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ACOUSTIC EMISSION MONITORING OF LUBRICATION REGIMES IN ROLLING CONTACTS

BY

CARMEN BUJOREANU[*], MARCELIN BENCHEA and DUMITRU OLARU

“Gheorghe Asachi” Technical University of Iaşi, România,
Department of Mechanical Engineering, Mechatronics and Robotics

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Abstract. Proper lubrication is a major factor to prevent early damage of gears and bearings rolling contacts. The paper presents an investigation concerning the acoustic emission monitoring of the rolling contacts under various lubrication regimes. Experimental investigations were realized with a 51100 thrust ball bearing lubricated with different oil viscosities and operating at various rotational speeds. The results highlight that AE based parameters are sensitive to the contact lubrication conditions and this technique can be successfully use for the contacts lubrication monitoring.

Key words: acoustic emission; condition monitoring; lubrication regime; rolling contact.

1. Introduction

The rotating rolling contacts of the bearings produce mechanical vibrations and audible noises. The range of the phenomena causing vibrations in a rolling bearing and the resulting spectrum of vibrations are very wide.

This spectrum includes random frequencies (20-50 kHz), natural frequencies of bearing components (0.1-20 kHz), bearing rotational defect frequencies (1-1000 Hz) and different modulation frequencies (Berry, 1991).

[*]Corresponding author; *e-mail*: carmen.bujoreanu@gmail.com

High frequency vibrations (above 50 kHz) are mainly initiated from events inside the material on a micro geometrical scale.

Mechanical contacts between micro asperities in the surface profiles of the rolling elements, raceways and cages are one of causes for high-frequency vibration. Commonly, condition monitoring uses the vibration analysis but the signals picked by accelerometers are considerable affected by the machine noise. This is a reason to use acoustic emission (AE) as a suitable technique, acquiring signals at a much higher frequency (50 kHz to 1000 kHz) and therefore avoiding the noise low frequencies.

Acoustic Emission (AE) is defined as a physical phenomenon occurring inside and/or on the surface of materials whereby spontaneous elastic energy is released in the form of transient elastic waves which cover a broad frequency range, typically from 20 kHz to 1 MHz (outside the range of human hearing) (Hamel *et al*., 2014; Mba & Rao, 2006). The interaction of the moving surfaces generates stress waves in the bearing material. The stress waves radiate as spherical wavefronts in the material and cause high frequency waves on the material surface. This means an acoustic emission and its parameters are obviously linked to the interaction degree of the contacting surfaces.

In most applications, bearings troubleshoots are assigned to lubrication problems and consequently, a proper lubrication management regime will help to an early prevention of failures and bearing damage. The operating life of bearings is determined by the ratio between the oil film thickness and the combined surface roughness of the contacting parts, this well-known lambda ratio λ being vital for maintaining the system mechanical integrity. According to EHL theory, the ratio λ changes with change in temperature, load, surface roughness and speed, consequently the contact area between the bearings surfaces elements is also changing. This can lead to severe adhesive wear (even scuffing) and the bearing malfunction. The AE signals offer an important advantage for bearing condition monitoring in fault detection at an incipient stage of their development. The paper presents an investigation concerning the acoustic emission monitoring of the rolling contacts under various lubrication regimes. Experimental investigations were realized with a 51100 thrust ball bearing lubricated with different oil viscosities and operating at various rotational speeds. The results highlight that AE based parameters are sensitive to the contact lubrication conditions and this technique can be successfully use for the contacts lubrication monitoring.

2. Theoretical Background

Research has been developing in order to establish some relationship between AE parameters and failure events, via dominant wear mechanisms or the lubrication regimes from the rolling/sliding contacts.

Boness and McBride (1991) found that wear material removal and the integrated RMS signal are related for the dry and lubricated contacts.

Zhang *et al.* (2006) measured with an ultrasonic technique the thickness of lubricating films for the rolling element contacts from 6016 ball bearing type. The obtained results were in good agreement with EHL theory in the 0.3–1.0 μm interval for radial loads above 2.5 kN and the shaft speeds below 200 rpm (3.3 Hz). Some ultrasonic measurements limitations include the pulse repetition rate which limits the number of measurement points and the scattering of the ultrasonic beam over a small area resulting an average value over that area. However it was clearly demonstrated that ultrasonic measurement has the potential for monitoring the thickness of lubricant films in industrial applications.

