THE INSTABILITY OF THE PIPELINE DUE TO TRANSPORTING FLUID'S PRESSURE RIPPLES

Evgeniy V. Shakhmatov, Andrey B. Prokofiev,
Alexander N. Kruchkov and Georgy M. Makaryants
Faculty of Aircraft engine, Samara State Aerospace University, Moskovskoe shosse St., 34
443086 Samara, Russia
shakhm@ssau.ru; prok@ssau.ru, kan@ssau.ru, mak-georgy@yandex.ru

Abstract

In the paper numerical solution schemes have been developed to analyze the pipeline vibration under acoustic excitation due to pressure ripples in transporting fluid. To predict high-frequency pipeline response the three-dimensional model is created.

1 Introduction

Operating hydraulic systems of various technical objects features the presence of dynamic loads, which are due to the discontinuous manner of operation of pump units, for instance. Pulsation processes in fluids determine force excitation which acts on pipeline system components, causing them to vibrate. Pipeline mechanical oscillations result in the reduction of the systems` reliability and efficiency, their breakdowns and depressurization. Thus, the description and investigation of the interaction between the pulsating fluid flow and the pipeline wall is a serious task with the great practical value.

There are analytical models describing the vibration of hydraulic systems` pipelines under force excitation by working fluid pressure pulsations [1 - 9, 13]. Let us consider the principal simplifications and assumptions accepted when creating the models:

- Simplification (idealization) of the pipeline shape – as a rule, behavior of rectilinear sections is investigated.
- Simplification of boundary conditions of both hydraulic and mechanical subsystems. In most cases either fixed blocks or hinges are considered as pipeline supports. Hydraulic subsystem boundary conditions which should be preset as impedances of joined circuits for established oscillations are not taken into account at all or idealized as cases of an acoustically closed or acoustically opened end.
- Parameter distribution and wave properties of the working fluid of hydrosystems are not taken into account.
- Resonance properties of the pipeline mechanical subsystem are not taken into consideration.

In [5] pipeline vibration initiation is explained by gyroscopic moments in elastic structure components of the mechanical subsystem. Gyroscopic moment is caused by Coriolis forces which are generated when the system is excited. The author shows that the pipeline has complex forms of oscillations (i. e. different points of the axis oscillate with phases shifting). The problem on pipeline oscillations with fluid flowing inside it is regarded as a non-linear spectral problem; ways of solving it are proposed. The disadvantages of the model created are as follows: only pipelines of rectilinear shape are considered; boundary conditions of differential equations describing motion are given only for the mechanical subsystem. Besides, the model does not provide a solution for the level of pipeline vibration under external force or kinematic excitation in a general form.

In [4] the model of Y-shaped pipeline oscillations is presented, the pipeline being rigidly fixed at the ends. The model is described by a system of equations of balance of elastic forces, centrifugal forces, Coriolis forces and inertia forces. The pipeline system is divided into three rectilinear sections. At the point of joining of these sections additional boundary conditions are specified.

In [6] the most complete model is given, which describes vibroacoustic processes in a pipeline system. The model allows considering mutual effect of pulsation and vibration processes. The disadvantage of the model is its being very cumbersome, especially for three-dimensional cases. Thus, for describing the motion of a pipeline with fluid in it in a plane problem a system of 12 equations is used, 9 of which are differential equations. Analytical solution this system of equations in a general is not possible. The authors admit this form disadvantage and suggest solving the system obtained by the finite-difference method.

2 Mathematical models

One efficient way to solve the problem under consideration is to use the finite-element method. The authors developed a procedure for calculating vibroacoustics characteristics of pipelines having arbitrary spatial configuration using the ANSYS software. In this paper the procedure is described, an elbowed pipeline being used as an example. Figure 1 shows geometrical characteristics of the pipeline taken as the object of the investigation.

