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VIBRATION DAMPING BEHAVIOR OF A CONVEYOR BELT AT LOW TEMPERATURE

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Abstract: Understanding the vibration-damping behavior of a conveyor belt at low temperatures is crucial for stable operation. The movement of the conveyor belt on idlers becomes more stable if the vibration damping is adequate. Material properties may undergo significant changes at low temperatures. This study examines the vibration-damping characteristics of a conveyor belt under low-temperature conditions. The objective is to ascertain the characteristic behavior of a conveyor belt with textile plies under specified conditions. This study examined the variation in the loss factor of the material across a temperature range of -50 °C to 20 °C, focusing on both static and dynamic shear moduli. The Brüel & Kjaer measurement system is utilized to adjust cyclic loads and vibration frequencies. A specialized refrigerator is utilized in our experiments, achieving temperatures as low as -60°C. The measurements were conducted using commercially available fabric-lined conveyor belt test specimens. Changes in natural frequency and the vibration-damping factor are critical for the application. The tests show that the material properties of the conveyor belt significantly depend on temperature.

***Keywords**: conveyor belt; refrigerated specimen; vibration, damping factor; natural frequency*

1. INTRODUCTION

Vibrations in conveyor belts are common dynamic phenomena during the standard operation of conveyor belt systems (Yan and HE, 2010). The appropriate vibration characteristics are an important factor for the smooth operation of the conveyor belt (Harrison, 1992; Rana et al., 2019), and unsuitable vibration behaviors can lead to failure (Zimroz and Król, 2009; Homišin et al., 2019). These vibrations during operation can be longitudinal or transverse according to the direction of travel of the conveyor belt (Chen, 2005; Hu et al., 2015). The longitudinal vibrations of the conveyor belt are the subject of a major research topic (Hu and Meng, 2008; He et al., 2016; Li and Pang, 2018).

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These are related to the propagation and damping of stress waves in the conveyor belt (Harrison, 1985). Transverse vibrations are the vertical cyclic movements of the belt that occur between the sets of idlers. These vibrations are especially hazardous because they can cause the conveyor belt to enter a state of resonance (Bauomy and El-Sayed, 2018). Various models are created to study these undesirable phenomena (Bortnowski et al., 2023).

The influence of transverse vibrations of the conveyor belt on noise emissions is examined. The noise generated by the conveyor belt is directly related to the vibrations (Brown, 2004). In recent years, the requirements for this have become increasingly strict, especially in open-pit mines. Research into transverse vibrations of flexible belts has been going on for a long time (Suweken and Van Horssen, 2003; Wickert and Mote, 1988; Pellicano and Vestroni, 2000; Harrison, 1986; Lodewijks, 1996). The main research goal of most publications is to determine the vibration frequency of the belt using analytical methods (Suweken and Van Horssen, 2003a; Suweken and Van Horssen, 2003b; Andrianov and Van Horssen, 2008; Ding and Li, 2014). During laboratory tests, the measurements are often limited to the static conditions of the stationary belt stretched with a constant force (Hou and Meng, 2008). The frequency of conveyor belt vibrations depends on parameters such as mass, stiffness, geometry of sections, spacing of support sets, linear movement speed and tension force (Abrate, 1992). Some of these parameters can be determined by in-situ measurement on the conveyor belt system, while others were determined by testing the belt sample in the laboratory (Kulinowski et al., 2020; Manjgo et al., 2018). Bortnowski et al. (2021) present the results of vibration measurements performed for a moving belt and obtained for various operating parameters of the conveyor, as well as for several configurations of the distance between idler supports. By creating a good theoretical model with these parameters, the belt vibration frequency can be accurately predicted even at the initial stages of conveyor design. The frequency of the vibrations is described with a high degree of approximation, based on simplified physical models, but studies have not yet verified their usability under dynamic conditions (Abrate, 1992).

It is necessary to look for new solutions, and one of them may be the analysis of vibration damping behavior. The research aims to determine the characteristic behavior of a conveyor belt with fabric plies. In this study, we investigated the variation of the material's loss factor at temperatures from -50 °C to 20 °C, as well as its static and dynamic shear moduli. The Brüel & Kjaer measurement system is used to vary the cyclic loads and the vibration frequencies. A special refrigerator is available to carry out our experiments, which can provide temperatures as low as -60°C. The measurements were made with commercially available fabric-lined conveyor belt test specimens.

2. MATERIALS AND METHODS

In most cases, vibrations are unwanted phenomena caused by cyclical, dynamic effects during equipment operation. Depending on their nature and intensity, vibrations can have adverse effects on the equipment itself over time, its

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environment, and even on the people around it. For this reason, vibration damping significantly reduces and ideally eliminates such adverse effects, whether under cyclic or pulsed loads. Damping can often be categorized into two fundamental types: material damping and non-material damping.

Material damping is present in all materials, as they possess internal damping; however, its magnitude can vary significantly. When comparing several materials under identical boundary conditions, geometrical dimensions, and periodic forcing of the same magnitude and frequency, one material may oscillate with a shorter duration and reduced amplitude than another. This results from the disparity in material characteristics. The damping force resulting from internal molecular friction in this material exceeds that of the other substance. This type of damping is referred to as material damping.

Non-material damping refers to other types of damping, which include friction damping, viscous damping, and acoustic radiation damping. A vibratory structural system will oscillate significantly longer in air than in water. This type of damping is referred to as viscous damping. The viscous damping force is contingent upon the surrounding medium's characteristics and the motion velocity. Friction damping arises from friction within the joints.