Ben Abdallah and Aguilar (2008) used a ball-on-cylinder lubrication test machine with a fixed ball sliding over a flat ring attached to a rotating cylinder. They investigated the dependence between the emitted sound and wear properties. The cylinder speed, namely the ball sliding speed of the ball, was the interest parameter and it was found that a sliding dry contact generated a continuous AE waveform. It was concluded that the RMS voltage of the AE signal produced by the ball-ring sliding had a relation with the friction coefficient.

Tan and Mba (2005) studied for isothermal conditions the RMS level of the AE signal generated in rolling contacts and they emphasized a minimal effect of the imposed loads but a significant effect on the speed of rotation. They confirmed that the RMS value of the AE signal was also dependent on the film thickness and the relative slide to roll ratio of the contact strongly affected the overall level of the AE signal.

Raja and Mba (2007) extended the previous work into the generation of AE as a function of λ and they are using helical and spur gears in an oil bath lubricated gearbox. It was found that the value of λ clearly and directly influenced the level of the AE signal for both types of gears.

2.1. AE Parameters

The AE technique supposes first the detection and then the conversion of high frequency vibrations generated from a source to electrical signals. For this purpose, a high frequency piezoelectric transducer is used and it can be mounted directly to a test sample. Most of AE sensors have a sensing element in the form of a thin disc of piezoelectric material; normally lead zirconate titanate (PZT) that can convert mechanical deformation into electrical voltage (Niknam *et al*., 2013). It has an excellent sensitivity and relative immunity to a wide range of industrial applications (Niknam *et al*., 2013).

However, AE signals are characterised by a rapid fall so the sensors have to be placed in the nearness of the AE source to avoid the attenuation problems.

The output of the AE sensor is usually amplified, passed through filters to remove noise and sent to a data processing unit. In general, AE signals can be considered of two types, namely, continuous signals and burst signals. Continuous signals are associated with dislocation movement through the crystal lattice and friction between the contacting surfaces. Burst signals are short duration pulses of high amplitude representing the energy released during crack initiation and growth.

There are some parameters of AE signals which can be analyzed including absolute energy, root mean square (RMS), amplitude, counts and average frequency (Warren and Guo, 2007).

Absolute energy is a real energy measure of the AE signal. Its value is determined from the integral of the squared voltage signal divided by the reference resistance (10 kΩ) over the duration of a waveform packet. As a hit feature, it reports the true energy contained in a detected acoustic emission burst signal *RMS* is a measure of the continuously varying and "averaged" amplitude X of the AE signal. It is proportional to the vibration power during a period T and it is defined as the time averaged AE signal. It is measured on a linear scale and reported in volts.

$$x_{ef} = x_{RMS} = \sqrt{\frac{1}{T}\int_{o}^{T} x^2(t)dt} \tag{1}$$

Amplitude is the maximum positive or negative AE signal rise during an AE hit. The amplitude is expressed in volts. It can be evaluated in decibels (dB) using the relationship:

$$dB = 20 \cdot \log(X_{max} / 1\mu V) \tag{2}$$

Counts are an AE hit feature that measures the number of signal rises over an AE threshold.

Average frequency is a calculated feature, reported in kHz, which determines an average frequency over the entire AE hit. It is derived from the other collected AE features, namely AE counts and duration. It is a real time calculation determined as the ratio between AE counts and the corresponding duration.

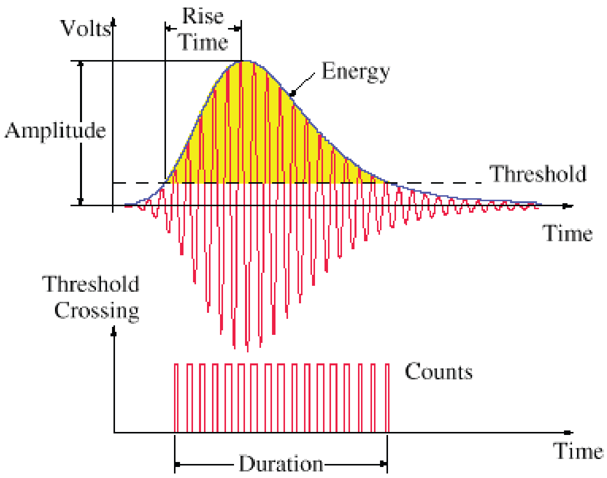


Fig. 1 − The main parameters of an acoustic emission hit (Huang *et al*., 1998).

