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Contents

118. The new Method of Belt Drive Testing

R. Mikalauskas, V. Volkovas
Kaunas University of Technology, Lithuania

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1. Introduction

The recent investigation [1] of the new dynamic model of the belt drive with longitudinal moving and local heterogeneous flexible element (v-belt) that evaluate parametrical, forced excitation and non - linear oscillations of the belt, showed that by reaching an appropriate quantitative and qualitative level of local heterogeneity, oscillation forms start to change, the meaning of force occurring due to deformation of a flexible element significantly increases.

Then it was created a new model of the belt drive with a flexible element (V-belt) and local heterogeneity [2], which was based on FEM. The model described the interaction of the flexible element with the pulley in the area of a contact angle. The task of dynamics of the system was solved: natural frequencies of the system calculated, system response to harmonic excitation was found. It was determined that there is a correlation between a defective state of the flexible element of the belt transmission and vibroactivity of the driving pulley bearing point, which can be analysed applying presented model. With the help of the suggested model, we could examine correlation between the defective state of the system's belt and dynamic component of vibroactivity of the reaction force of the driving pulley. The results of analysis - the identified spectrum components, are useful for practical vibrodiagnostics of transmission.

2. Experimental research of real objects

2.1 The measuring object and used equipment

On purpose to test experimentally the possibility to apply a new method for transmission belt diagnostics, oscillation measurements were made for bearing point of driving pulleys exploited in the joint-stock company "Dirbtinis pluoštas".

For measurements were chosen belt transmission of pumps, which supply acetyl cellulose and acetone solution for spinning machines. In the process belt transmission is used to transfer rotational movement from electromotor shaft to the primary shaft of the reducing gear. The secondary shaft of the gear is connected with the shaft of hydraulic pump through the sleeve. The main diagram of this technological process is shown in Figure 1.

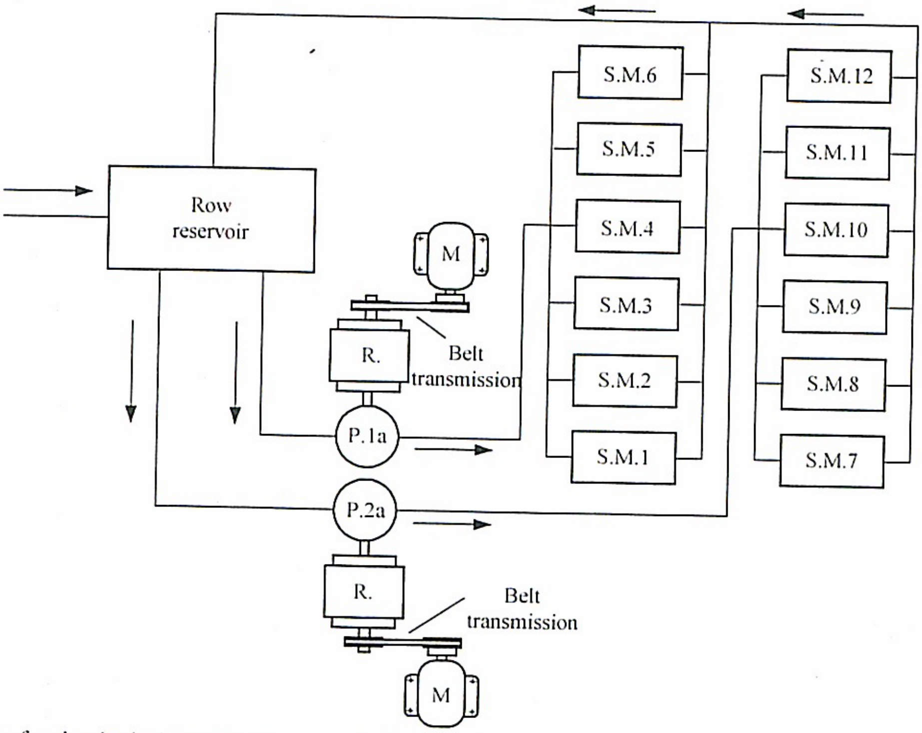

Fig. 1. Main diagram of technological process of raw supply for spinning machines: P1a, P2a – pumps; R. – reducing gear; M – motor; S.M. – spinning machine

In workload mode of hydraulic pumps maximum pressure is up to 20 MPa.

When fixed distance between axes is given, the belt transmission package consists of three A-type 13 x 1150 belts, made by firm "Barum" (Czechia). Geometric and kinematical parameters of hydraulic pumps' belt transmissions are given in Table 1.

