Diagnostics features of technical condition for inking unit drive of printing machines

G B Kulikov¹, A I Vinokur¹, V P Petrov¹, L G Varepo², P S Belyaev³

¹ Moscow Polytechnic University, 38, B. Semenovskya str., Moscow, 107023, Russia
² Omsk State Technical University, 11, Mira av., Omsk 644050, Russia
³ Tambov State Technical University, 106, Sovetskaya Str., Tambov, 392000, Russia

E-mail: kulikov.gb@mail.ru

Abstract. The paper considers approaches to assessing the technical condition of the inking unit drive of printing machinery in the work environment. As the main diagnostic feature, it is proposed to use the noise spectrum, created when working with inking systems. Reasons for noise of inking systems are considered. To create a mathematical model of sound radiation by quill cylinders of inking system, it is shown that behaviour is described by oscillation equation of tubular girder. Theoretical research has shown, there is a possibility in principle to use the noise spectrum emitted by the inking system to diagnose the technical condition of its drive. To improve the accuracy of diagnosis, it is proposed to use neural network technology of image identification.

1. Introduction
Stable and continuity of service of inking system depends on condition of drive, as well as on quality of fault and timely maintenance. Various types of defects can be identified by analyzing the noise and vibration levels created when the inking system is working.

In numerous studies on the sources of noise and vibrations of printing equipment [1], it is shown that inking systems (IS) are the most powerful sources of noise in printing units. And if sufficiently effective solutions [1] were found to reduce the noise emitted by these objects, then at present the reliable service of the inking system comes to the forefront. The reliability of the inking system depends primarily on the state of the drive, in particular on bearing systems. In this case, the rolling bearings of inking system shafts, in contrast to the supports of the printing pair, are usually not equipped with pressure-feed oil system, which adversely affects their resource.

2. Problem statement
It is common knowledge that almost all types of defects that determine the resource of mechanical equipment change the parameters of noise and vibration that accompany the operation of mechanisms. In view of this, vibroacoustic diagnostic features that are best suited for detecting defects in mechanical engineering assemblies [2, 3]. Thus, it becomes an relevant objective to identify components in noise and vibrations created by the IS, that can give objective information about drive problems. In this case, from a practical point of view, use of a noise signal would be preferable, since it does not require the installation of vibration detectors on the machine.
Studies, conducted earlier [1] have shown that the frequency components corresponding to the tuned frequencies of grinding IS (cylinders) are clearly distinguished in noise spectrum emitted by the printing section. The reason for this is that IS shafts are powerful resonators with a high degree of \( x/r \) ratio (the structural loss coefficient is equal to \( \eta = 0.0009 \div 0.0015 \) [1]), and a large emitting surface creates noise levels much higher than mechanisms that excite them.

One might assume that excitation source of IS shafts are drive, in particular rolling element bearings. Thus, it is necessary to find out, how the vibration of the rolling bearings affects the noise emitted by IS. Any IS consists of a set of hollow cylinders, which are a shell with permanently attached trunnions and gear systems. Analysis of printing equipment designs has shown [1] that cylinders with a diameter of 100-130 mm, with a wall thickness of 6-8 mm and a length of 800-1200 mm are most common in IS. Therefore, cylinders used in IS are fairly rigid cylindrical shells.

The main task of the study is to investigate the possibility of diagnosing the technical condition of bearings of the printing machine’s ink apparatus from the emitted noise using vibration measured on the frame as an additional source of information.

3. Results and discussion

Vibrations of cylindrical shells were studied by the classics of acoustics Reley and T. Lamb, then this topic was studied by E. Skushik, V. Breslavsky, D. Plakhov and many others. Of modern researchers, studies of HongSong, FuchunYang should be noted [4, 5]. Problems of determining acoustic fields and vibration fields of cylindrical shells are of great importance in aviation, astronautics and engineering, so they are well developed.

In the deriving equation of a cylindrical shell, the differential equation of plate vibrations is usually taken as a basis, in which additional terms are introduced that take into account the peculiarities of shell vibrations. The reason for this is the fact that at high frequencies the radius of curvature becomes many times smaller than the length of the bending wave and cylindrical shell must approach the plate in its behavior [6, 7]. On the other hand, at low-frequency, at which the length of the bending wave becomes greater than the circumference of the cylinder cross-section, the shell makes beam-type vibrations, which can be described using the theory of bending waves.