Fig. 1 presents the main parameters of an acoustic emission hit (Huang *et al*., 1998). The AE transducer notifies a signal over a certain level (*i.e*., the threshold), and the AE event is acquired. The event amplitude is defined at the peak of the signal. The count of the AE event is considered the number of the signal rises and crosses the threshold. The duration of the AE event is the period between the rising edge of the first count and the falling edge of the last count. The period between the rising edge of the first count and the peak of the AE event is called the rise time. The energy is practically the area under the envelope of the AE event.

2.2. Oil Film Thickness Computation

The correlation between some AE based parameters and the lubrication regimes in the rolling contacts of a ball bearing also involves the oil film calculation.

In order to determine the film thickness in ball–race contacts, it is important to establish if the lubrication regime is IVR or EHD, depending on the oil viscosity and rotational speed. It has been used Hamrock's methodology (Hamrock, 1994) and maps of lubrication regimes for all the testing conditions have therefore been established (Bălan *et al*., 2014). So, for all lubricants and rotational speeds tested, the dimensionless viscosity parameter g_v and the dimensionless elasticity parameter g_E as described by Hamrock (1994) are calculated:

$$g_V = \frac{G \cdot W^3}{U^2},\ g_E = \frac{W^{8/3}}{U^2} \tag{3}$$

U is the dimensionless speed parameter; W − the dimensionless load parameter; G − the dimensionless material parameter.

The lubricant film thickness in the ball–race contacts can be determined as follows:

In the *IVR lubrication regime*, Brewe's relationship found in (Houpert, 1987) is used:

$$h_{IVR} = R_x \cdot \left[\left(1 + \frac{2}{3 \cdot k}\right) \cdot \frac{W}{U} \cdot \left(0.131 \cdot \arctan\left(\frac{k}{2}\right) + \frac{1.683}{1}\right) \cdot (128 \cdot k)^{0.5} + 2.6511 \right]^{-2} \tag{4}$$

In the *EHL lubrication regime*, Hamrock–Dowson's equation (Hamrock, 1994) is used:

$$h_{EHL} = 2.69 \cdot R_x \cdot U^{0.67} \cdot G^{0.53} \cdot W^{-0.067} \cdot \left(1 - 0.61 \cdot e^{-0.752 \cdot k^{0.64}}\right),\ k = \frac{R_x}{R_y} \tag{5}$$

R_y is the equivalent radius of contact curvature in the direction perpendicular to the rolling path and R_x − the equivalent radius in the rolling direction.

The transition IVR to EHL lubrication regime is considered the maximum value of oil film thickness of the two lubrications regime as above mentioned:

$$h_{IVR\_EHL} = \max(h_{IVR}, h_{EHL}) \tag{6}$$

The lubrication parameter λ is defined by the relation:

$$\lambda = \frac{h}{\sqrt{R_{qr}^2 + R_{qb}^2}} \tag{7}$$

where $\sqrt{R_{qr}^2 + R_{qb}^2}$ represents the composite roughness of the race-ball contact.

3. Experimental Procedure

Experiments were carried out on a fully-instrumented universal micro-tribometer at room temperature (Fig. 2) using an AE sensor with a frequency response from 0.2 to 5.5 MHz and a bandwidth of ± 3 dB with a gain of 60 dB.

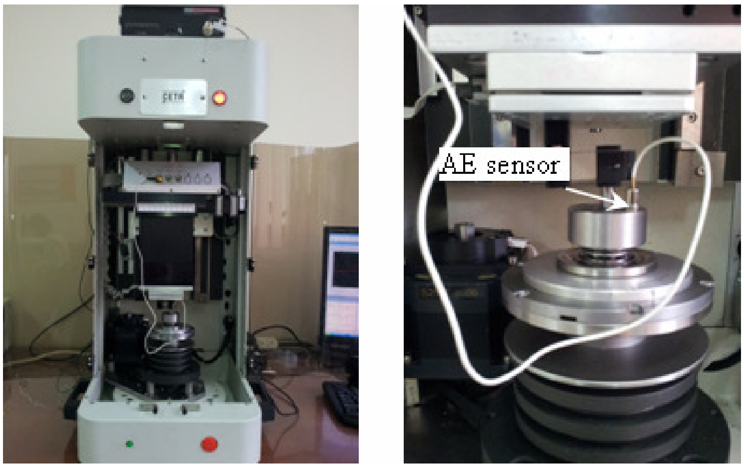


Fig. 2 − Experimental setup.