Table 1

Characteristics of belt transmissions

Calculating diameter of driving pulley d_1, mm	90
Calculating diameter of driven pulley d_2, mm	195
Length of belt branch having no contact with pulleys L, mm	370
Frequency of motor shaft rotation f_1, Hz	25
Frequency of the gear primary shaft rotation f_2, Hz	12
Belt linear velocity v, m/s	7.4
Belt revolution frequency f_{ap}, Hz	6.5
Belt linear density ρ, kg/m^3	0.1

Also, before measuring bearing point oscillations, was measured initial tension of belts that construct transmission package. The tension was founded by changing the distance between motor and primary shaft of the reducing gear. Deflection of belt branches were measured in lateral direction in order to find initial tension.

Later, according to known initial tension values and kinematical parameters of transmissions (Table 1), lateral oscillations natural - frequencies of belt branches were measured. Calculated values are provided in Table 2.

Table 2

Parameters of transmission packages belts

Belt transmission of pump "1a"	
Initial tension force of the first belt F_{01}, N	284.0
Lateral oscillation frequency of the first belt branches, $f_{sk.1}$, Hz	71.5
Initial tension force of the second belt F_{02}, N	440.0
Lateral oscillation frequency of the second belt branches, $f_{sk.2}$, Hz	88.5
Initial tension force of the third belt F_{03}, N	352.0
Lateral oscillation frequency of the third belt branches, $f_{sk.3}$, Hz	79.0
Belt transmission of pump "2a"	
Initial tension force of the first belt F_{01}, N	506.0
Lateral oscillation frequency of the first belt branches, $f_{sk.1}$, Hz	95.0
Initial tension force of the second belt F_{02}, N	312.0
Lateral oscillation frequency of the second belt branches, $f_{sk.2}$, Hz	74.7
Initial tension force of the third belt F_{03}, N	430.0
Lateral oscillation frequency of the third belt branches, $f_{sk.3}$, Hz	88.0

The measurements were performed on driven pulleys of belt transmissions "1a" and "2a" bearing points, that is on primary shafts bearing points of corresponding pumps reducers. Bearing-point vibration velocity was measured in horizontal and vertical directions. Measuring points are shown in Figure 2.

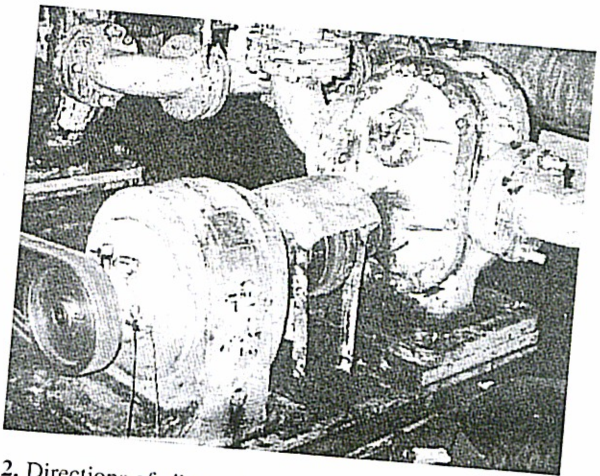

Fig. 2. Directions of vibroactivity measuring for gear primary shaft bearing point

For experiment was used tipical vibration measurement equipment. It consists of seismic transducer, amplifier, interface and PC. For vibration analysis was used a special program [4] designed for PC. It performs correlation spectral analysis and is designed for MATLAB® programmable package. The final results are spectral graphs of low frequency vibration in two different measuring directions. Measurements were performed while pumps were working in workload and free regimes. Belt transmission bearing point vibrations were measured every 0.5±0.1 and 1.0±0.2 hour taking every time twenty realizations for each work mode individually. Averaging these realizations for different pump work modes, bearing points vibration spectrum was found.

2.2. *Measuring data of bearing point vibration*

After measuring bearing-points vibration velocity signals in horizontal and vertical directions, when different pumps work modes are given, and after performing their correlation spectral analysis, our attention is paid to harmonics which are multiple for frequencies of belt revolution, revolution of belt transmission driving and driven pulleys. These frequencies are spectrally closest to frequencies of belt branches lateral oscillations. In such case, on the ground of precalculated and provided in Tables 1 and 2 data, the frequency of pump "1a" belt transmission will be component of 72 Hz (it is multiple to revolution frequency of reducing gear) and the frequency of the pump "2a" belt transmission will be component of 75 Hz (it is multiple to revolution frequency of motor shaft).