All materials lose energy during cyclic deformation. Certain materials, including rubber, plastics, foams, and elastomers, dissipate significantly more energy in each deformation cycle than steel or aluminum. The energy dissipation per unit volume per cycle in conventional structural materials is significantly lower than in some high-damping alloys, polymer matrix composites, and elastomeric materials. These are viscoelastic substances.

Material damping is contingent upon various factors. A viscoelastic substance is often referred to as a material with memory. This indicates that the behavior of a viscoelastic material is influenced by both the present loading conditions and its loading history. The material damping depends on the type or grade of the material, stress level in the damping material, temperature and frequency. The temperature significantly influences the properties of viscoelastic materials.

Based on the temperature dependence of the storage modulus, four regions can be identified (Sun and Lu, 1995) based on the behavior of the storage modulus (E') and the loss factor (η): glassy, transition, rubberlike, and flow regions. The glassy state typically manifests at ambient temperature. In this region, the storage modulus (E') slightly decreases with increasing temperature, while the loss factor (η) demonstrates a sharp increase with rising temperature. The trend will persist into the transition region until the transition temperature (T_t) is attained. At the transition temperature (T_t), the loss factor attains its maximum value, while the slope of the storage modulus (E') remains nearly constant. In the rubberlike region, both E' and η remain nearly constant. This trend persists in the flow region, where the values of E' decrease and η subsequently increase. In engineering applications, structures are utilized beyond the transition temperature (T_t). The transition region can range from approximately 20 °C to 200 °C. The loss factor is minimal in the glassy region and attains a high value at the transition temperature.

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Frequency also significantly influences damping. The influence of frequency on the storage modulus and the loss factor can be categorized into four distinct regions: glassy, transition, rubberlike, and flow zones. Generally, *E'* escalates with an increase in frequency. The rate of rise is minimal in the glassy and rubberlike sections, while the highest rate of increase occurs in the transition region. The loss factor diminishes with increasing frequency in the glassy zone, attains its peak in the transition region at the transition temperature, and escalates in the rubberlike region. In the flow zone, *E'* diminishes and η increases with rising temperature.

The damping material's stress level is the most critical factor. The consequences of cyclic strain amplitude on damping behavior are exceedingly intricate. Initially, the material's temperature will increase due to the increased energy dissipation caused by a high dynamic strain amplitude. Subsequently, the damping behavior is coupled with these two effects. In the glassy region, the variation of *E'* and η with strain amplitude is negligible; *E'* experiences a rapid decline and η reaches its maximum value in the transition region. The modulus and loss factor are affected by temperature to a lesser extent in the rubberlike region. Consequently, the dynamic strain amplitude affects *E'* and is more severe in the rubberlike region than in the other regions. The modulus will be increased, and the loss factor will be decreased due to the tensile static preload.

In particular, the resonance phenomenon can cause intense, even destructive effects. Each piece of equipment has one or more natural frequencies specific to its structural design and material quality. Upon excitation, the structural elements will vibrate, and these natural frequencies will be the parts of the spectrum that show dominantly superior vibration levels and will be the longest to decay completely after the excitation is removed. These are the frequencies where the most significant effect can be achieved with the least excitation energy. The attenuation of the free vibration - at the natural frequency - after the excitation has ceased is characterized by the logarithmic decrement (δ), which is the ratio of two consecutive amplitudes. With constant damping, it is shown in Figure 1 (Farkas and Jármai, 1997).

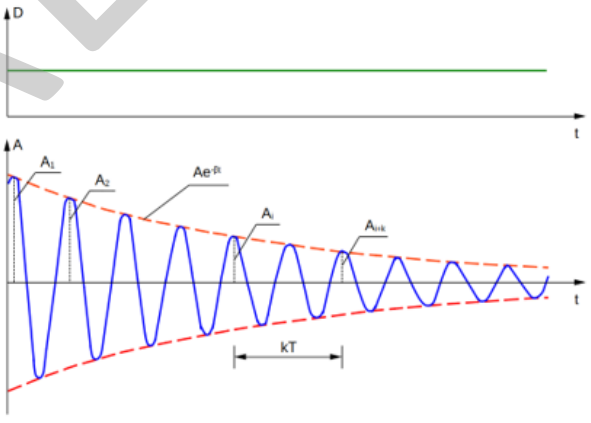


Figure 1
Free vibration with constant damping

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In general, the logarithmic decrement is defined as

$$\delta = \frac{1}{k} \cdot ln \frac{A_i}{A_{i+k}} \tag{1}$$

The damping factor can be calculated as

$$\eta = \frac{\delta}{\pi} \tag{2}$$

Suppose the excitation corresponding to the natural frequency is continuously maintained, and the phase of the excitation signal - at a constant amplitude - is in proper relation to the phase of the motion of the already vibrating structural element. In that case, its amplitude gradually increases as a self-excitation process (Figure 2), and a resonance phenomenon occurs, the extent of which is limited only by the damping characteristic of the structure. In theoretical limit cases, if this damping is zero, the vibration amplitude will be infinite, and if the damping is infinite, the amplitude is zero. In practice, this can be characterized by the resonance magnification (v), which is given by the ratio of the magnitude of the stimulus signal (A_s) to the magnitude of the response signal (A_r) (Dömötör, 2008)

$$v = \left|\frac{A_r}{A_s}\right| \tag{3}$$

where in the case of resonance $v \gg 1$.

