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# PIPELINE STRESS ANALYSIS UNDER SUPPORTING STRUCTURE VIBRATIONS

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Abstract

The pipelines are subject to various constraints variable in time. Those vibrations may result in both the fatigue damage in the pipeline profile at high stress concentration and the damage to the pipeline supports. If the constraint forces are known, the system response may be determined with high accuracy using analytical or numerical methods. In most cases, it may be difficult to determine the constraint parameters, since the industrial pipeline vibrations are due to the dynamic effects of the medium in the pipeline. In that case, a vibration analysis is a suitable alternative method to determine the stress strain state in the pipeline profile. If a straight pipe section supported on both ends is excited by the bending moments in the support profile, the starting point for the stress analysis are the strains, determined from the Euler–Bernoulli equation. In practice, it is easier to determine the displacement using the experimental methods, since the factors causing vibrations are unknown. The industrial system pipelines, unlike the transfer pipelines, are straight sections at some points only, which makes it more difficult to formulate the equation of motion. In those cases, numerical methods can be used to determine stresses using the kinematic inputs at a known vibration velocity amplitude and frequency. The study presents the method to determine the stresses.

Keywords: pipeline vibrations, natural vibrations, transverse vibrations, resonance frequencies

## ANALIZA STANU NAPRĘŻENIA W DRAJACYCH RUROCIĄGACH PRZY WYMUSZENIU DRGANIAMI KONSTRUKCJI WSPORCZEJ

Streszczenie

Rurociągi podlegają różnym, zmiennym w czasie wymuszeniom. Drgania mogą powodować zarówno uszkodzenia zmęczeniowe rur, jak też uszkodzenia podpór. Jeśli wielkość i charakter wymuszenia oraz sztywności podpór są znane, wówczas odpowiedź układu może być określona z dużą dokładnością metodami analitycznymi lub numerycznymi. W większości przypadków, nie jest jednak łatwym określenie parametrów wymuszenia ponieważ drgania rurociągu przemysłowego są wywołane zarówno dynamicznym oddziaływaniem przepływającego płynu jak też drganiami konstrukcji wsporczej. Wiadomo, że w przypadku odcinka rury podpartego na końcach, jego odpowiedź na wymuszenie w dowolnym przekroju na długości odcinka może być określona przez pomiar parametrów drgań w dwóch różnych przekrojach zlokalizowanych między podporami. Jeżeli prosty, podparty na końcach odcinek rury jest wzbudzany momentami gnącymi działającymi w przekrojach podparcia, to punktem wyjścia dla analizy stanu naprężenia są odkształcenia, które można wyznaczyć z równania Bernoulliego-Eulera. W praktyce prościej jest określić przemieszczenia metodą doświadczalną, bowiem czynniki wzbudzające drgania nie są znane. Rurociągi instalacji przemysłowych, w przeciwieństwie do rurociągów przesyłowych, tylko w pewnych fragmentach są odcinkami prostymi, co utrudnia formułowanie równań ruchu. W tych przypadkach do wyznaczenia naprężeń można stosować metody numeryczne, wprowadzając do obliczeń wymuszenia kinematyczne o znanej wartości amplitudy prędkości drgań i częstotliwości. Taki sposób wyznaczania naprężenia jest przedstawiony w pracy.

Słowa kluczowe: drgania rurociągów, częstości własne, drgania poprzeczne, częstości rezonansowe

## 1. INTRODUCTION

The pipeline vibrations are mainly induced by a pulsating medium flow and transferred into supports and structural components used as the supporting structure of a pipeline. Less often, the vibrations are induced on the supporting structure of a pipeline resulting in pipeline vibrations. The natural frequency of the structure as a resultant of its mass and rigidity is usually low, however it may induce pipeline vibrations at significantly higher natural frequencies. The damage is usually indirect, however long-term vibrations may cause fatigue damage.

The analysis of the effect of vibrations on the stress-strain state in the pipeline profile at the entire length is a complex process. Several attempts have been made in the past to find a simple and efficient method to determine the effects based on the relationship between the vibration velocity and the strain on the pipeline [1-12].