Experiment results, vibration velocity spectrum of bearing points measured in horizontal and vertical directions, are given below.

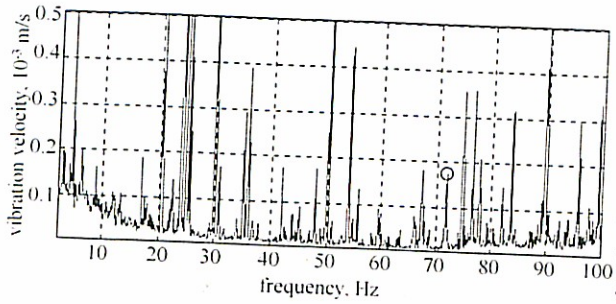

Fig. 3. Work without load of pump "1a". Spectrum of horizontal vibration of reducing gear bearing point

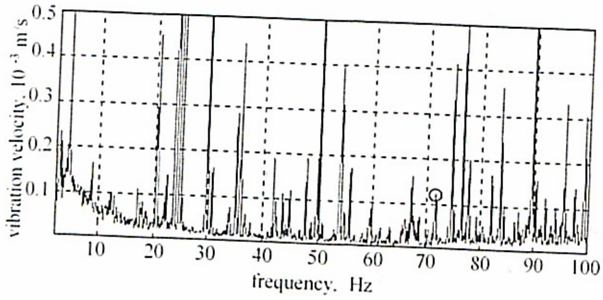

Fig. 4. Work in workload mode of pump "1a". Spectrum of horizontal vibration of reducing gear bearing point

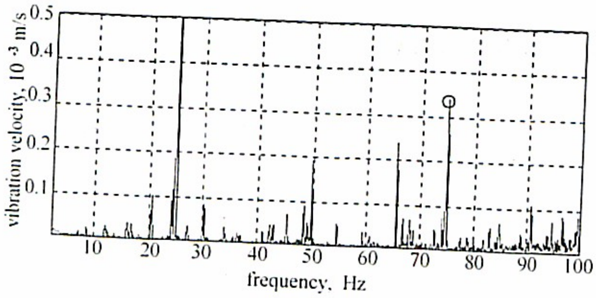

Fig. 5. Work without load of pump "2a". Spectrum of vertical vibration of reducing gear bearing point

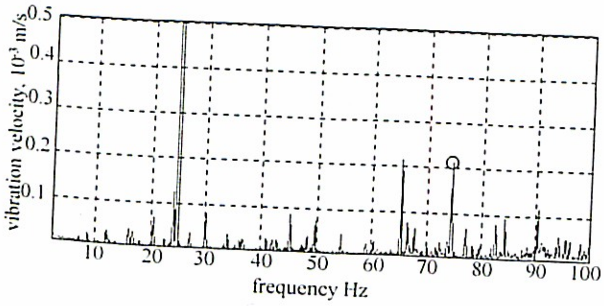

Fig. 6. Work in workload mode of pump "2a". Spectrum of vertical vibration of reducing gear bearing point

After measuring bearing point vibration of both pumps reducing gear we see, that spectrum harmonics, which are multiple for revolution frequencies of reducing gear and motor shaft, their amplitudes decreases in both directions, when pumps are working in workload mode. This can be explained on the ground of theoretical research, performed in work [1]. As above was mentioned, before measuring bearing point vibration, according to known parameters of exploited belt transmissions and the belts themselves, we excluded spectrum harmonics above 72 Hz and 75 Hz. It was made so since lateral oscillation frequency of first belt branches of pump "1a" belt transmission was 71.5 Hz, while pump "2a" belt transmission second belt branches lateral oscillation frequency was equal to 74.7 Hz. In other words, lateral oscillation natural-frequency of mentioned belts was coincident with system excitation frequencies (at one time it was frequency multiple to reducing gear shaft revolution frequency, at another time was multiple to revolution frequency of the motor shaft). While pumps were working in a free mode, by given lesser resistance moment at the side of belt transmission driven pulley, the difference between tension of transmission belt driving and driven branches is not large and that means, that their lateral oscillation frequencies remain practically unchanged and still are close to system excitation frequencies. While pumps are working in workload mode, at the side of transmission driven pulley resistance mode increases. That causes increase of belt orbicular force and the difference between tensions of belt branches. As a result, transmission belts driven and driving branches lateral oscillation natural frequencies change – driving branch frequency increases and driven decreases. Thus it happens so, that while working pump "1a" is in workload mode, its lateral oscillation frequencies of transmission first belt driving and driven branches "move" from system excitation frequency (from transmission driven shaft revolution frequency move 72 Hz) and in case of pump "2a" load mode its belt transmission second belt appropriate frequencies "move" from multiple driving shaft revolution frequency that is from value of 75 Hz. In such conditions it is proved by the results of theoretical calculations [3], that when lateral oscillation natural frequencies of transmission belts branches do not coincide with system excitation frequencies, driven pulley bearing point vibroactivity also decreases. That is proved by obtained vibration measurements of belt transmissions driven pulleys bearing point.