The pipeline is made of stainless steel (0.12% carbon, 18% chrome, 9% nickel). The working fluid used was aviation hydraulic oil (density ρ=870 kg/m^3, sound velocity in the medium c=1280 m/sec, bulk modulus of the medium elasticity E=1450 MPa). A reasonably simple spatial configuration of the pipeline was chosen because, on the one hand, the existing analytical models do not make it possible to calculate its vibration parameters in the case of force excitation by working fluid pulsations, and, on the other hand, the developed simulation procedure can be fully illustrated by the example under review.

The first stage of investigation is the vibroacoustics analysis of pipeline based on simplified finite-element models. Using the ANSYS software a three-dimensional beam model of a pipeline on the basis of BEAM188 element was created. The element characteristics were set by the appropriate values of parameters Real Constant and Material Properties:

- cross section (form and geometrical dimensions);
- density, Young's modulus, modulus of elasticity in shear of the pipeline material;
- linear mass of the added distributed load (fluid mass properties taken into account).

Since provision was made for the experimental checking of simulation results, mass element MASS21 was introduced into the beam model to take into account the added mass of the piezoaccelerometer. Viscous and elastic properties of pipeline supports were determined experimentally. They were specified by the MATRIX27 element in the model.

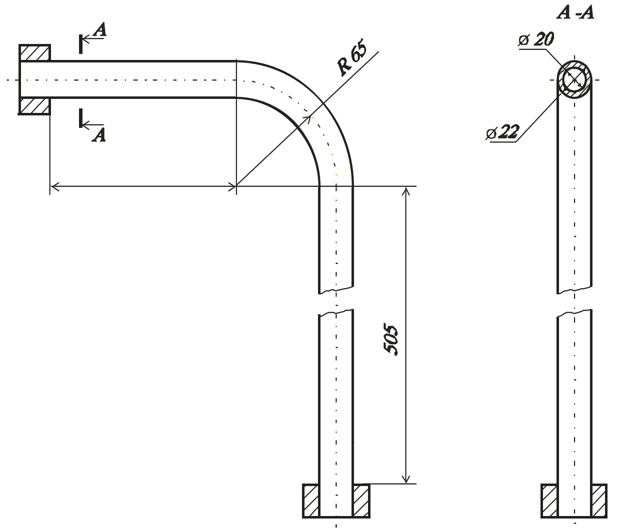

Fig. 1 Geometrical characteristics of the pipeline under investigation

The next stage of investigations was the construction of a more detailed three-dimensional model of the pipeline under consideration. An eight-node spatial element FLUID30 was used to simulate fluid. The mentioned element is intended to describe acoustic properties of fluid, as well as its dynamic interaction with an elastic structure. The nodes of the element have four degrees of freedom: displacements along the x, y, z coordinates and pressure. Mathematical description of this type of element is based on the discretization of a wave equation

$$\frac{1}{c^2}\cdot\frac{\partial^2 P}{\partial t^2}-\nabla^2 P=0\,, \qquad (1)$$

were P is acoustical pressure, describing the pressure ripples in working fluid; t is the time variable; $\nabla=\{L\}^T=\left(\frac{\partial}{\partial x}\frac{\partial}{\partial y}\frac{\partial}{\partial z}\right)$ differential operator.

According to finite element method the wave equation can be written in matrix form:

$$\begin{gathered}\left[M_e^P\right]\{\ddot{P}_e\}+\left[K_e^P\right]\{P_e\}+\\+\rho_o[R_e]^T\{\ddot{U}_e\}=\{0\}\end{gathered}\,, \qquad (2)$$

were $\left[M_e^P\right]=\frac{1}{c^2}\int\limits_{vol}\{N\}\{N\}^T d(vol)$ is mass matrix of the fluid; $\left[K_e^P\right]=\int\limits_{vol}[B]^T[B]d(vol)$ is stiffness matrix of the fluid; $\rho_o[R_e]=\rho_o\int\limits_S\{N\}\{n\}^T\{N'\}^T d(S)$ is stiffness matrix of vibroacoustics interaction; $\{P_e\}$ is a vector of nodal pressures; $\{U_e\}$ is a vector of nodal displacements; $\{N\}$ is pressure element shape function; $\{N'\}$ – is movement element shape function; $[B]=\{L\}\{N\}^T$; $\{n\}$ is a vector normal to surface S.