In general, three characteristic frequency ranges can be distinguished for shells:

- Frequencies from 0 to \( f_{ct1} \), at which the condition is met \( \lambda_b > 2\pi R \), where \( \lambda_b \) is the length of bending wave in the shell, \( R \) is mid-radius of the shell. In this range, the shell behaves like a cylindrical bar.
- Frequency range from \( f_{ct1} \) to \( f_{ct2} \), in which \( \lambda_b < 2\pi R \), \( \lambda_l > 2\pi R \), where \( \lambda_l \) is the length of dilatational wave in shell. This is the area of shell vibrations in which the radius of curvature becomes commensurate with the length of bending wave and the length of the dilatational wave is greater than the circumference of cross-section of shell.
- And finally, the third frequency range from \( f_{ct2} \) to \( \infty \), when \( \lambda_l < 2\pi R \). This is the frequency range above the first radial resonance \( (f_{ct2} = \frac{C_l}{2\pi R} \text{ where } C_l - \text{the speed of dilatational waves in the shell}) \). In this range, the shell behaves like a plate.

Thus, we need to find out which of the above ranges the IS cylinders fall into. Knowing the typical dimensions of IS hollow cylinders, we will determine the required boundary frequencies.

It is a common fact that phase rate of propagation of bending waves along thin-walled cylinders is equal to:

\[
C_b = \frac{\omega}{K_b}
\]

where \( K_b \) is wave number of bending waves that depends on frequency and is equal to:
\[ K_b = \sqrt{\frac{\rho_0 \omega^2}{Er^2}} = \sqrt{\frac{\omega}{C_l r}} \]

It follows that

\[ C_b = \sqrt{C_l r \omega} \] (2)

In the following expressions:
- \( \rho_0 \) — density of the cylinder material;
- \( E \) — modulus;
- \( C_l \) — velocity of dilatational waves in shell;
- \( r \) — section radius of cylinder;
- \( \omega \) - circular frequency.

Equating the length of bending wave to the circumference of cross-section of cylinder, we determine the first boundary frequency:

\[ f_{ct1} = \frac{2C_l r}{\pi d_{mn}} \] (3)

where

\[ d_{mn} = \frac{d_1 + d_2}{2} \]

where \( d_1 \) and \( d_2 \) are the outer and inner diameters of cylinders, respectively.

After simple calculations, we get that hollow cylinders with a diameter of up to 300 mm, and with a wall thickness of 2-12 mm in the frequency range from 0 to 5000 Hz behave like tubular girders. Thus, almost the entire range of IS cylinders does not go beyond the first frequency range and their behavior can be described by the differential equation of oscillations of a rod of arbitrary cross-section [1], in partial derivatives, it is written in the following form:

\[ m \frac{\partial^2 y}{\partial t^2} + B \frac{\partial^4 y}{\partial x^4} = 0, \] (4)

where \( y \) — transverse displacement of the rod;
- \( x \) — axial coordinate;
- \( m \) — mass per unit length of the rod;
- \( B \) — bending stiffness.

This equation, corresponding to the harmonic bending wave, is solved as follows:

\[ y(x, t) = y(x) \sin \omega t \] (5)

Substituting (5) in (4) we get:

\[ \frac{\partial^4 y}{\partial x^4} - K_b^4 y(x) = 0 \] (6)

where \( K_b \) is a wave number of bending waves:

\[ K_b = \frac{m \omega^2}{B} = \frac{\omega}{C_b} \] (7)

where \( C_b \) is the phase velocity of bending wave, which characterizes the distribution of transverse displacement phase along the rod when the bending wave propagates along it (see formula 2).

The solution of the rod motion equation can also be represented either as a trigonometric sequence Fourier or as Krylov functions.
Value of kinetic and potential energy of bending wave per unit length of the rod is equal to:

\[ W_{\text{kin}} = \frac{m}{2 \lambda_b} \int_0^{\lambda_b} \left( \frac{\partial y}{\partial t} \right)^2 dx = \frac{m \omega^2 y_0^2}{4} = \frac{B K_b^4 \pi}{2 \lambda_b} y_0^2 = \frac{B K_b^4 \pi}{4} y_0^2 \]  