3. Method for belt drive diagnostics

With the help of mentioned models [1, 2] and practical measuring of real objects was created a new method for diagnostic procedure of belt drive. The method enable to estimate on the state of flexible element during exploitation. Method of diagnostics consists of calculation and practical measuring procedures. Its nature is shown by the algorithm, which sequence of actions is listed below.

3.1. Calculation procedure

1. For mounted drives the distance a (mm) between axes of the driving and driven shafts and respectively the

diameters of their pulleys d_1, d_2 (mm) are determined accordingly to its documentation or by measuring.

2. Length l of the belt branch, which has no contact with the pulleys is calculated.
3. For calculation the length L of belt is determined (mm).
4. Initial tension force of belt branch is determined. Using known value loads over belt branch centre, its bend δ by transverse direction is measured. The measured value of transverse bend and known value of load calculate value of the tension force.
5. Revolutions of driven and driving shafts n_1, n_2 (min^{-1}) of belt drive are measured while working in free motion regime. Linear velocity of belt v and multiple frequencies of belt revolution $f_{ap,k}$ are calculated or measured.
6. Accordingly to known characteristics of used belts, linear density of belt ρ (kg/m) is calculated. If there is no such data, the value of linear density is determined experimentally.
7. Using provided, calculated parameters of the belt drive and of the belt itself and the differential equation expressing transversal oscillations of the belt branch [1], we calculate the natural frequency of transverse oscillations of flexible element branches.
8. Based on the known data of exploited belt drive, nomograms are made for belt diagnostics. The main point of nomograms is dependency of system (belt drive) excitation frequency and the belt natural frequency upon belt linear velocity. Belt natural frequency is determined by tension force, which depends on belt state.

As an example we shall provide specific diagrams of relationship between belt drive excitation frequencies and its belt natural frequency. Parameters of belt drive, working in free-motion regime: F_0 = 350 N; d_1 = 90 mm; L = 370 mm; ρ = 0,11 kg/m; v = 7,45 m/s.

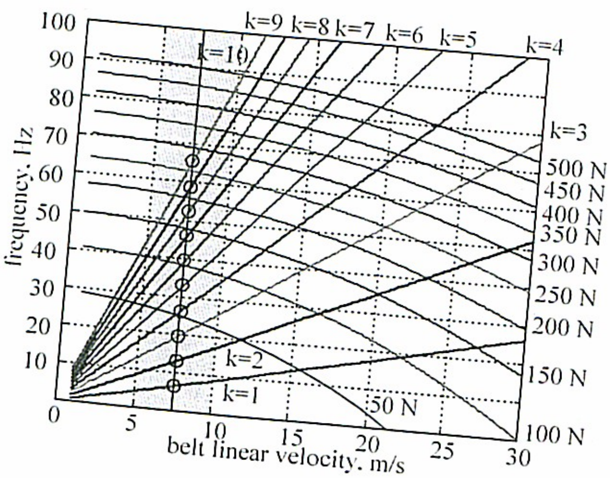

Fig. 7. Dependency of multiple for belt revolution frequencies and belt lateral oscillation frequency upon belt linear velocity

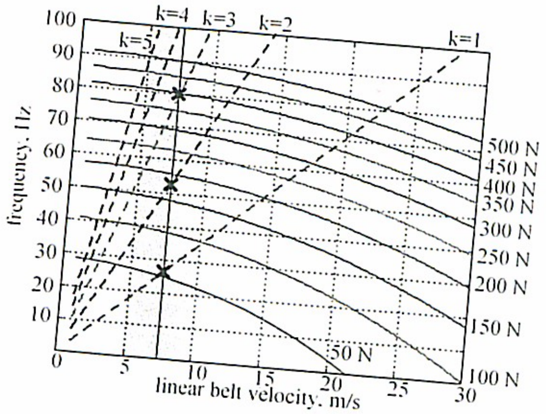

Fig. 8. Dependency of pulley revolution frequencies and belt lateral oscillation frequency upon belt linear velocity. Belt transverse oscillation frequency is determined by tension force