Boundary layer elements and elements which are under action of a load (pressure ripples and acoustic impedance) have a property of interaction with the structure. “Internal” fluid elements do not have that property. Interaction between fluid and pipe wall is modeled by FSI operation. Mathematical description of this operation is presented by the follow equation:

$$\begin{gathered}\begin{bmatrix}[M_e] & [0]\\ \left[M^{fs}\right] & \left[M_e^P\right]\end{bmatrix}\begin{Bmatrix}\{\ddot{U}_e\}\\ \{\ddot{P}_e\}\end{Bmatrix}+\\+\begin{bmatrix}[C_e] & [0]\\ [0] & \left[C_e^P\right]\end{bmatrix}\begin{Bmatrix}\{\dot{U}_e\}\\ \{\dot{P}_e\}\end{Bmatrix}+\\+\begin{bmatrix}[K_e] & \left[K^{fs}\right]\\ [0] & \left[K_e^P\right]\end{bmatrix}\begin{Bmatrix}\{U_e\}\\ \{P_e\}\end{Bmatrix}=\begin{Bmatrix}\{F_e\}\\ \{0\}\end{Bmatrix},\end{gathered} \qquad (3)$$

were $[M_e]$ is mass matrix of the structure; $\left[M^{fs}\right]=\rho_o[R_e]^T$; $[U_e]$ is a vector of nodal displacements of the structure; $[C_e]$ is damping matrix of the structure; $\left[K^e\right]$ is stiffness matrix of the structure; $\left[K^{fs}\right]=-[R_e]$; $\{F_e\}$ is a vector of external forces.

Acoustical load (as a sound absorption coefficient) is specified by IMPD operation. This is achieved by adding $\left[C_e^P\right]\{\dot{P}_e\}$ expression to the left part of matrix form of the wave equation. $\left[C_e^P\right]=\frac{\alpha}{c}\int\limits_S\{N\}\{N\}^T d(S)$ is damping matrix of the fluid, that specify dissipation conditions on the boundary of interaction.

As the input data the following properties are specified: density, sound velocity and sound absorption coefficient.

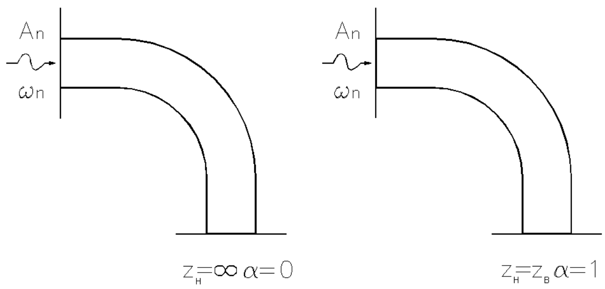

Fig. 2 Boundary conditions of the system

Traditionally, input impedance is used as a dynamic load in the calculations of pulsation characteristics [4]. In ANSYS acoustic load can only be set by the coefficient of absorption feature α. Impedance and absorption coefficient are related as follows [12]:

$$Z_H=\left(x'\cdot\frac{\rho c}{s}\right)+\left(y'\cdot\frac{\rho c}{s}\right)\cdot j\,, \qquad (4)$$

were ρ is density of working fluid; c – sound velocity in working fluid; s is cross-section of pipeline; $j=\sqrt{-1}$;

$$x'=\frac{2N}{\left(N^2+1\right)-\left(N^2-1\right)\cdot\cos\delta};$$

$$y'=\frac{\left(N^2-1\right)\cdot\sin\delta}{\left(N^2+1\right)-\left(N^2-1\right)\cdot\cos 2\delta};$$

$N = \dfrac{p_{max}}{p_{min}} = \dfrac{1+\sqrt{1-\alpha}}{1-\sqrt{1-\alpha}}$ is standing wave ratio;

$\delta = \left(x_0 - \dfrac{\lambda}{4}\right) \cdot \dfrac{2\pi}{\lambda}$ is phase of the reflection coefficient;

α – sound absorption coefficient;

$\boldsymbol{x_0}$ – nearest to the end of the pipe coordinate of pressure minimum.

