and recommendable include the measurement data of vibrations during the mill's idle run.

That is so because, unlike measurements under loading, they are not disturbed by variable parameters of the forging process during the roll pass.

Bearing in mind that early detection of bearing damages is of key importance for good operation of the roll mill, the authors chose the peak factor (WSK), as in their previous publications:

$$WSK = \frac{Peak}{RMS} \tag{1}$$

WSK factor is expressed as the quotient of the peak signal value

$$Peak\ () = \max |x(t)| \tag{2}$$

and the root mean square *RMS*

$$RMS\ () = \left(\frac{1}{T} \int x^2\ (t)\ d\,t \right)^{\frac{1}{2}} \tag{3}$$

Table 1. Effective velocity amplitude Vrms [mm/s] of bearing vibrations in the drive shaft (HYDROMECH 2007), H, V, A – measurement in the horizontal, vertical and axial direction

Motor-end bearings			Mill stand- end bearings		
H	**V**	**A**	**H**	**V**	**A**
I. Results after the repair					
Idle run					
0.17	0.12	0.20	0.17	0.19	0.24
Forging					
1.40	0.60	1.20	1.20	1.30	2.70
II. Results after 3 months' service					
Idle run					
0.20	0.24	0.22	0.22	0.34	0.30
Forging					
1.60	0.80	1.60	1.80	1.80	3.00
III. Results after 6 months' service					
Idle run					
0.24	0.17	0.22	0.29	0.56	0.65
Forging					
2.50	2.50	3.30	3.10	2.60	6.00

This approach is justified by the fact that the WSK factor is most sensitive to part damage accompanied by impact force excitations, particularly when impact excitations produced during the pilger rolling process are co-periodic with rotations of a journal on which the bearing is mounted.

Many years' research work and expertise of other authors show that the occurrence and progress of damage of roll bearings will come relatively faster that in other types of machines and processes.

This phenomenon is closely associated with the nature of acting loads and the effects of thermal stresses.

The growing degradation of machine components leads to the change of frequency bandwidths in the generated signals, to the frequency modulation effect and finally, to the decay of characteristics frequency and spectrum randomisation. That is why diagnostic procedures are required that are more sensitive to stochastic disturbances of vibro-acoustic signals. It appears that the curtosis in selected frequency bands (variations of the concentration factor) fully answers the case (Radkowski and Zawisza 1996; Radkowski 2002).

$$K = \frac{\mu_4}{\sigma^4} \tag{4}$$

where:

$\mu_4 = \int_{-\infty}^{\infty} (x - \bar{x})^4 \, f(x)\, dx$ – central moment of a fourth-order stochastic process,

σ – standard deviation.

The full working order of a bearing corresponds to the normal distribution of amplitudes in particular frequency bands. As some elements get damaged, this distribution, and hence the curtosis, is liable to changes. At first those changes occur in low frequency bands only, shifting to the high-frequency section of the spectrum with the progressing degradation of machine components.

The changes of the probability distribution need not be associated with the fourth moment only, but they might also affect the third moment, being the measure of the skewness of the distribution.

$$\gamma_1 = \frac{\mu}{\sigma^3} \tag{5}$$

where:

μ – central moment of the third order,
σ – standard deviation.

It appears that in many cases the skewness factor (being the measure of asymmetry of the pattern) becomes a major indicator of the bearing damage.

Using curtosis and the skewness factor to the diagnostics of bearing condition in roll mill proves to be most useful as they enable early detection of the phase when manufacturing imperfections (wavy finish, spalling, local scattering of sizes of plastic deformations of rolled elements) are evenly distributed along the bearing track.

Applications of the curtosis to the assessment of the bearing condition in the working rolls of a pilger roll mill are illustrated in Figure 5.