The vibration of pipelines with constant curvature radius at elbow analyzed in frequency domain is described by De Jong [1]. A spectral finite element formulation for the analysis of a stationary vibration of straight fluid-filled pipes is introduced in work [2]. This work also presents element formulations for flanges and rigid masses attached to the pipe. In the spectral finite element formulation, the base functions are frequency-dependent to the local equations of motion. The formulation is valid for arbitrarily long pipes where losses may be distributed in the system and may vary with frequency. Predictions of pipe vibration are made using finite element method with direct methods formulated in frequency domain and with statistical energy analysis. Most of these methods are not yet sufficiently developed to provide a valid result [3]. In work [4] a velocity method for estimating dynamic strain and stress in pipe structures is investigated. With this method, predicted or measured spatial average vibration velocity and theoretically derived strain factors are used to estimate maximum strain at the ends of pipes. The works about slender structures and axial flow interaction are extensively reviewed by Paidoussis [5]. The case of simply supported pipes conveying fluid was developed by Thurman and Mote [6] based on the assumption that the flow is steady and takes into account the nonlinearities associated with the pipe axial elongation. Autors obtained the two nonlinear partial-differential equations that couple the axial and lateral deflections of the pipe. Dupuis and Rousselet [7] developed the nonlinear equation of motion for the in-plane vibration of a circular-arc pipe containing flowing fluid. The forces and moments induced in a pipe element by the flowing fluid were analyzed as a function of the instantaneous local curvature of the pipe. For a fixed-end circular-arc pipe with an arbitrary arc angle, the nonlinear solutions revealed that the dynamic behavior of the system can differ substantially from that predicted by a linear analysis. Analytical and numerical models of pipeline were developed by Lee et al. [8]. The analytical modeling was applied to a simply supported inclined straight pipeline to investigate the stabilities and dynamic responses of the pipeline. It was found that unstable regions in the stability chart expand as the flow velocity and the mass density of the fluid increase. For the damped system, it was found that the original stable system remains stable when the pulsating frequency increases cross the stability boundary and becomes unstable when the pulsating frequency decreases cross the stability boundary. Gorman et al. [9] derived the partial-differential equations of a flexible pipe conveying unsteadily flowing fluid. A combination of the finite-difference method and the method of characteristics was employed to extract displacements, hydrodynamic pressure, and flow velocities from the equations. It was found that with increasing pulsation frequencies, longitudinal vibration tends to be larger. The magnitudes of hydrodynamic pressures were found larger than those obtained without full consideration of the fluid structure interaction. The nonlinear equations of motion of pipes conveying fluid were developed using energy and Newtonian methods by Semler et al. [10]. The lateral vibration of a cantilevered flexible pipe hanging vertically and conveying fluid, which is parametrically excited by the vertical motion at the upper end, was studied by Semler and Païdoussis [11]. The effects of excitation frequency and amplitude on the steady-state response were examined just under below the flow velocity corresponding to the neutral stability of the self-excited vibration of the pipe. As the mean flow velocity approaches the critical value the pipe becomes unstable by flutter through a Hopf bifurcation. A nonlinear model of a straight clamped-clamped pipe conveying fluid was developed by Lee and Chung [12]. From the extended Hamilton’s principle the coupled-nonlinear equations of motion for the longitudinal and transverse displacements were derived. The discretized equations were linearized in the neighbourhood of the equilibrium position, and the natural frequencies were computed from the linearized equations. The transverse vibrations of a supported pipeline with a medium flow were analyzed by Ashley and Haviland [13]. Housner [14] has derived the equations of a motion for the pipeline carrying a liquid medium and the relations defining the fundamental frequencies of the transverse vibrations of a supported pipeline as a function of the medium flow rate.

A new experimental approach for estimating buckling critical velocities from measuring several natural frequencies at relatively small flow rates is presented in work [15].

## 2. RELATION BETWEEN VIBRATION VELOCITY AND STRESS IN PIPE PROFILE

Let us consider a pipe with the flow diameter *A*, length *L*, modulus of elasticity *E* and moment of inertia *I*. The medium density is $\rho$. The pressure inside the pipe at the measured point is *p*. The flow is steady at the velocity **v**.