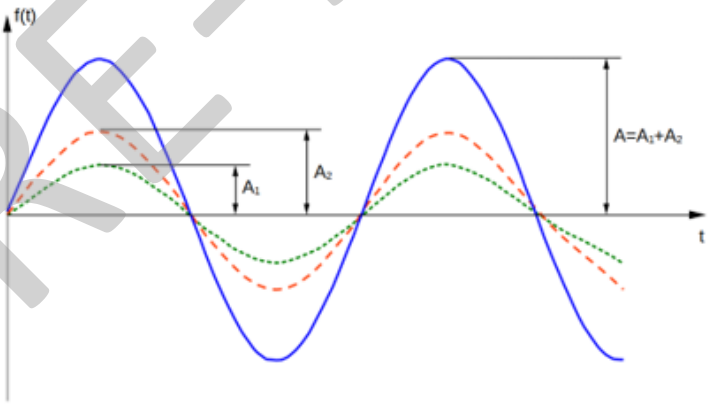


Figure 2
Amplification of unidirectional harmonic oscillation

In practice, the maximum vibration amplitude that is ultimately produced for a given excitation is limited by the damping properties of the structure. Several methods are available for quantifying the damping, of which the Oberst method is

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used in the tests. In this case, the value of the damping factor is given by the ratio between the resonant frequency (f_n), and the frequency bandwidth that belongs to the corresponding amplitudes of 3 dB damping points ($1/\sqrt{2} = 0.707$ times, meaning half power) (f_1 and f_2) (Figures 3 and 4) (Farkas and Jármai, 1997).

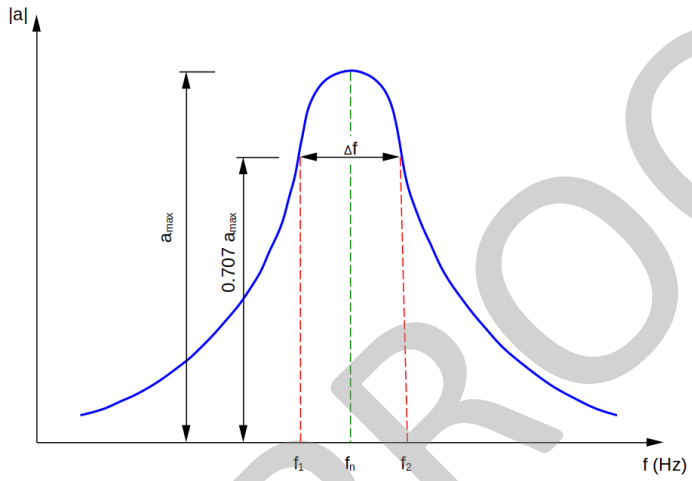


Figure 3

Determination of the damping factor based on Oberst method

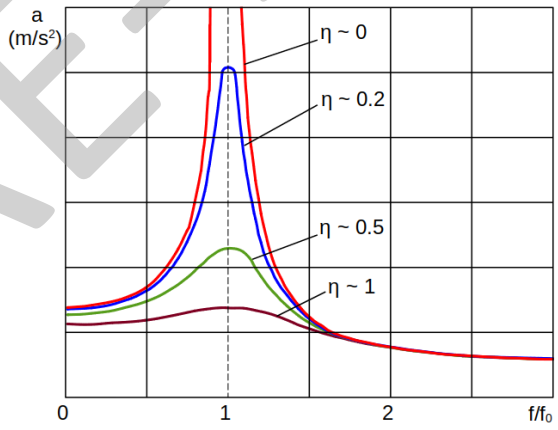


Figure 4

Determination of resonance curves at different damping factors

Suppose the magnitude of the excited vibration is too high compared to the permissible level. In that case, vibration damping is required at the resonance

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frequency, or the frequency range should be outside the resonance. There are several possible solutions, such as:

- structural and material design of the (chassis) structure,
- mechanical and electrical design of the rotating machine drive,
- installation of damping devices (springs, shock absorbers, masses, etc.),
- use of damping materials (rubber, foam, composites, etc.).

If we want to optimize the damping behavior of our structure or material, we must use mathematical optimization techniques (Jármai et al., 2021). For other structural applications, one can use these damping materials (Hirohata et al., 2021).

3. TESTING THE CONVEYOR BELT

This section may be divided by subheadings. It should provide a concise and precise description of the experimental results, their interpretation, and the experimental conclusions that can be drawn.

This research aims to investigate the temperature dependence of a given damping, which is mainly related to using damping materials from the above, as their structural properties can change significantly at extreme temperatures. This must be considered when the damping structure is used, for example, in extremely cold winter climates or aeronautical applications. From another approach, this research aims to develop the necessary test method with the available tools and to test it in practice. Thus, at this stage, the aim is not to obtain absolute scientific precision of the results concerning the material under investigation but to gain experience in the applicability of the method, the excitation and data acquisition tools and the measurement evaluation program, including further development possibilities and needs.

The test subject is the ENERGUM EP/630/4, 10+2 mm fabric carcass conveyor belt test specimen (Figure 5).



Figure 5
Energum EP/630/4 strap cross-section

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EP630/4 conveyor belts dominate the market mainly because most conveyor lines worldwide run short distances under 200 m, and carry relatively light loads. At that range, the belt only experiences moderate longitudinal pull, so the tensile strength is not very high. On the other hand, the 4-layer requires a 360-600 mm conveyor pulley diameter, which is currently the most common on the transportation site. Finally, this type of conveyor belt has strong versatility and can be applied to most conveyor sites. It is very popular in the used conveyor belt market, and the economic benefits will be maximized. Its main properties are shown in Table 1.