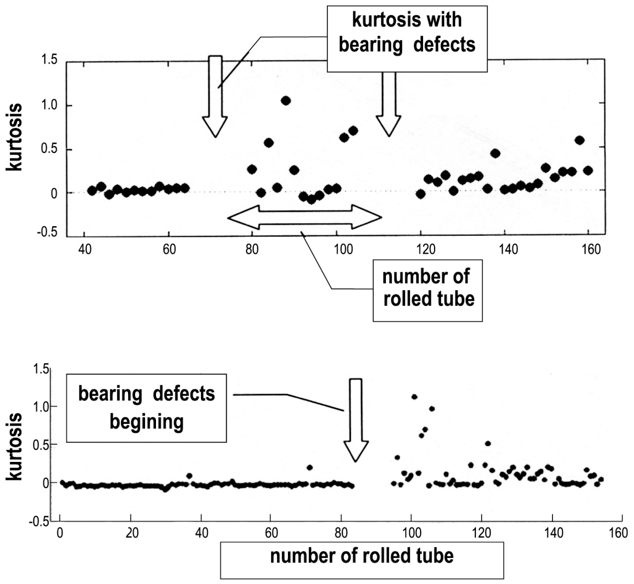


Fig. 5. Curtosis of a vibration signal from a bearing in a pilger mill when the bearing track gets damaged

Increase of the curtosis value is suggestive of the damage of the internal bearing ring in the working roll. That is most typical kind of damage, widely encountered in those roll mills.

Apart from vibration signals, the acquisition module in the operator's computer receives the process variable signals, which describe the fundamental process parameters, such as:

- current in the coil in the main drive motor,
- rpm of the rotor and working rolls,
- roll separating force.

Once the required expertise is gained, the monitor system shall be expanded to incorporate:

- stress measurements in shafts and heads of the roll driving spindle,
- stress measurements in the main coupling components,
- load measurements in the feeding apparatus.

3. MATHEMATICAL MODEL OF THE SYSTEM

Based on the literature on the subject and expertise of diagnosis specialists from Thysen-Krupp Stahl A.G, Voeast Alpine Stahl Linz GmbH, Bethehem Steel Corporation, it is generally assumed that detection and identification of processes that might adversely impact on the machine performance should involve the relationships between the process variables.

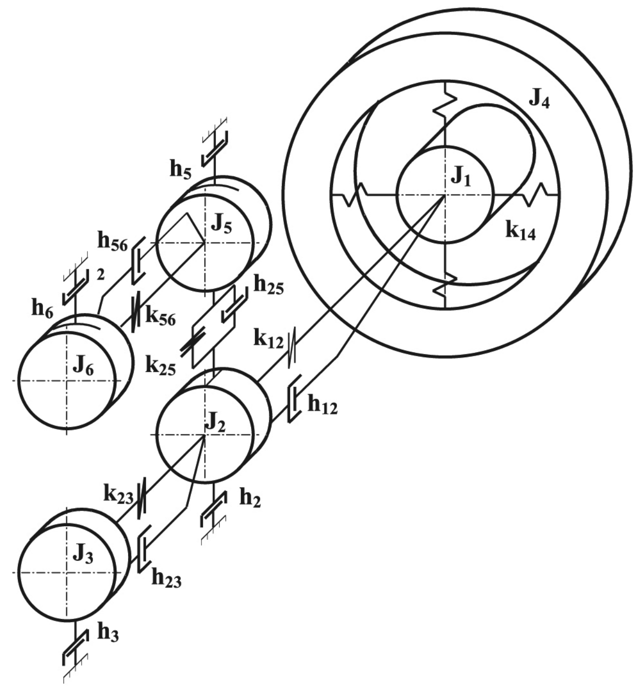
In order to generate the residua at the stage of detection, it is required that the mathematical model be formulated of the phenomena occurring within the investigated system. This model shall be supplemented by models of emergency conditions for particular roll mill components and by cause-and-effect relationships between defects and their symptoms.

A structural discrete model of a pilger roll mill, that would fully meet those objectives, is shown in Figure 6.

The equations of motion (Fig. 6) are written as:

$$\mathbf{J} * \Phi'' + \mathbf{H} * \Phi' + \mathbf{K} * \Phi = \mathbf{M} \tag{6}$$

where: $\mathbf{J} = \mathbf{Diag}[J_1, J_2, J_3, J_4, J_5, J_6]$



Model parameters	
reduced moment of inertia J_i	
J_1	reduced moment of inertia of the motor's rotor and the main coupling
J_4	reduced moment of inertia of the flywheel
J_2, J_5	reduced moment of inertia of the pinion stand rolls
J_3, J_6	reduced moment of inertia of working rolls
reduced elasticity k_{ij} and damping factors h_{ij}	
k_{12}, h_{12}	shaft and main coupling
k_{14}, h_{14}	wheel spokes
k_{25}, h_{25}	mashing of the gear rollers
k_{23}, k_{56}, h_{23}, h_{56}	connectors