The upper specimen is connected to a vertical linear motion system that has a travel length of 150 mm. The measurements can be performed by the instrument to an accuracy of 50 nm. A precision spindle can rotate the lower specimen at speeds from 0.001 rpm up to 5000 rpm. Accurate strain gauge sensors perform simultaneous measurements of load and torque in two to six axes. The forces can be precisely measured with a resolution of 0.00003% of the full-scale and very high repeatability.

The rolling contacts behaviour, in terms of acoustic emission, is analyzed for the 51100 thrust ball bearing and the operating conditions are monitoring. This means an experimental investigation concerning the AE parameters related to the operating conditions through dry, mixed and EHL lubrication regimes.

The tested bearing has a rolling path with the radius of 8.4 mm, the transversal curvature radius of 2.63 mm, the ball diameter of 4.76 mm. The Talysurf based measured roughness for rolling path and ball is 0.03 μm and 0.02 μm respectively. The tests are realized using a normal load on the bearing of 10 N and for the following rotational speed of the bearing: 50 rpm, 80 rpm, 110 rpm, 140 rpm, 170 rpm, 200 rpm, 230 rpm, 260 rpm and 290 rpm.

The experiments are conducted for two lubricants covering both the mixed and full film lubrication regimes. The full lubrication film ($\lambda > 3$) is obtained by the eqs. (3)-(6), using a lubricant with dynamic viscosity of 0.35 Pa·s, whilst the use of another oil with 0.05 Pa·s dynamic viscosity leads to mixed lubrication regimes ($\lambda < 3$). The variation of the lubrication parameter related to the bearing rotational speed for the oils used in the experiments is shown in Fig. 3.

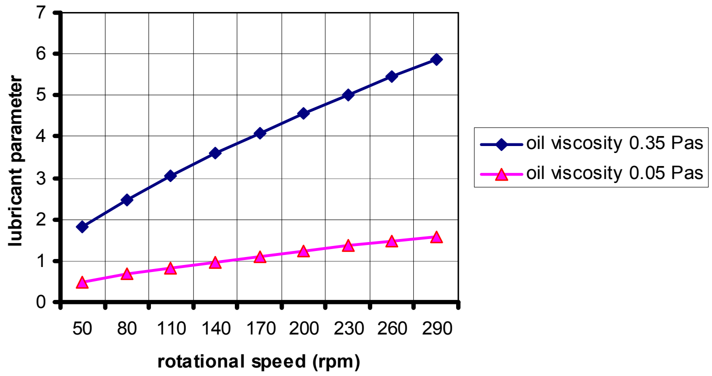
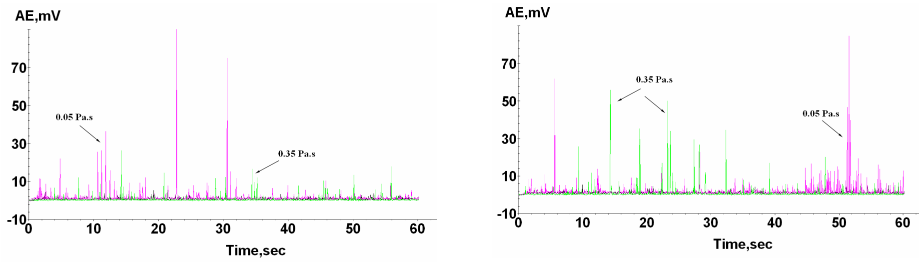


Fig. 3 – Lubricant parameter *versus* bearing rotational speed.

4. Results and Discussions

Some examples of the amplitude level history of AE waveforms acquired from the tested bearing are presented in Fig. 4 for two different rotational speed of the bearing (320 rpm, 260 rpm) and two different oil viscosities (0.05 Pa·s; 0.35 Pa·s).