Table 1

Main features of Energum EP/630/4 strap

Total thickness	15,5 mm
Its top + bottom cover thickness	10 + 2 mm
Fabric ply material	EP 160 polyester/polyamide
Operating temperature	-30...+80 °C
Mass	18,5 kg/m^2

With this material, two types of measurements have been performed, one with only the conveyor belt itself (cyclic bending load) and the other as a Jones device (cyclic shear load), as shown in Figures 6 and 7.



Figure 6

Simple conveyor belt specimen on the shaker

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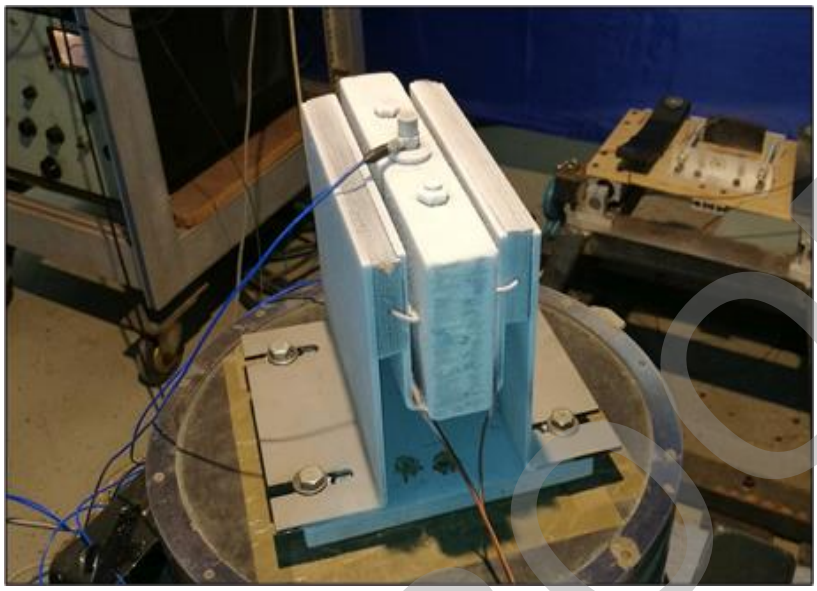


Figure 7
Deep-cooled Jones device on the shaker

For extreme cooling of test specimens, we have a VT147-type ultra-low temperature freezer that can reach -60 °C. This currently available cooling method allows the test to be performed only by inserting the specimen into the freezer, cooling it, and then removing it for excitation and measurement while it is heated back to ambient temperature. This is not ideal, firstly because the temperature of the specimen does not vary homogeneously across the whole cross-section (this was investigated using a Jenoptic VarioCam thermal camera (Figure 8), and secondly, because the variation, measured at a given point with a thermocouple, is not linear in time (Figure 9) and cannot be controlled. However, the primary objective above can be achieved with this method. The ideal solution would be a suitable climate chamber, which would entail unrealistic investment costs in the current economic situation.

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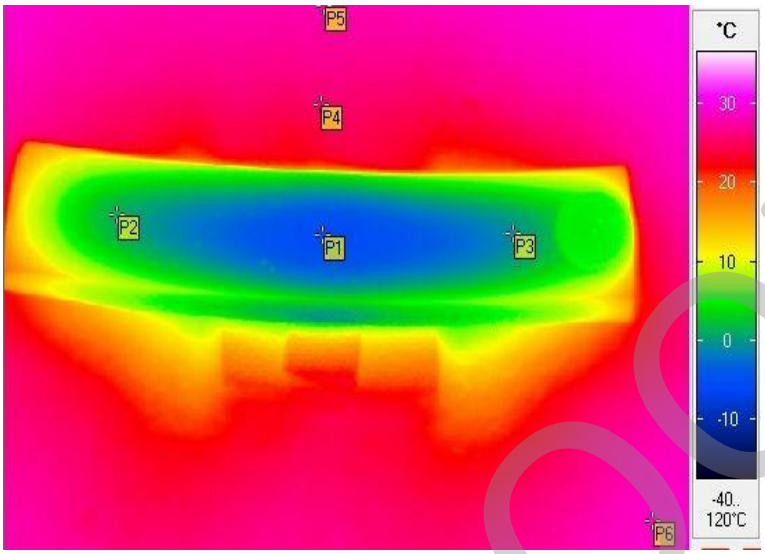


Figure 8
Top-down thermal image of the specimen on a thermally isolated table

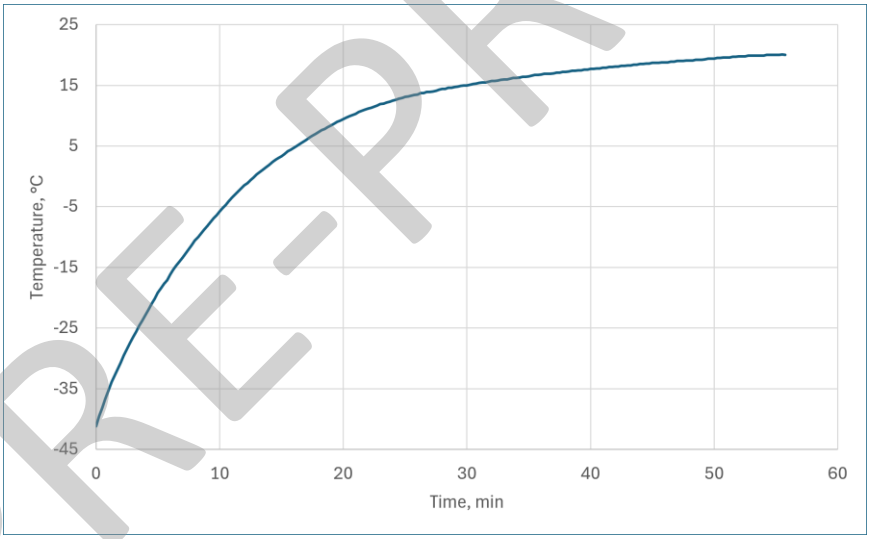


Figure 9
Test point temperature variation

The nature of the excitation, in our case, can be of three types:

- Pulse: a special “impact hammer” is used to test the decay of the vibration in response to a known energy impact on the test piece.
- Sine wave: sweep the entire frequency band with constant acceleration by feedback and variable frequency excitation according to defined parameters. In this case, the effect of only one frequency spectrum component is

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investigated at a time. This method is not ideal in this case because under the combined conditions of the continuous variation of temperature and frequency with a given time factor (linear for frequency and non-linear for temperature) and the minimum sampling time per "frequency point" to be measured, the measurement results (in our experience) cannot adequately reflect the real behavior (Figure 10).

- White noise: The white noise contains all components of the frequency spectrum under the test with equal probability so that it can be equated to the sum of many individual sinusoidal excitations, but in this case, over a significantly shorter (even orders of magnitude shorter) period. This is an essential consideration because of the relatively rapidly changing temperature during the measurement. However, the nonlinearity of this change has no significant effect on the final result due to the shorter measurement time. In this case, the measurement is performed without feedback by setting a constant excitation signal level. In this case, this is the best of the three available methods.

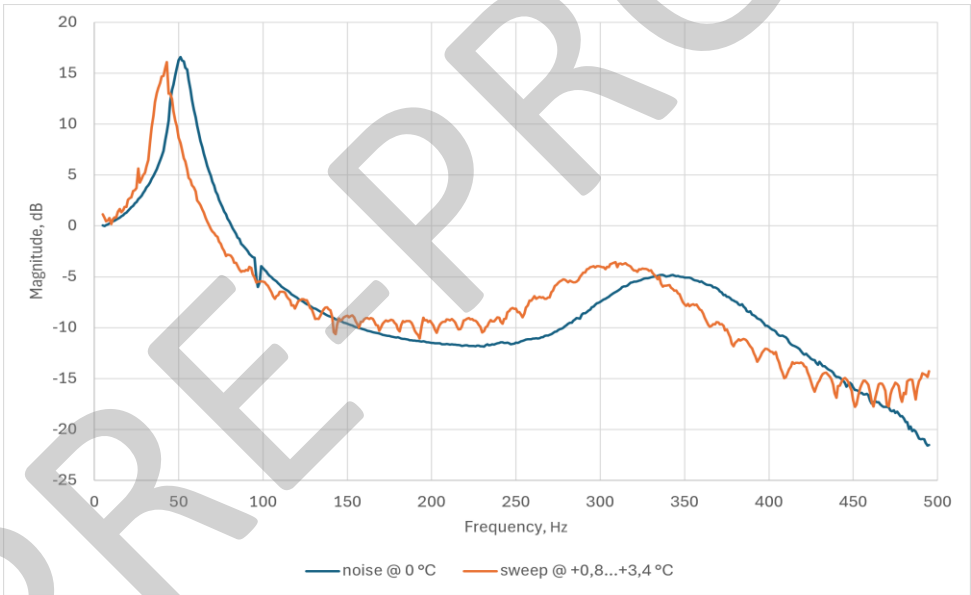


Figure 10

Comparison of white noise and sweep excitation results at nearly the same temperature

The excitation is provided by a high-performance Brüel & Kjaer shaker system and the data acquisition by a National Instruments CompactDAQ system with SignalExpress software (Figure 11).

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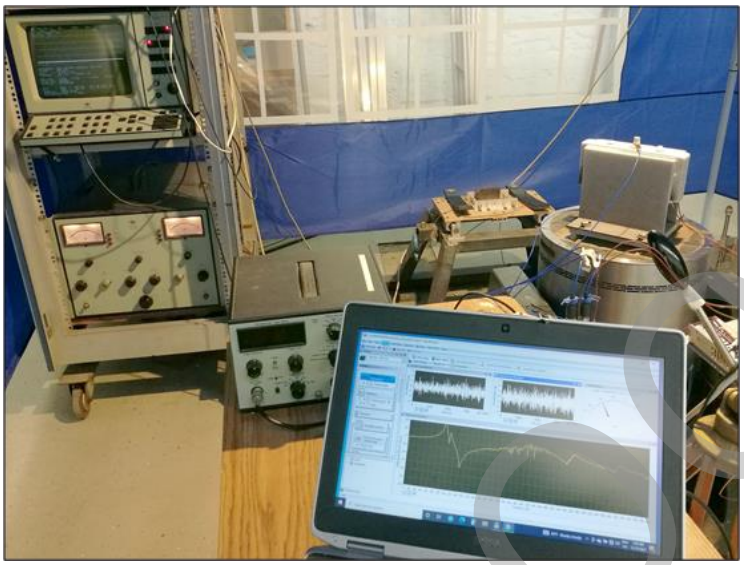


Figure 11
The B&K and NI excitation measurement system

Characteristics of the main elements:

- B&K 4818 type electrodynamic shaker: with the full load (including the fasteners), it can produce >200 m/s^2 acceleration, which is well within the 10-15 m/s^2 that we have found to be sufficient for this task. The shaker stands on an adequately designed vibration-free base.
- It is powered by a B&K 2708 (1200 W) power amplifier.
- The B&K 2034 spectrum analyzer has a built-in signal generator to generate white noise and B&K 1050 controller for sine wave sweep.
- The instrumentation is NI cDAQ-9178 USB frame with NI-9234 4ch. accelerometer and NI-9219 temperature measurement module with two PCB Piezotronics 352C33 accelerometers with 100 mV/g sensitivity and a T-type thermocouple.
- NI SignalExpress software for measurement data collection.
- Octave (MATLAB-compatible) software for measurement evaluation.