(8)

\[ W_{\text{pot}} = \frac{E J}{2 \lambda_b} \int_0^{\lambda_b} \left( \frac{\partial^2 y}{\partial x^2} \right)^2 dx = \frac{B K^2 \pi}{2 \lambda_b} y_0^2 = \frac{B K^2 \pi}{4} y_0^2 \]  

(9)

where the transverse displacement \( y \) is defined by the expression:

\[ y = y_0 \cos (\omega t - K_b x) \]  

(10)

Total energy per unit length of rod or energy density of bending wave, taking into account the expressions (8) and (9) is equal to:

\[ W = \frac{B K^4}{2} y_0^2 \]  

(11)

or with expression (7)

\[ W = \frac{1}{2} m \omega^2 y_0^2 = \frac{1}{2} m y_0^2 \]  

(12)

If losses in the material are taken into account in expression (4), then the equation of motion for bending waves will be (for \( \eta << 1 \) [8]):

\[ \frac{\partial^2 y}{\partial t^2} + \frac{E J}{m} (1 + j \eta) \frac{\partial^4 y}{\partial x^4} = 0 \]  

(13)

where \( \eta \) is the coefficient of losses of rod material.

Solution for vibrations of a bounded rod can be written as follows:

\[ y(x, y) = [A \cos K_b x + B \sin K_b x + C \cosh K_b x + D \sinh K_b x] e^{-\delta t} e^{j \omega t} \]  

(14)

where \( \delta \) is the time index of flexural wave attenuation.

Next, we define the temporal and spatial attenuation ratio of bending waves. Substituting the expression (14) into equation (13), it is easy to obtain the time decrement of bending waves at \( \eta << 1 \):

\[ \Delta t = \frac{\delta}{f} = \frac{\eta \omega}{2 f} = \pi \eta \]  

(15)

Spatial attenuation ratio of bending waves

\[ \Delta_s \beta \lambda_b = \frac{K_b \eta \lambda_b}{4} = \frac{\pi \eta}{2} \]  

(16)

From a comparison of expressions (15) and (16), it can be seen that the spatial attenuation ratio of bending waves is two times less than the time one. This is due to dispersivity of bending wave velocity. Based on (16), it is possible to determine the attenuation of bending wave amplitude per unit length of the cylinder:

\[ \Delta_s = 20 \lambda \delta e^{-\beta \Delta x} = \frac{13.8 \eta}{\lambda_b} \]  

(17)

For IS cylinders, this value does not exceed the value \( \Delta_s = 0.1 \) dB/m, that is for the entire range of cylinders under study, vibration level practically does not change over the entire
surface, regardless of point impedance. In the presence of decrement, value $\Delta_s$ can take values $\Delta_s = 1 \div 2 \, dB/m$, that also slightly affect the pattern of vibration distribution on cylinder surface. In view of the foregoing, it follows that almost all the energy coming to IS shafts from the drive is radiated by them to the surrounding space at their own frequencies uniformly over the entire surface of cylinder. This indicates that we can install the microphone in immediate proximity to IS cylinders in any convenient place for measurements.

Hollow IS cylinders are usually installed in rolling bearings, so their natural frequencies can be calculated in the same way as for beams with fixed endpoints. In methodology [1], for calculating the natural frequencies of hollow cylinders of printing machines is given. You can use a dependency to do this:

$$f_i = \frac{i^2 \pi}{2l^2} \sqrt{\frac{EJ}{m}}$$

where $i = 1, 2, 3, \ldots$;

- $l$ — length of forming cylinder;
- $E$ — modulus of elasticity of cylinder material;
- $J$ — moment of inertia about neutral axis;
- $m$ — mass per unit length (the weight of trunnions must be entered in bulk weight, that is the mass of trunnions is evenly distributed in mass of shell).

The biggest contribution to radiation of hollow cylinder is given by the first four, five modes of bending vibrations.

When considering rolling bearings as a source of vibration that excites the IS cylinders, it is necessary to consider their possible defects, and how defects affect the vibration activity of the bearing.

During operation, increased vibration levels of bearings can cause the following types of defects [2,9,10]:

- fatigue failure of material;
- damage caused by increased wear;
- damage due to insufficient or stoppage of oil supply;
- beating of rings relative to each other due to a violation of the geometry of surfaces.

Fatigue failure of rolling bearings is manifested in the form of discoloration of the material of ring tracks and ball. Excessive loads and exceeding the permissible rotation speed can cause fatigue failures.