Thereby, from the known impedance, sound absorption coefficient can be calculated. For the case of traveling wave the input impedance is equal to the wave resistance of the pipeline $Z=Z_в$. At the same time amplitude of the pressure wave is constant along the pipeline (in the case when the friction force negligible). So $x_o=0$, and $\alpha=1$. If in pipe standing wave exists, than $Z=\infty$. It's known [12], that for standing wave, minimum of pressure pulsation locates at cross-section removed at acoustically closed end of one quarter of wave length. That is $x_o=\lambda/4$. This case is realized in ANSYS software by default.

An eight-node spatial element SOLID45 was used to model the pipeline wall. This element is used for three-dimensional modelling of strained solid bodies. Its nodes have three degrees of freedom: displacements along the *x*, *y*, *z* axes. Mathematical description of the element is based on the discretization of a strained solid body dynamics equation [10].

$$[M]\{\ddot{U}\}+[C]\{\dot{U}\}+[K]\{U\}=\{F^a\}, \qquad (5)$$

$[M]$ is mass matrix of the structure;

$[C]$ is damping matrix of the structure;

$[K]$ is stiffness matrix of the structure;

$\{F^a\}$ is vector of external load.

As in the case of beam approximation, viscous and elastic properties of supports were modelled with the help of the MATRIX27 element.

The geometry of a one-branch pipeline admits of a regular grid. Since the finite elements used in the process contain quadrilateral edges the areas meshed by them are to meet this requirement as well. That is why the cross section being modeled is divided into tetrahedral areas, as shown in figure 3.

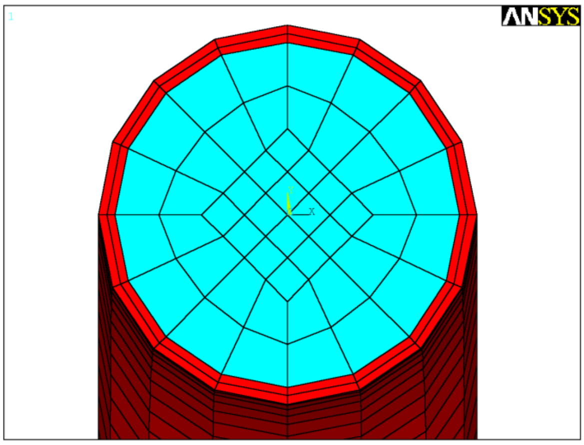

Fig. 3 Finite-element grid of the solid model

Dynamic interaction between the fluid and elastic pipeline wall is simulated with the help of the procedure "Fluid structure interaction". A mathematical model of that includes the joint solution of a wave equation and an equation describing the dynamics of a pipeline wall [11]. Further, the type of matrices of acoustic elements in pipeline ends is changed to asymmetrical. This operation is performed to specify the boundary conditions of the hydraulic subsystem – impedances at the pipeline outlet. The final stage of modeling is the presetting of hydrodynamic load as amplitude and frequency of pressure pulsations in the inlet to the system (for solving problems of pipeline force excitation by working fluid oscillations).

2 Experimental set-up

A validating of simulation was performed using set-up shown schematically in figure 4.

An experimental set-up may be divided into: a block of working fluid preparation; an oscillation block with acoustics decouples; a block of the test object with a system of acoustics loads.

The block of working fluid preparation is equipped with: a fuel tank (volume – 80 l) for storing working fluid; a centrifugal pump (capacity – 80 l/min; outlet pressure 0.45 MPa), necessary to prevent cavitation in the main pump; filters providing reliable fluid cleaning; water-oil heat exchanger maintaining the working fluid temperature within the range from 15° to 50°C; a plunger high-pressure pump motor-operated via reducer (supplying working fluid pressure up to 20 MPa); a system of pipelines and armature.