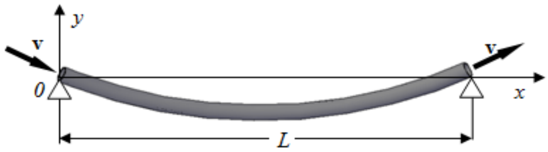



Fig. 1. Sketch of the system considered in the analysis

Dynamical stresses $\sigma_g$ due to the transverse vibrations of a pipe are related to the bending moment $M_g$ and its axial bending strength indicator *W* as follows:

$$\sigma_g = \frac{M_g}{W} \tag{1}$$

For the annular cross-section of the pipe:

$$W = \frac{\pi\left(D^4 - d^4\right)}{32D} = \frac{2I}{D} \tag{2}$$

where $I$ is the moment of inertia corresponding to the bending axis, $D$ and $d$ is outside and inside diameter of pipe. A relation between the bending moment and the pipe strain can be expressed as:

$$\frac{1}{\rho}=\frac{\frac{d^2y}{dx^2}}{\left[1+\left(\frac{dy}{dx}\right)^2\right]^{\frac{3}{2}}}=-\frac{M_g}{EI} \quad (3)$$

For small bend angles the value is $\frac{dy}{dx}<1$ and moreover $\left(\frac{dy}{dx}\right)^2 \ll 1$. Finally:

$$\frac{d^2y}{dx^2}=-\frac{M_g}{EI}=-\frac{2\sigma_g}{ED} \quad (4)$$

The bend line equation for the pipe with length $L$ articulated on both ends and oscillating in accordance with the first form is expressed as:

$$y(t,x)=\varphi(t)sin\left(\frac{\pi x}{L}\right)=\varphi_0\, sin(\omega t)sin\left(\frac{\pi x}{L}\right) \quad (5)$$

where $\omega$ is the natural frequency of the pipe. To calculate a second derivative of the expression:

$$\frac{\partial^2 y}{\partial x^2}=-\frac{\pi^2\varphi_0}{L^2}sin(\omega t)sin\left(\frac{\pi x}{L}\right) \quad (6)$$

Therefore:

$$\sigma_g=-\frac{ED}{2}\frac{\partial^2 y}{\partial x^2}=\frac{ED}{2}\frac{\pi^2\varphi_0}{L^2}sin(\omega t)sin\left(\frac{\pi x}{L}\right) \quad (7)$$

In (5-7), $\varphi_0$ is the amplitude of non-damped vibration of the pipe depending on the constraint value. The profile in which the highest stresses are observed is located in the middle of the pipe length, for $x=0.5L$. Then:

$$\sigma_{g\,max}=\frac{ED}{2}\frac{\pi^2\varphi_0}{L^2}sin(\omega t) \quad (8)$$

Differentiating the expression (5) over time we get:

$$\dot{y}(t,x)=v(t,x)=\omega\varphi_0\, cos(\omega t)sin\left(\frac{\pi x}{L}\right) \quad (10)$$

and:

$$\ddot{y}(t,x)=p(t,x)=-\omega^2\varphi_0\, sin(\omega t)sin\left(\frac{\pi x}{L}\right) \quad (11)$$

As per the equation (11), the maximum vibration velocity can be observed in the same profile as the maximum pipe displacement and also:

$$\sigma_{g\,max}=\frac{ED}{2}\frac{\pi^2\varphi_0}{L^2}sin(\omega t)=-\frac{ED}{2}\frac{\pi^2}{L^2}\frac{p\left(t,\frac{L}{2}\right)}{\omega^2}= =-\frac{ED}{2}\frac{\pi^2}{L^2}\frac{p\left(t,\frac{L}{2}\right)}{4\pi^2f^2}=-\frac{ED}{8L^2}\frac{p\left(t,\frac{L}{2}\right)}{f^2} \quad (12)$$

The analysis applies to the straight sections of pipelines with articulated support (Fig. 1). This condition includes the possibility of arotation of the fixed end of the pipe.

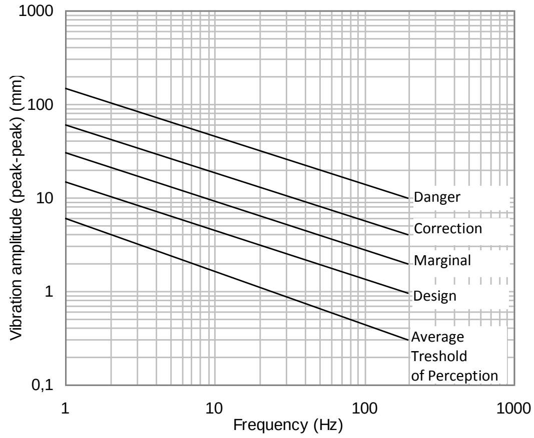



Fig. 2. Allowable pippes vibration levels vs frequency

The literature includes the criteria for the analysis of the vibration levels for pipelines, which define the permissible displacement amplitudes or the vibration velocities as a function of the frequency [16]. Graph (Fig. 2) of the relationship between the vibration amplitude and the frequency shows the permissible vibration levels.