Increased wear of bearing parts, especially rolling elements, and ring surfaces, leads to increased radial clearances that cause runout and increased shaft vibration.

Reason for wear of separator sockets may be incorrect mounting of bearings, large axial loads, painted raceways and a number of other reasons.

As evidenced in practice, the main reason for the destruction of IS bearings is a violation of the lubrication conditions, which leads to guttering and wear of rolling bodies, pilling-up on the surface of tread face, wear of separator, and an increase in the temperature of the unit.

When calculating the main frequencies of rolling bearing perturbations, the hypothesis of the precedence of shock excitatory forces is used for this purpose, well-known dependencies can be used. Numerical values of frequencies of these components depend on the ratio of geometric dimensions of bearing and rotor speed mechanism:

- shaft speed (inner ring)

$$f_r = \frac{n}{60}$$

(19)
frequency of rotation of the separator:

\[ f_{sep} = \frac{f_r}{2} \left( 1 - \frac{d}{D} \cos \beta \right) \]  

(20)

frequency of rotation of rolling elements:

\[ f_{r.e.} = \frac{f_r D}{2d} \left[ 1 - \left( \frac{d}{D} \right)^2 \cos^2 \beta \right] \]  

(21)

frequency of flashing of rolling elements on the outer ring:

\[ f_{out} = \frac{z f_r}{2} \left[ 1 - \frac{d}{D} \cos \beta \right] \]  

(22)

frequency of flashing of rolling elements on the inner ring:

\[ f_{in} = \frac{z f_r}{2} \left[ 1 + \frac{d}{D} \cos \beta \right] \]  

(23)

In the above formulas:

- \( n \) — rotor speed, min\(^{-1}\);
- \( d \) — diameter of rolling bodies;
- \( D = (D_{in} + D_{out})/2 \) — diameter of a circle passing through centers of rolling bodies;
- \( D_{in} \) and \( D_{out} \) — respectively, diameters of inner and outer rings of bearing;
- \( \beta \) — contact angle in degrees;
- \( z \) — number of rolling elements.

At the same time as during operation, when damage occurs and develops to bearing structural elements, discrete components appear in the vibrational spectrum of operating mechanism, frequencies of which are determined by the type of damage. Defects are quite diverse: undulation on raceways, increased clearance in cage, ovality of rolling bodies and rings, cutting, increased clearance in separator slots, damage to bodies and raceways due to painting. Main excitation frequencies that occur due to damage to rolling bearing elements are as follows:

Due to a defect in shape of rolling elements

\[ f_1 = \left( \frac{D + d}{d} \right) \left( \frac{D - d}{d} \right) \frac{n}{30} \]  

(24)

Due to a change in the shape of the internal raceway

\[ f_2 = \left( \frac{D + d}{D} \right) \frac{nz}{120} \]  

(25)

Due to a change in shape of external raceway

\[ f_3 = \left( \frac{D - d}{D} \right) \frac{nz}{120} \]  

(26)

Resonance vibrations of bearing elements can occur as a result of the periodic impacts of rolling elements when passing defective points of the raceway. These vibrations are characterized by attenuated high-speed pulses. One of the resonances is the resonance of balls, frequency of which is calculated using the formula [11]:

\[ f_b = \frac{0.848 \, E}{d \, 2 \rho} \]  

(27)
where \( E \) - modulus;
\( \rho \) – specific density of balls.

In addition to the basic excitation frequencies listed above, a variety of combination frequencies are excited in rolling bearings due to defects in contacting surfaces [12]. As a result of simultaneous interaction of several defects, multiple modulation of main frequency by different signals is possible. For example, a separator defect excites modulation components \((k_{\text{f, out}} \pm n_{\text{sep}})\) in the vicinity of harmonics flicker frequency along the outer ring \( k_{\text{f, out}} \); inner ring defect excites the frequencies of interaction with the outer ring \((k_{\text{f, out}} \pm n_{\text{in}})\) and with rolling bodies \( k(f_{\text{in}} - f_{\text{sep}})z_{m} \). Various combinations and other components are also possible, resulting in combinational frequencies appearing in the vibration spectrum.

Presented formulas for calculating bearing frequencies may be useful for use in diagnostics, but it should be remembered that they should be used very carefully, reliability of results obtained with their use may not be too high. There is a clear defect in the vibration signal in bearing, characteristic frequencies may be completely absent, have a frequency shift or have a very low level.