It is obvious the presence of more peaks of AE signal for the situation when the bearing lubrication is realized with the 0.05 Pa·s oil viscosity compared to that of 0.35 Pa·s oil viscosity lubrication. At the same, the increase of the rotational speed of the bearing (from 260 rpm to 320 rpm, as in Fig. 4) leads to the increase of the amplitude of the AE signal, evident for both oil viscosities.



a *b*

Fig. 4 – Amplitude level history of AE for oil viscosities of 0.05 Pa·s and 0.35 Pa·s: *a*) 260 rpm; *b*) 320 rpm.

Fig. 5 presents the raw waveform captured from the tested bearing operating at 320 rpm without lubrication (dry regime).Under dry conditions it can be seen transient peaks in the AE signal which are superimposed into the continuous emission. At the first view, it can be noticed that the peaks in the dry regime are smaller than those observed in lubricated conditions, what seems difficult to explain since the asperities contacts are considered a source of noise and vibrations and there are no oil film between them. But these are the raw waveforms and the results may not be so convincing.

This is a reason to process the signal and to use the AE parameters in order to have a proper perspective on the link between acoustic emission and lubrication.

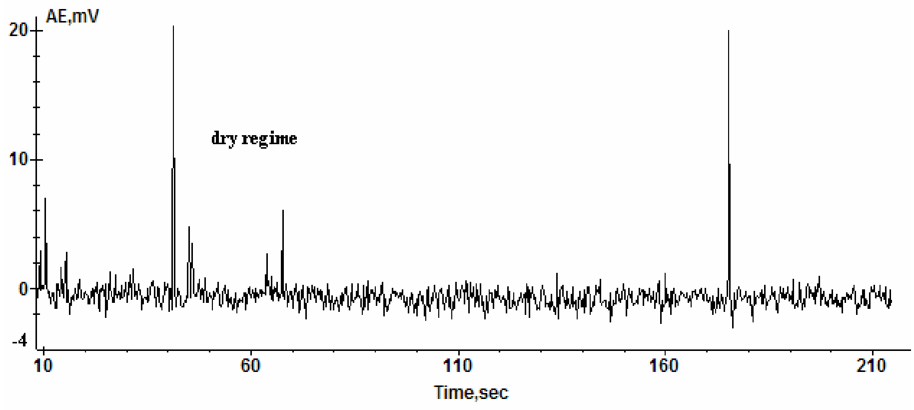


Fig. 5 − Amplitude level history of AE signal for dry regime at 320 rpm bearing speed.

Fig. 6 presents the AE - RMS level obtained by averaging the amplitudes of the signals (eq. (1)) captured from the tested bearing for the above mentioned lubrication conditions.

The AE - RMS increases with the increase of the bearing rotational speed and decreases with the oil viscosity increasing, as it is shown in Fig. 6. In the dry regime, as is expected, the AE signal is greater than in the lubricated regimes as the result of the direct contact between the bearing rolling surfaces. The higher is the bearing rotational speed the higher is its AE level and more distinctive for the two different oil viscosities used in our tests.

Fig. 7 shows the change in AE RMS level (expressed in dB, according to eq. (2)) in a relation with the bearing rotational speed. There is an AE - RMS increasing for the lower oil viscosity of 0.05 Pa·s from 50.6 dB to 64 dB. For the other oil with 0.35 Pa·s viscosity it was recorded an increasing of AE - RMS level from 47.7 dB to 50.6 dB in the same time duration. The higher viscosity oil expresses a tendency to better attenuate the AE amplitude level and to stabilize the noise level even if the bearing speed further increases.