4. MEASUREMENT RESULTS

The excitation was performed with white noise as described above for both types of test specimens (also with a sine wave as a test, but this was discarded based on the results obtained as described above). For the measurement, one channel measured the acceleration of the excitation (stimulus), the other the response, and the measurement program developed in SignalExpress determined a frequency response spectrum in the frequency range 1...850 Hz and in the magnitude range of approx. -60...+20 dB by means of 15-second data averages, which were recorded in separate files on the computer with the corresponding measured temperature.

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Since we do not have a suitable climate chamber (for the time being), the temperature of the specimen taken out of the freezer has already increased despite the quick fixation. Without active heating, it has not warmed above ambient temperature (the aim is basically to test the behavior in cold conditions), so the possibilities for the simple specimen for the presented measurement are between -43... +20 °C, with 225 files recorded in about one hour due to the rate of heating, while for the Jones test specimen, which has much more thermal capacity, we were able to perform a complete measurement between -53...+13 °C, in about three hours and 674 files, limited in the latter case by the excessive heating of the shaker due to continuous operation and the lower ambient temperature.

A preliminary examination of the measurement data of the simple test specimen (at seven different temperatures) using only a simple spreadsheet data process showed results that justified further analysis (Figure 12).

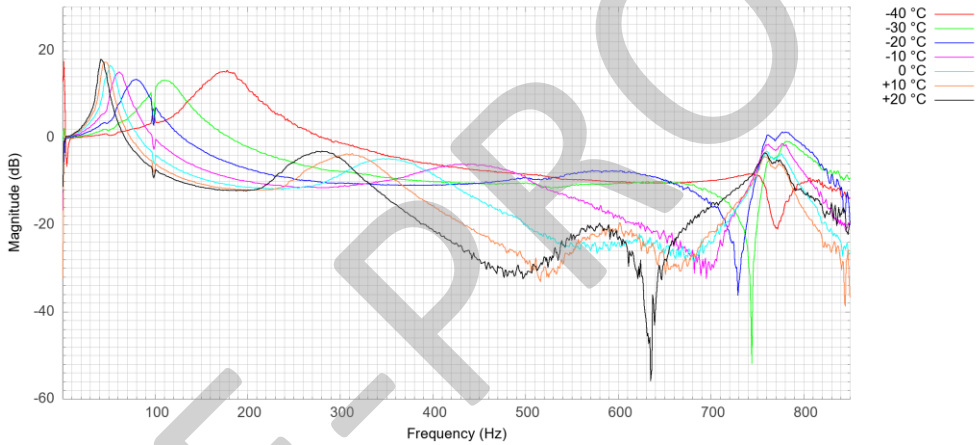


Figure 12

Frequency response diagram of a simple test specimen at different temperatures

It can be clearly observed that the first two natural frequencies shift significantly to the left in the spectrum, and their magnitude increases somewhat with increasing temperature. The non-varying frequency "hole" at around 100 Hz is caused by the system's own characteristic, the vibration from the shaker's cooling fan. Above about 500 Hz, the complex behavior of the specimen, the mounting structure, and the shaker and its support structure are already apparent. Therefore, the temperature dependence is less pronounced, thus we will focus on the range below 500 Hz. The determination of the damping factor (especially as a function of temperature) would have been a lengthy task here, especially for hundreds of files, so an Octave (MATLAB-compatible in our case) program (www.octave.com) was prepared to do it automatically and fast:

- scan the files one by one and display the individual spectrum diagrams,
- find the natural frequency local peaks and determine their magnitude,

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- from these, determine their 3 dB (half-power) bandwidth (Figures 13 and 14),
- from these, calculate the Oberst damping factors,
- finally, save all these data in a file so they can be imported into a spreadsheet for further analysis as required.

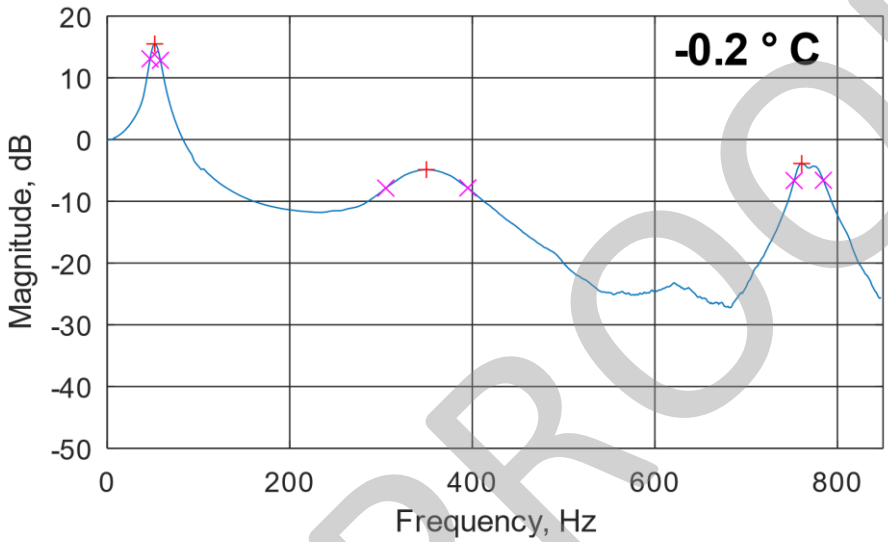


Figure 13

Natural frequencies of a simple test specimen at a given temperature, showing the local peaks and 3 dB points