To assess the technical condition and diagnosis of defects in rolling bearings, quite a lot of different methods have been developed at present; considered in detail in numerous works [13–18]. It is natural that all methods, which differ in theoretical assumptions, have different complexity and reliability, require different instrumentation and are intended for different purposes. However, they have one thing in common — they are all based on a deep analysis of vibration parameters measured in the immediate vicinity of bearing. As practice shows [13], it is almost impossible to install a vibration detector on bearing housing in a printing house. The only place that is suitable for installing detectors is a section of the frame that is free of shrouds and attachments, usually located quite far from vibration source under investigation. This seriously complicates the task.

In order for harmonics with characteristic frequencies to be reliably detected in a vibration spectrum in the presence of a clear bearing defect, a number of requirements must be met.

The main requirements are as follows:

- bearing must be loaded with a sufficient force close to the nominal one;
- defective zone must periodically pass through the bearing load zone;
- mechanism must not have other sources of vibration signals with a frequency equal to the frequency of defects;
- vibration detector must be located close enough to loaded area of bearing;

Requirements apply to all methods of diagnostics of rolling bearings based on vibration spectrum and envelope spectra, which are based on the use of above formulas for calculating the characteristic frequencies of bearings.

4. Summary and conclusions

The task of the study is to investigate the possibility of diagnosing the technical condition of bearings of the printing machine’s ink apparatus from the emitted noise using vibration measured on the frame as an additional source of information.

Theoretical research has shown, there is a possibility in principle to use the noise spectrum emitted by the inking system to diagnose the technical condition of its drive.

It is obvious that it is impossible to solve such a problem without using modern methods of image recognition and artificial intelligence, as well as conducting additional laboratory studies. It can be very useful to use the temperature of bearing assembly as an additional diagnostic sign, since when vibroacoustic diagnostics of mechanical systems of printing machines using neural network technologies, the addition of diagnostic signs of a different physical nature significantly increases the reliability of diagnosis.
References

[1] Kulikov G B 1979 Research and development of active means of dealing with noise emitted by hollow shafts and cylinders of printing machines. PhD thesis (Moscow: MSUP)

[2] Balitskii F Y, Genkin M D, Ivanova M A, Sokolova A G and Khomiakov E I 1990 Modern methods and means of vibration diagnostics of machines and structures Ed. 25. Scientific and technical progress in mechanical engineering (Moscow: International Center for Scientific and Technical Information)

[3] Dąbrowski Z and Dziurdź J 2016 Archives of Acoustics 41 783–789

[4] Song H, Huang W and Chang S 2020 Ocean Engineering 195 106746

[5] Yang F and Du F 2019 International Journal of Mechanical Sciences 157-158 198 – 206

[6] Plakhov D D 1973 Acoustic magazine 19.1 80 – 87

[7] Berlint M V 1975 Acoustic magazine 21.6 839 – 844

[8] Genkin M D and Sokolova A G 1987 Vibration acoustic diagnostics of machines and mechanisms (Moscow: Mechanical engineering)

[9] Niu L, Cao H, Hou H, Wu B, Lan Y and Xiong X 2020 Mechanical Systems and Signal Processing 138 106553

[10] Liu J, Shi Z and Shao Y 2017 Nonlinear Dynamics 89 1 – 18

[11] Bykov A V 2002 Development of a diagnostic technique for rolling bearings of a printed pair. PhD thesis (Moscow: MSUP)

[12] Nikolaychuk A N and Doroshev Y S 2014 Mining informational and analytical bulletin (miab) S4-11 38 – 51

[13] Kulikov G B 2006 Basics of vibration-acoustic diagnostics of printing equipment: Monograph (Moscow: MSUP)

[14] Dybała J 2018 Measurement 126 143 – 155

[15] Barbini L, Eltabach M, Hillis A and du Bois J 2018 Mechanical Systems and Signal Processing 103 76 – 88

[16] Liu J and Shao Y 2016 Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics 230 387–400

[17] Stanik Z 2014 Transactions on Maritime Science 3 111–118

[18] Schwack F, Byckov A, Bader N and Poll G 2017 72nd Annual Meeting and Exhibition of the Society of Tribologists and Lubrication Engineers, STLE pp 21–25