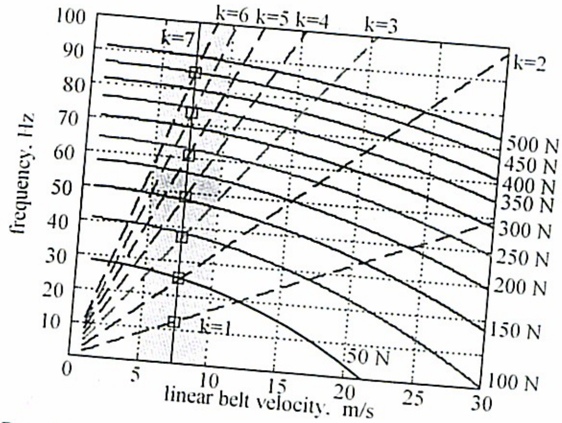

Fig. 9. Dependency of multiple driving pulley revolution frequencies and belt lateral oscillations frequency upon the belt linear velocity. Belt lateral oscillation frequency is determined by tension force

In Figures 7 - 8 depicted ordinate points of intersection of vertical black lines with multiple excitation frequencies (blackened with circles, squares and laterales) will be informative component of vibration spectrum from diagnostic point.

After measuring vibrations of bearing point of new or assembled after repair belt drive and performing spectral analysis of the signal, we fix values of obtained spectral informative components, which are in frequency range (0.6-1.0) $f_d^{(1)}$, where $f_d^{(1)}$ - the first natural frequency of belt lateral oscillation, calculated accordingly to drive parameters and this data is considered as basic.

3.2 Procedure of Practical Measuring

Measuring bearing point oscillations periodically, the values of obtained spectrum informative components are compared with basic data. If the value of the component of specific measuring spectrum, corresponding to one of the system excitation frequency increases more than 15% comparing with basic value, it is considered that the frequency of transverse oscillations of belt branches is close to mentioned excitation frequency. Accordingly to earlier made nomograms and current frequency of belt branch lateral oscillations we indirectly identify current belt state, which is determined by tension force value. Decision of belt suitability to exploitation is accepted accordingly to determined tension force value.

As example we provide spectrum of oscillation velocity of belt drive bearing point. Here accordingly to earlier provided system frequencies and belt lateral oscillation frequency, which is determined by tension force, its dependencies upon linear velocity multiply components of individual system excitations were excluded.

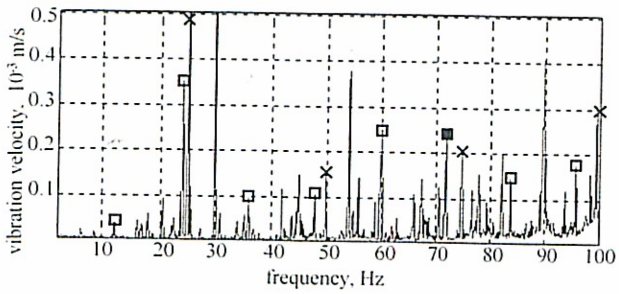

Fig. 10. Suggestion on the base of nomograms exclude one pump belt drive multiple driving and driven pulleys excitation frequencies and component of belt lateral oscillations: x – driving pulley; □ - driven pulley; ■ - frequency of belt lateral oscillation

Provided method may extend possibilities of ISO 10816 standard, where beside diagnostics of bearing of mechanisms with belt drives, the state of belt may be estimated.

4. Conclusions

Experimental research of real objects, using mathematical models [1, 2], proved that there is a possibility to determine indirectly the state of the belt itself. On that base was created a new method of the belt transmissions diagnostics. According to it, during maintenance it is possible to identify the state of flexible element and determine reliability of the mechanism itself.

Suggested methodology is simple, general-purpose and can be applied to transmissions with flat, round and toothed belts, because theoretical modelling used only generalised mechanical system parameters (for example, belt tension force). Experimentally analysed A-type belt exploitation defects are typical for round, flat, toothed and transmission package constructing belts, in modelling used equations do not depend on the type of flexible element. Therefore from qualitative point of view during exploitation showed defects will have the same influence for generalised parameters of mentioned belts.

5. References

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