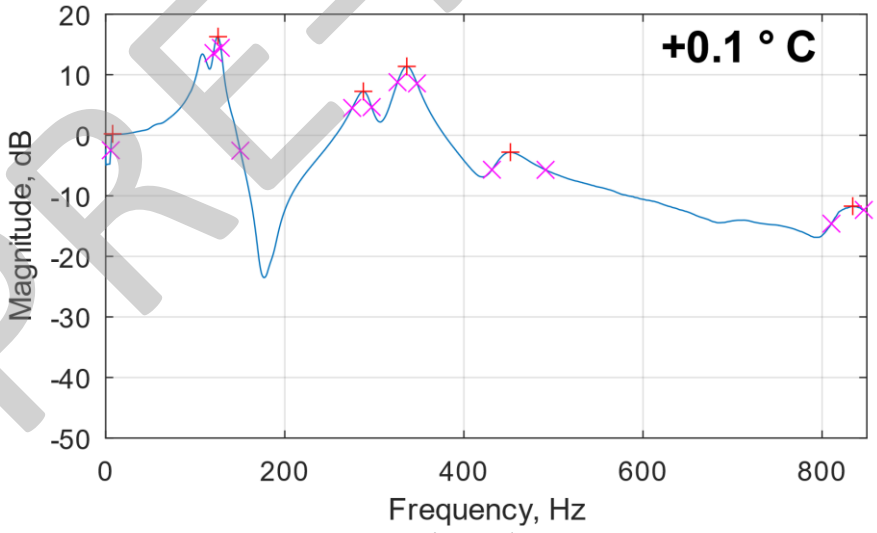


Figure 14

Natural frequencies of the Jones device at a given temperature, showing the local peaks and 3 dB points

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Based on this, Figures 15 and 17 show the variation of the first four natural frequencies as a function of temperature. Figures 16 and 18 show the variation of the Oberst damping factors, also as a function of temperature. The incompleteness of the plots is because at a given temperature, the amplitude of the given natural frequency component does not reach the 3 dB points of the local peak and is therefore not taken into account by the processing program, as can be seen, for example, in the 600-800 Hz range in Figure 14.

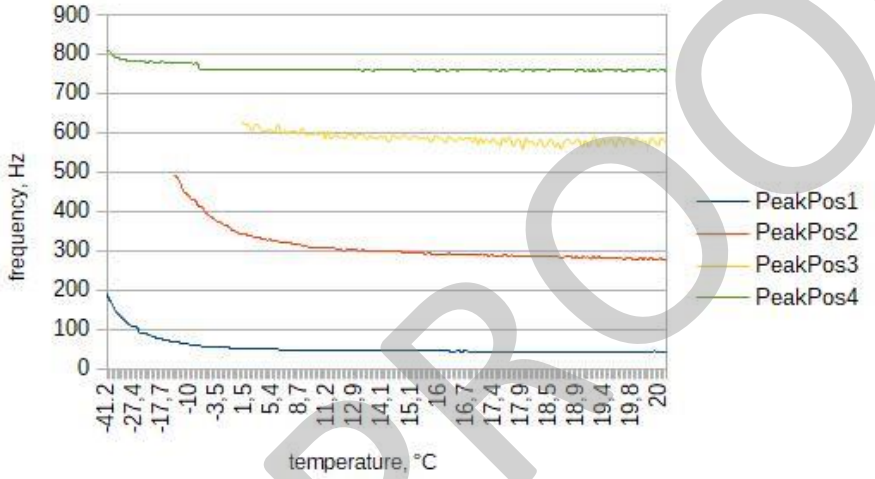


Figure 15

Relationship between temperature and 1st four natural frequencies for a simple test specimen