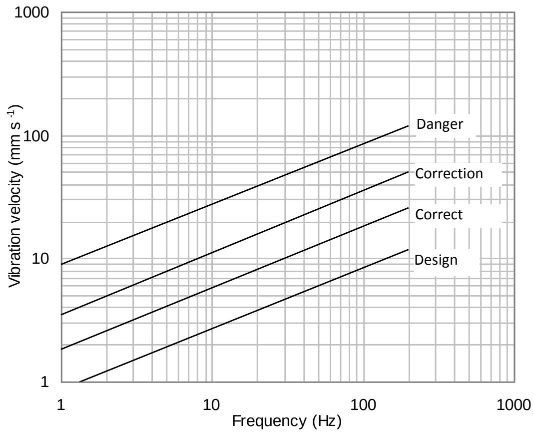



Fig. 3. Orientation values for permissible pipe vibrations

The graph in Fig. 3 shows that the permissible vibration velocity can be higher with the increase in the vibration frequency. The result may be contradictory to the expectations. High frequency vibrations are considered more dangerous.

## 3. INDUSTRIAL PIPELINE TESTING

The subject of the test was a pipeline in the liquor circulation line used for transferring wood chips to the digester. The pipeline length at *33 m* dia. is *323.9 mm*. The medium density is similar to water density. The pipeline is pressurized *9 bar*. The medium flow rate is $4.5\ m\ s^{-1}$.

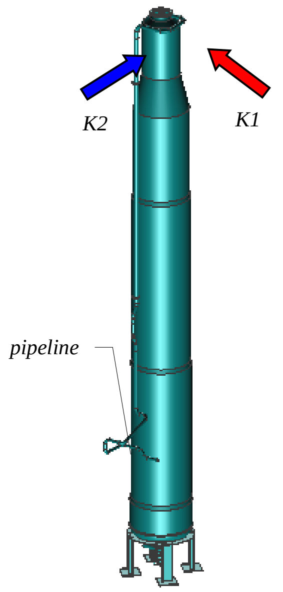



Fig. 4. Digester with a pipeline

The digester is a *1.2·10⁵ kg* vessel mounted on four supports. The digester filled with medium weights twice as much, approx *2.4·10⁵ kg*. The wood chips are fed to the loading chamber via a screw conveyor. The wood chips are fed to the digester from the low pressure zone to the high pressure zone via a high-pressure feeder. Fig. 5 shows the feeder design.

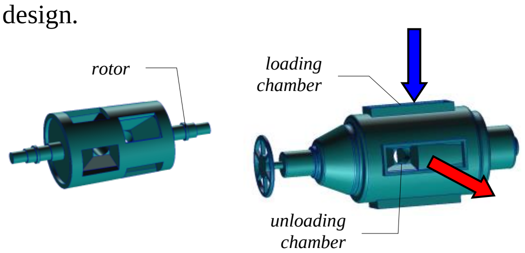



Fig. 5. High-pressure feeder design

The feeder operation is similar to an air lock. The loading chamber filled with wood chips is pressurized *~2 bar*, and the wood chips are carried under pressure *~9 bar* through the pipeline to the digester. The unloading chamber is emptied with a stream of lye forced by a pump. Fig. 6 shows various stages of feeder operation.

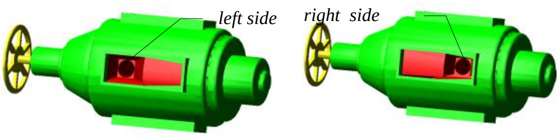



Fig. 6. High-pressure feeder operation

The feeder design, due to the chamber positioning should ensure identical volume flow rate of the lye at each impeller position. The actual flow section is constant, however flow disturbances can be observed due to the changes in the stream direction as a result of opening and closing left and right section of the unloading chamber. With eight feeder chambers, alternately filled and drained during each impeller turn, the excitation frequency should be from *0.67 Hz* (at the feeder speed *5 min⁻¹*) to *2 Hz* (at the feeder speed *15 min⁻¹*).