Fig. 6. Mass spring model of a pilger mill's drive reduced to a 6-DOF system

$$\mathbf{\Phi}=\begin{bmatrix}\varphi_1\\ \varphi_2\\ \varphi_3\\ \varphi_4\\ \varphi_5\\ \varphi_6\end{bmatrix};\quad \mathbf{\Phi}'=\begin{bmatrix}\varphi_1'\\ \varphi_2'\\ \varphi_3'\\ \varphi_4'\\ \varphi_5'\\ \varphi_6'\end{bmatrix};\quad \mathbf{\Phi}''=\begin{bmatrix}\varphi_1''\\ \varphi_2''\\ \varphi_3''\\ \varphi_4''\\ \varphi_5''\\ \varphi_6''\end{bmatrix};\quad \mathbf{M}=\begin{bmatrix}M_{e1}\\ 0\\ M\\ 0\\ 0\\ M\end{bmatrix}$$

$$\mathbf{K}=\begin{bmatrix}k_{12}+k_{14} & -k_{12} & 0 & -k_{14} & 0 & 0\\ -k_{12} & k_{12}+k_{23}+k_{23} & -k_{23} & 0 & -k_{25} & 0\\ 0 & -k_{23} & k_{23} & 0 & 0 & 0\\ -k_{14} & 0 & 0 & k_{14} & 0 & 0\\ 0 & -k_{25} & 0 & 0 & k_{56}+k_{25} & -k_{56}\\ 0 & 0 & 0 & 0 & -k_{56} & k_{56}\end{bmatrix}$$

$$\mathbf{H}=\begin{bmatrix}h_{12} & -h_{12} & 0 & 0 & 0 & 0\\ -h_{12} & h_{12}+h_{23}+h_{23}+h_2 & -h_{23} & 0 & -h_{25} & 0\\ 0 & -h_{23} & h_{23}+h_3 & 0 & 0 & 0\\ 0 & 0 & 0 & 0 & 0 & 0\\ 0 & -h_{25} & 0 & 0 & h_{56}+h_{25}+h_5 & -h_{56}\\ 0 & 0 & 0 & 0 & -h_{56} & h_{56}+h_6\end{bmatrix}$$

With deformation of subsequent portions of the tube blank under the pressure of rolls (Fig. 1), the moment M applied to the roll mill is equal to:

$$M = M_w + M_t = F\psi\eta\sqrt{2\rho_x \Delta g_x} + 0.5 F_o \Delta g_x + M_t \qquad (7)$$

where:

- M_w – rolling moment,
- M_t – moment of friction in bearings,
- F – roll separating force,
- F_o – total axial force caused by the feeding apparatus and inertia forces,
- ψ – position coefficient of the resultant roll separating force (ψ = 0.82–0.78),
- η – coefficient of the contact surface contour (η = 0.96–1.13),
- ρ_x – roll diameter in the given cross-section in a rollgap,
- Δg_x – tube wall draft (variable along the rollgap length).

And varies in the function of tube displacement along the rollgap (Kazanecki 2003; Świątoniowski and Marczak 2007). In order to solve the equations of motion (6), the moment M is derived in the function of time, at the same time estimating the effects of sliding on the metal-roll surface.

4. CONCLUSIONS

Successful implementation of the system for monitoring of the roll mill condition and the pilger rolling process for producing seamless tubes is a major step towards improving the manufacturing quality, in accordance with the technical standards of the series ISO 9000.

Conditions are now ripe for collecting ongoing information about the rolling process. That provides new quality in relation to impromptu tests which, because of their specificity, left out such important factors as errors in calibrating the roll groove curvature or disturbances of the roll stand – feeding apparatus interactions.

However, experimental data gathered by various research teams under different conditions were difficult to correlate.

It is worthwhile to mention that the designs of drive systems in these types of machines make it extremely hard to effectively utilise the expertise gained whilst diagnosing other types of milling machines or test data collected in other rolling plants as they might be unique. Of particular importance is the currently updated database available at the rolling plant, containing the information about duly identified machine failures.

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