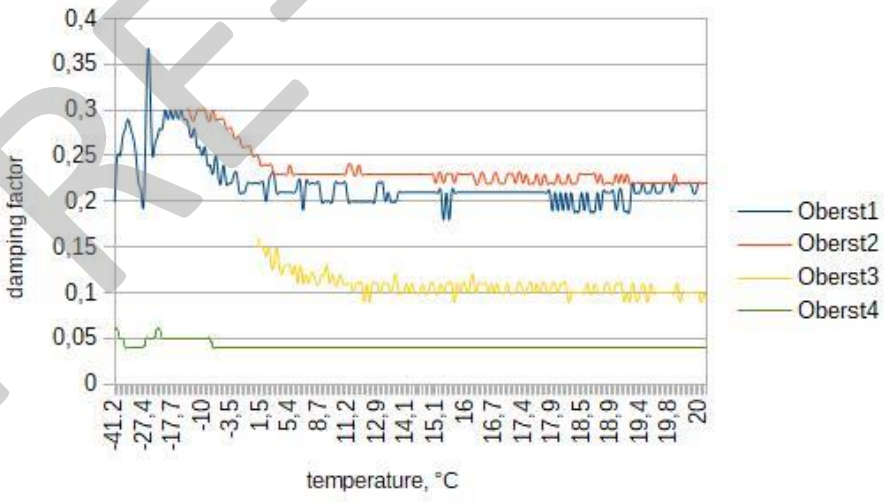


Figure 16

Relationship between temperature and damping factor for a simple test specimen

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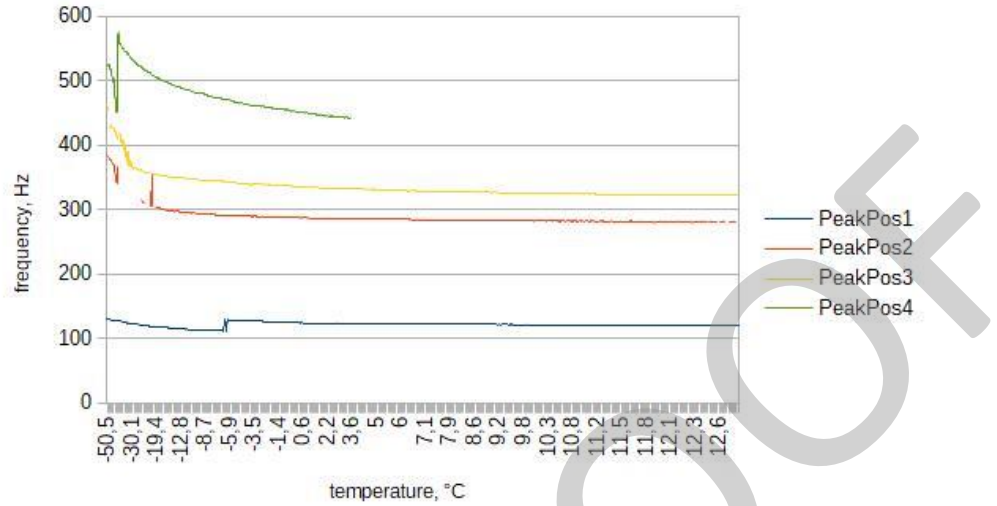


Figure 17
Relationship between temperature and 1st four natural frequencies for the Jones device

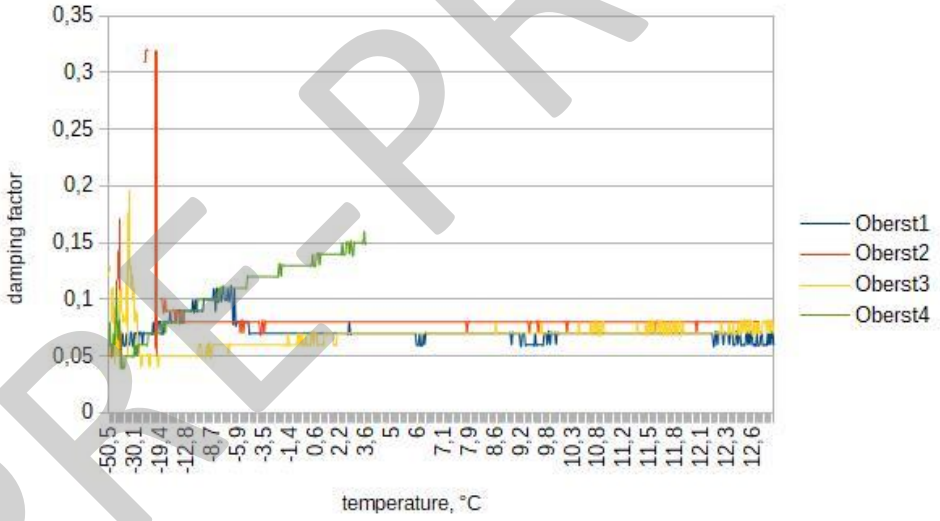


Figure 18
Relationship between temperature and natural frequency for the Jones device

Our faculty also has an MTS Dynamic Materials Test System, which has a climate chamber option that can be cooled to the desired temperature with liquid nitrogen. However, this equipment operates on the hydrodynamic principle, so for dynamic tests, its frequency range is significantly smaller (up to about 10-20 Hz, depending on several factors) than the B&K electrodynamic shaker (4 kHz). Then, the natural frequency values specified by the latter would be required. Therefore,

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replacing the B&K shaker for this task is unsuitable because of its significantly higher operating costs. Instead, it is only suitable to perform some additional tensile tests, which may open another line of research in the future. We plan to extend the research to include temperature-dependent testing of the static and dynamic shear modulus of the same rubber belt on an MTS tensile test system and to carry out all these tests on other materials.

5. CONCLUSIONS

Conveyor belt vibrations are a frequent dynamic phenomenon during the standard operation of conveyor belt systems. The nature of these vibrations plays a crucial factor in ensuring the efficient and uninterrupted functioning of the conveyor belt. Unsuitable vibration conditions can result in system failures. Therefore, analyzing the vibration-damping behavior is essential. This research aims to identify the characteristic behavior of a conveyor belt equipped with textile plies.

The tests show that the conveyor belt's material properties significantly depend on temperature. At low temperatures, not only do the material properties worsen, but so do the lower eigenfrequencies. The more rigid rubber decreases the eigenfrequencies and decreases the damping. When tested at temperatures between -40 … +20 °C, the belt's vibration-damping capacity and natural frequency change significantly. The belt's first natural frequency decreases as the temperature increases from 187 Hz to 38 Hz. The change in natural frequency and the vibration-damping factor are decisive for the application. Static and dynamic tensile tests at different temperatures require further investigation.

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