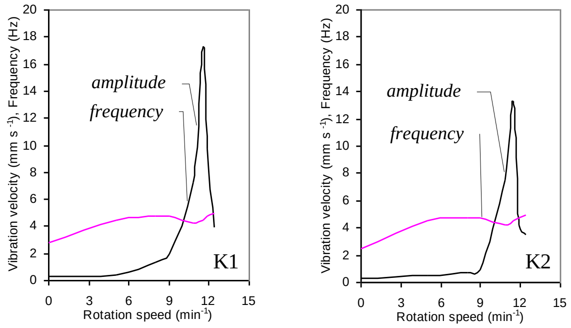
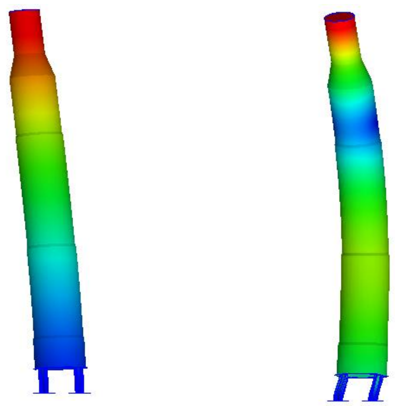



Fig. 7. Amplitude and frequency of the prevalent vibration velocity component in K1 and K2 direction as a function of the feeder speed

The digester vibration tests were initially run at the feeder speed of *0-10.8 min⁻¹*. An increase in the feeder speed has always resulted in an increase in the digester vibration amplitude. It was concluded that the digester vibrations are sub-critical within the entire high-pressure feeder speed range, i.e. at the target output reached at the feeder speed of *12 min⁻¹*. Further tests within the feeder speed range *10.8-12.4 min⁻¹* showed that the resonant excitation was achieved at *11.5 min⁻¹* and the amplitude of the digester vibrations was reduced. It indicates that the resonance is achieved at the excitation of the feeder speed frequency *24xf*.

The resonance curve of the digester vibrations is very steep and the frequency range within which a sudden change in the vibration amplitude is observed is narrow (Fig.7). It is a preferred response, since the transition through the resonance zone is very rapid. The position of the resonance zone shows that the excitation of the digester vibrations is due to the pressure pulsation in the pipeline as a result of opening and closing the feeder chambers.

The modal analysis of the digester vibrations confirms the previous hypothesis that the frequency is in fact the resonant frequency.



2.1 Hz 10.2 Hz

Fig. 8. Forms of the natural vibration of the digester at 2.1 Hz and 10.2 Hz

Fig. 8 shows modal forms for the first and second frequency of the natural vibrations of the digester. The resonant frequency *2.1 Hz* is lower than the experimental frequency *4.5 Hz*. Please remember that the calculations do not allow for the stiffening effect of the pipelines on the digester.

Fig. 9. Shows the nature of the digester vibrations at *33 m* in K1 direction (see Fig. 4) at selected high-pressure feeder speeds. The medium flow in the pipeline with a stationary impeller does not excite any digester vibrations.

To determine the stresses in the pipeline profile exerted by the digester vibrations, a parameterized pipeline configuration model was developed and the natural vibration frequencies for each pipeline branch were determined. The rigidity of the supports used in the model were selected to achieve the natural vibration frequencies consistent with the experimental values.

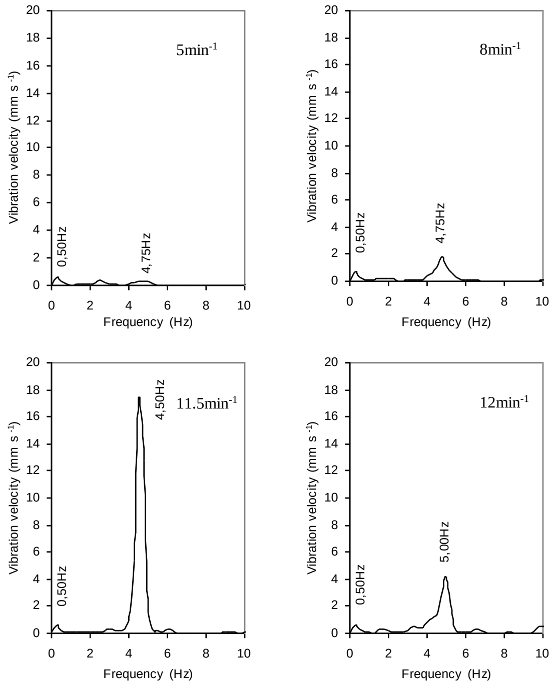



Fig. 9. The vibration velocity spectra of the digester in K1 direction for selected high-pressure feeder speeds