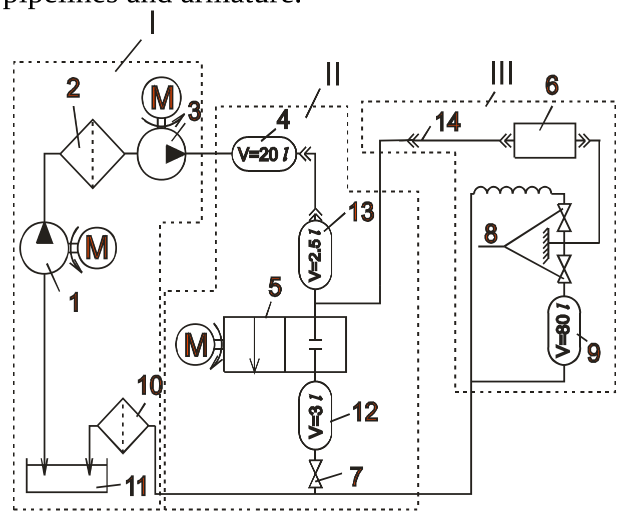

Fig. 4 Experimental set-up for investigating vibroacoustic characteristics of pipeline system:
I – block of working fluid preparation; II – oscillation block with acoustic decouplers; III – test object block with a system of acoustic loads;
1 – booster pump with a drive; 2, 10 – filters; 3 – high pressure pump with a drive; 4, 9, 12, 13 – decoupling acoustic capacitors; 5 – oscillator with a drive; 6 – pipeline system under investigation; 7, 8 – valves; 11 – tank; 14 – metal hose.

The oscillation block contains: an electrically operated siren-type oscillator; a system of tanks and pipelines, providing acoustic decoupling.

The block of test object contains a set of supports for fixing the sections of the pipeline system that are being investigated. To simulate various acoustic conditions in the outlet of the testing pipeline there are an "infinitely long" pipeline, a tank and valves. Since in this work provision was made for loading the acoustically closed end, the outlet section of the pipeline under investigation was muffled and the system of dynamic loads was disjoined. Kinematic decoupling of the pipeline from the pulsator was performed using a metal hose.

Set-up equipment makes it possible to investigate pipeline vibroacoustic characteristics in the following ranges: the frequency of the fundamental tone of working fluid oscillations – from 30 to 800 Hz; static pressure – up to 20 MPa; pressure pulsations' amplitude – up to 1 MPa.

For numerical treatment of experimentally obtained time realizations of dynamic processes the following algorithm was used:

- removal of the signal dc component;
- fast Fourier transform (the number of read-outs analysed - 2^{12});
- picking out the most important spectrum constituents (pulsator harmonics). The number of constituents analised did not exceed 3 or 4, those with the highest amplitude. This procedures were performed both for the vibroacceleration channel, and for the channel of pressure pulsations.

While investigating pipeline vibroacoustic characteristics much importance was attached to providing the form of signal time realization close to sinusoidal in terms of pressure. The pulsator windows of siren type were specially programmed and a system of decoupling tanks was provided in order to do that (see figure 4). Pulsator non-linearity coefficient defined as the ratio of pressure pulsation amplitude at the fundamental tone frequency to the mean square value of upper ultraharmonic constituents changed with varying frequencies in a fairly wide range from 2.5 to 41.5. It should be noted that if the value of the coefficient is less than 3, the signal is to be regarded as a polyharmonic one in terms of pressure. If the non-linearity coefficient is of the order of 10 or more the signal form is practically indistinguishable from the sinusoidal one and is monoharmonic. Nevertheless, for a number of investigated fundamental tone pulsator frequencies the signal was polyharmonic. In these cases it is easy to pick out amplitude peaks corresponding to pulsator harmonics in frequency in the vibroacceleration spectrum (see figure 5).

However, the relationship of the amplitudes of these harmonics is different and is defined by:

- the relationship of harmonic amplitudes in the pressure pulsations spectrum;
- agreement or disagreement of the harmonic under consideration with some natural frequency of the pipeline mechanical subsystem;

- harmonic frequency (at lower frequencies it is rather difficult to expect obtaining higher magnitudes of vibroacceleration);
- oscillation form at a given frequency.