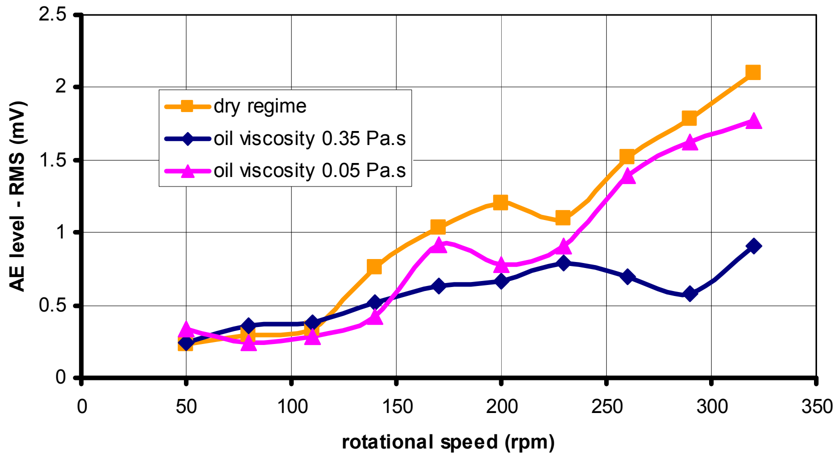


Fig. 6 − AE - RMS level (in milivolts) *versus* bearing rotational speed.

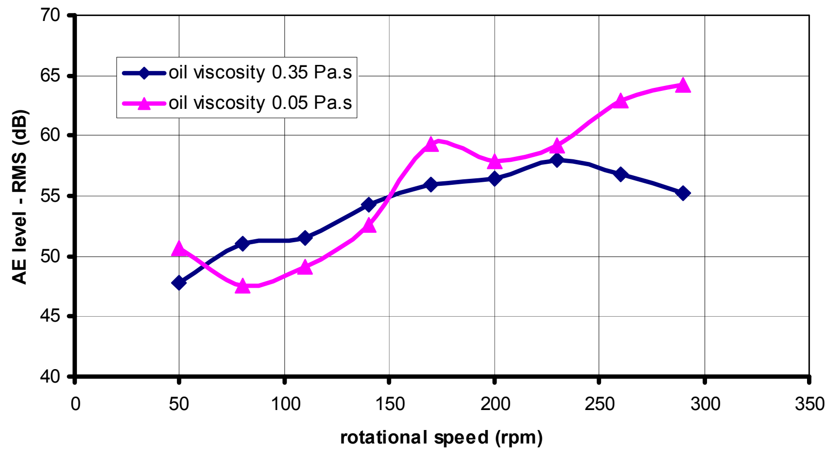


Fig. 7 − AE - RMS level (in decibels) versus bearing rotational speed.

Therefore there is a change in AE - RMS as a function of bearing rotational speed and the lubrication regimes evidenced through the lubricant parameter values.

5. Conclusions and Future Work

The paper presents an investigation concerning the relation between the acoustic emission technique and the lubrication regimes in the rolling contacts. The tests are realized on 51100 thrust ball bearing lubricated with different oil viscosities and operating at various rotational speeds.

The results suggest the possible sources of acoustic emission such as asperities contacts, secondary pressure peaks in lubricated contacts, balls – cage random collisions.

The AE level has an evident change due to the variation in lubrication conditions through dry and lubricated regimes. The generation of AE amplitude in the lubricated regimes is relatively lower than under dry conditions and the oil viscosity increasing leads to AE amplitude level decreasing.

The results highlight that an AE based parameter (RMS in our investigation) is sensitive to the contact lubrication conditions and this technique can be successfully use for the contacts lubrication monitoring.

For the future applications, the AE captured data can be postprocessed to perform more complex analysis and diagnosis. It can be added statistical values, including Kurtosis, Crest Factor (ratio of maximum to RMS levels) or auto-correlation techniques and wavelet analysis to identify periodicities and nonlinearities in rolling contacts.

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MONITORIZAREA EMISIEI ACUSTICE A REGIMURILOR DE UNGERE ÎN CONTACTELE DE ROSTOGOLIRE

(Rezumat)

Ungerea corectă reprezintă un factor important în prevenirea uzurii premature a rulmenţilor şi roţilor dinţate. În această lucrare se prezintă o analiză a semnalului de emisie acustică produs în contactele de rostogolire pentru diferite regimuri de ungere. Testele experimentale au fost realizate pe un rulment axial cu bile tip 51100, lubrifiat cu uleiuri cu vâscozităţi diferite, funcţionând la diverse turaţii de rotaţie. rezultatele experimentale au scos în evidenţă faptul că parametrii principali ai semnalului de emisie acustică variază în funcţie de regimul de ungere, iar tehnica de monitorizare a emisiei acustice poate fi aplicată cu succes în cazul contactelor lubrificate la rulmenţi şi roţi dinţate.