The pipeline length is *l=33 m*. A medium transported in the pipeline is a soda lye (i.e. white liquor) at a concentration of several percent. It is a liquid with the density similar to water.


Fig.10. The upper pipe section mounted to the digester

The pipeline pressure is $p=1.1\ MPa$. A volume flow rate of the medium is $Q=0.07\ m^3 \cdot s^{-1}$.

The article includes an excerpt from the stress strain state analysis of the pipeline in its upper section with the diameter of *d=273 mm*, i.e. by the digester and in its lower section including the section from level *h=0 m* (pump level) to level *h=9 m*. The pipeline section diameter is *d=219 mm*. Fig. 10 shows the upper section as an approach to the digester. Fig. 11 shows a resonant response of the pipe section.

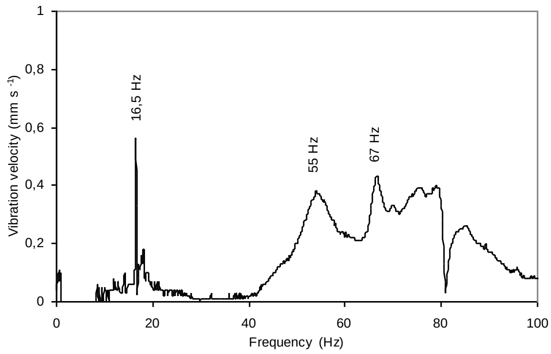



Fig.11. The resonant response of the pipe section

Fig. 12 shows the frequencies and corresponding natural vibration forms determined for the model.

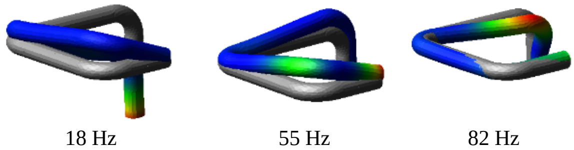



Fig.12. The frequencies and corresponding natural vibration forms determined for the model

A comparison between the calculation results and the measured values shows that the rigidity, damping and weight of the pipe section with medium are correct. The method of excitation of the resonant vibrations in the pipeline shows Fig.13.


Fig. 13. The method of excitation of the resonant vibrations in the pipeline

The vibration velocity of the pipeline was measured at the fixing points at the support and at the straight pipe section. Fig. 14 shows the vibration amplitudes at prevalent frequencies.

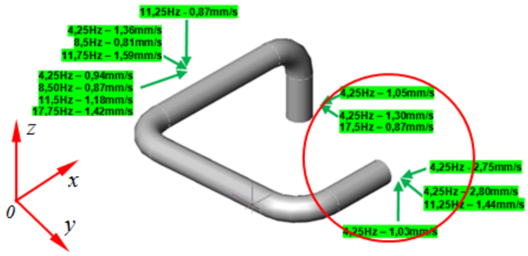



Fig. 14. The vibration amplitudes at prevalent frequencies

Fig. 15 shows the vibration spectra for the analysis of the characteristics of the pipe vibration in time in horizontal and vertical direction in point shown in Fig. 14.

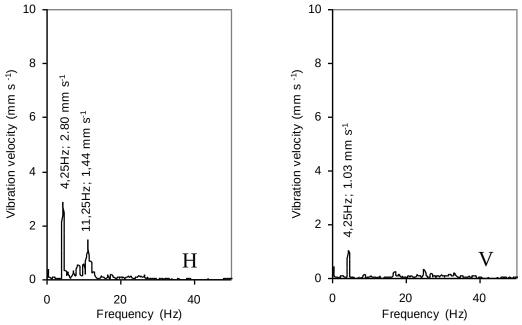



Fig. 15. The vibration spectrums

The constraint used in the numerical analysis corresponds to the actual constraint due to the similarity between the measured and the calculated spectrum of the vibration velocity (Fig. 16).