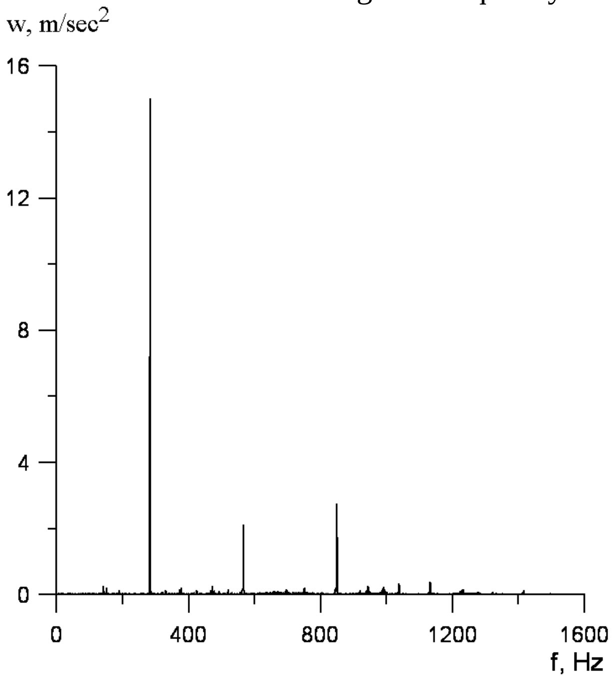

Fig. 5 Experimental vibroacceleration spectrum

The results of calculating pipeline natural frequencies and forms using the beam and three-dimensional models are presented in Table 1. The results of experimentally determining the natural frequencies of oscillations are also given there.

The values of mechanical subsystem natural frequencies experimentally obtained coincide with the results obtained using the beam model (the difference does not exceed 15 Hz). Experimental investigations of natural oscillation forms were conducted. Comparison of the experimental data and the results obtained using the beam model is presented in figure 6. Pipeline natural oscillations take place both in its plane (see figure 6,a and 6,c – 252 and 590 Hz respectively), and in the direction normal to the plane (see figure 6,b – 352 Hz). Figure 6,b shows oscillation forms in 3 plan views. Analysis of figure 6 demonstrates both qualitative and quantitative agreement between the results of modelling and experiment as to the number and location of natural form nodes and maximum.

3 Results and discussion

Table 1 The results of calculating natural frequencies and forms of oscillations of the pipeline

Experiment, ***Hz***	Natural frequency for the model on the basis of ***Beam188*** element, ***Hz***	Natural frequency for the model on the basis of ***Solid45*** and ***Fluid30*** elements, ***Hz***
175	175	175
252	265	264
352	354	363
Hydraulic resonance		416
590	594	644

In the process of simulation one of the boundary conditions was the impedance at the pipeline outlet section equal to infinity (or sound absorption coefficient equal to zero) which corresponds to the load on the acoustically closed end. It should be noted that the fourth form in table 1 corresponds to the first form of oscillations (1/4 – to wave resonance) of the pipeline hydraulic subsystem. The analytical expression for the natural frequencies of the pipeline hydraulic subsystem in case of loading the acoustically closed end can be written as [12]:

$$f_n = \frac{c'(2n-1)}{4l_p}, \qquad (6)$$

where n - is the number of natural frequency; l_p - total length of the pipeline; c' - sound velocity in the working fluid with regard to pipeline wall pliancy:

$$c' = \frac{c_0}{\sqrt{1 + \frac{2r_0 E}{\delta_w E_w}}}, \qquad (7)$$

where r_0 - is the pipeline inner radius; δ_w - pipeline wall thickness; E_w - modulus of elasticity of the pipeline wall material (E_w=2 10^{11} Pa).

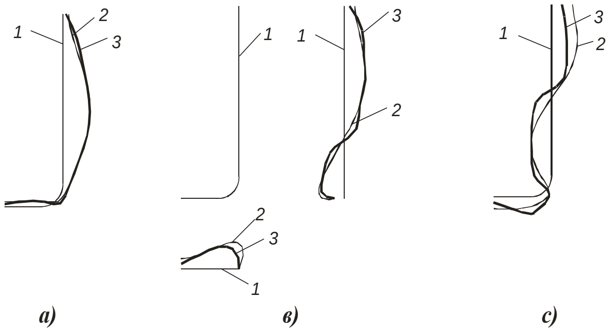

Fig. 6 Natural forms of oscillations of the pipeline under investigation:
a) second form, f = 252 Hz; b) third form, f = 352 Hz;
c) fourth form, f = 590 Hz;
1 – neutral line; 2 – calculated form;
3 – experimental form

For the case under consideration c' =1210 m/sec. Substitution in the relation (1) of numerical values of the parameters of the pipeline being investigated (c' =1210 m/sec (2), l_p =0,727 m) gives the following set of hydraulic subsystem natural frequencies: 416, 1248, 2081 Hz etc. It can be seen from these data that the values of the first natural frequency of the hydraulic subsystem defined analytically and by the finite-element model coincide. For the mechanical subsystem the difference between the results of calculating by the three-dimensional solid model and those obtained experimentally increases with the number of natural form of oscillations, which is caused by the quality of the finite-element grid used and by the errors of the numerical method of modeling.

The results of processing experimental data as frequency dependence of relative vibroacceleration ν are presented in figure 7 which also shows a calculated curve of relative vibroacceleration obtained using the finite element method and the above algorithm. The graph shows good agreement between the design and experimental data, both in terms of quantity and quality (the adequacy was defined with the help of Fisher's variance ratio). In the graphs in figure 7 one can distinguish 3 resonance areas. The most intensive resonance (f=265 Hz) corresponds to the natural frequency of the second form of oscillations of the pipeline being considered. Second form oscillations take place in the plane of the pipeline. The remaining 2 resonance areas correspond to the first and third natural forms when oscillations take place in the direction normal to the plane of the pipeline. It should be noted here that a linear model was used in calculations, which does not take account of friction losses in the supports and pipeline material. This explains the fact that the experimentally obtained values of relative vibroacceleration are smaller than the calculated values in the 250 Hz resonance area.

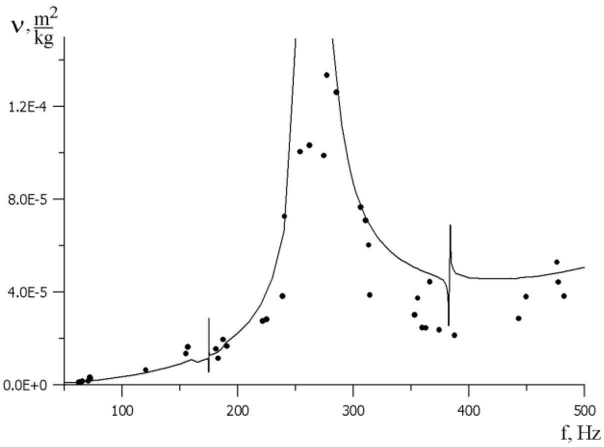

Fig. 7 Amplitude- frequency characteristic of relative vibroacceleration υ of the considered section of the pipeline being investigated

– simulation results; ● – experimental data

4 Conclusions and future work

A procedure of describing processes of vibroacoustic interaction in a pipeline system using the ANSYS software is proposed. The task of simulating vibroacoustic characteristics of an elbowed pipeline under force excitation by working fluid pressure pulsations specified at the inlet section is considered. Experimental investigations showed good agreement between the results of numerical modelling and tests in the frequency range up to 500 Hz. Trustworthy simulation results are likely to be obtained in the frequency range specified and for other pipelines of an arbitrary complex spatial configuration. This paper offers an opportunity to solve the tasks of reducing vibroacoustics loads in pipeline systems with pulsating fluid due to the elimination of resonant modes of operation by choosing the optimal pipeline configuration on the basis of the simulation methods developed.

5 Acknowledgement

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