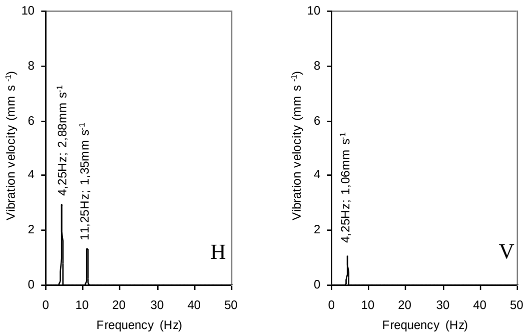



Fig. 16. The calculated spectrum of the vibration velocity

Fig. 17 shows the stress pattern in the pipeline profile determined based on the measured vibration velocities.

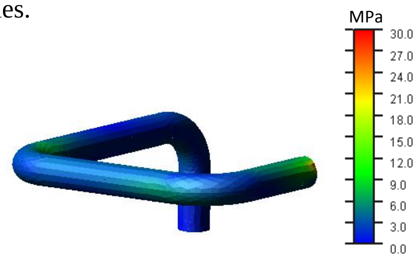



Fig. 17. The stress distribution in the pipe sections

Maximum values of the stress reduced according to the Huber-von Mises-Hencky hypothesis in the analyses pipe section are between $\sigma_{zred}$ =*10-18 MPa*. The values reach $\sigma_{zred}$ =*30 MPa* at the joint between the pipeline and the digester only. The values are considered safe. For comparison, the stress values due to the pressure *p=1.1 MPa* inside the pipe were determined for the pipe profiles (Fig. 18). The stresses reach $\sigma_{zred}$ =*40 MPa* in the most strained profiles. A pipeline with correct pipe section joints should not fail due to the analyzed stress pattern. The pipeline's stress strain state depends on the thermal stresses, which at the incorrectly designed supports without any expansion capabilities may reach several hundred *MPa*, if the pipe section length between the rigid supports is high, and the pipe temperature changes within a wide range.


Fig. 18. The stress distribution in the pipeline for pressure *p=1.1 MPa*

The methods and the vibration modeling results as well as the stress analysis results are presented considering the complex shape of the pipeline profile in its bottom section (Fig. 19).


Fig. 19. The pipeline profile in its bottom section

The natural frequencies of the pipeline section are determined based on the resonant response shown in Fig. 20.

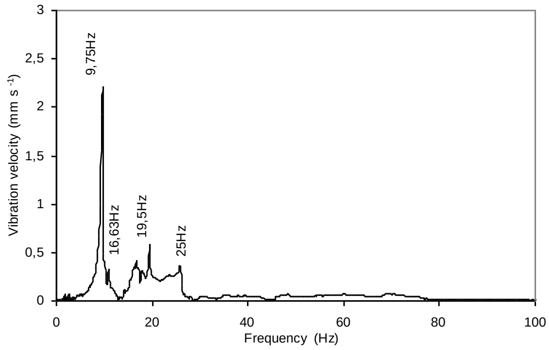



Fig. 20. The resonant response of the pipe section

Within the frequency range up to $f_{gr}$=*100 Hz*, two distinct natural frequencies $f_1$=*9.75 Hz* and $f_2=2xf_1$=*19.5 Hz* can be observed. The following natural frequencies were determined using the modal analysis: $f_1$=*7 Hz*, $f_2$=*8.3 Hz*, $f_3$=*21.8 Hz* (Fig. 21). Allowing for any possible errors when determining the resonant response of the pipeline and difficulties in determining the support rigidity, the results are deemed acceptable.

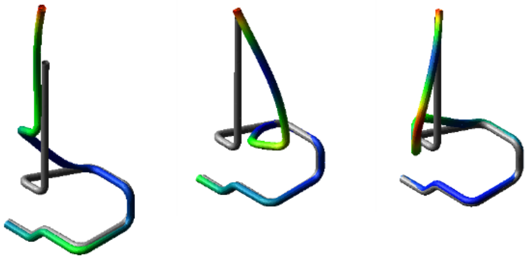



Fig. 21. The frequencies and corresponding natural vibration forms determined for the model

The vibration patterns of 7 Hz, 8.3 Hz, 21.8 Hz correspond to the flexural vibrations.

The vibration velocities at the bottom pipeline section were measured at the joint between the pipeline and the pump and in two other points between the supports (Fig. 22).

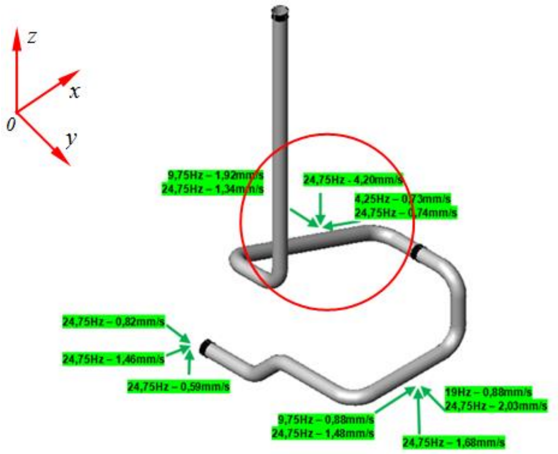



Fig. 22. The vibration amplitudes at prevalent frequencies

The vibration spectra shows a wide range of near-resonant frequencies visible as faint sidebands (L) - left and (R) - right (Fig. 23)

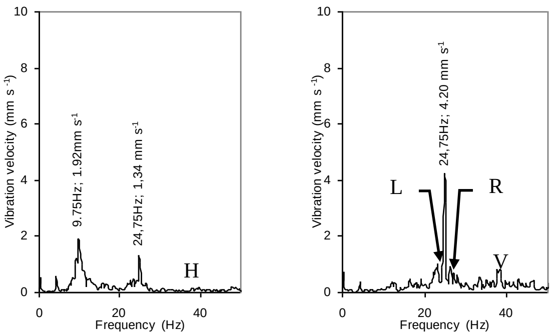



Fig. 23. The vibration spectrums

The kinematic constraint for the oscillating system model are the vibration velocity spectra in the measurement points, filtered by the prevalent frequencies. Fig. 24 shows the example spectrum shape in the *x-y* direction in a specified measured profile.

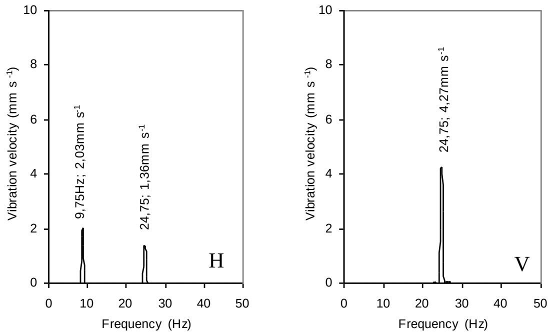



Fig. 24. The calculated spectrums of the vibration velocity
Widma prędkości drgań wyznaczone dla modelu rurociągu

Fig. 25 shows the stress pattern in the profiles of the bottom pipeline section determined using the numerical method.

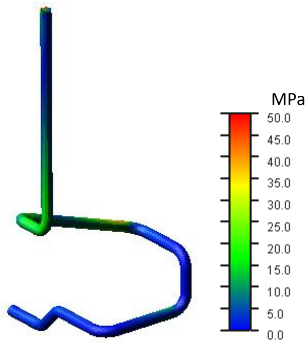



Fig. 25. The stress distribution in the pipeline

The reduced stresses in some profiles may reach *50 MPa*. It is considered a safe level, however, intermittent checks of the welded joints at each section may be required. The complex-shaped pipelines characterized by a change in medium momentum at high temperature and high pressure are subject to cavitation. The degradation in pipe material leads to an increase in stress and thus pipe cracking.

## 4. CONCLUSIONS

The presented method to determine the stresses in pipe structures based on their vibration velocities has a sound theoretical background. It is a tool, which is expected to be in common use due to the simplicity of both the measurement of the vibration parameters and the numerical analysis. It allows to determine the maximum strain and thus the stress strain state of the pipe. The main issue is, that the method requires to determine the parameters of the relative pipeline vibrations, when the pipeline is constrained by the structure characterized by vibrations at other frequencies. Use of the laser vibrometers to determine the time characteristics of the vibrations allows automatic measurements and stress strain state analysis of both short and very long as well as very complex pipelines.

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