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CRITICAL FLOW VELOCITY IN A PIPE WITH ELECTROMAGNETIC ACTUATORS

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We study the influence of electromagnetic damping on the dynamics of a pipe conveying fluid. Pipes supported at both ends as well as cantilever ones (both discharging and aspirating the fluid) are considered. We assume physical parameters of the systems which allow an experimental verification of results. We develop simple methods of calculation of the internal and external damping coefficients which are based on known models, and do not require experiments. The governing partial equation of the pipe is discretised with Galerkin's procedure, and the stability of the resultant dynamical system is determined with eigenvalues of its linearization. The actuators destabilise the pipe supported at both ends, but can remarkably improve stability of cantilever ones. The effect of magnetic damping strongly depends on the position at which actuators are attached to the pipe.

Key words: pipe conveying fluid, dynamic stability, flutter, electromagnetic actuators

1. Introduction

Pipes conveying fluid are an example of nonconservative elastic systems with follower force. The problem of their dynamics has become a new paradigm in mechanics; moreover, theoretical predictions can be verified experimentally (Païdoussis, 2008). Because pipes conveying fluid are widely used – in water home installations, cooling systems, pipelines, ocean mining – therefore the results might have practical applications.

Sufficiently high flow velocity results in the buckling of a pipe supported at both ends – simply supported (Feodos'ev, 1951; Housner, 1952) and clamped-clamped one (Païdoussis and Issid, 1974). The external and internal damping of the system depends on the velocity of deflection of a pipe, thus it has no influence on divergence. On the other hand, a cantilever pipe is always prone to flutter instability (Gregory and Païdoussis, 1966a). Then, depending on the mass ratio of the fluid and pipe, both types of energy dissipation can stabilise the system as well as decrease the critical flow velocity (Gregory and Païdoussis, 1966b).

The elastic Winkler foundation stabilises a non-damped pipe with a flow (Lottati and Kornecki, 1986). Partial foundation can increase the critical flow as well as destabilise the pipe, and also change the type of instability between divergent and oscillatory one (Impollonia and Elishakoff, 2000), (Elishakoff and Impollonia, 2001). Moreover, a significant effect on the critical flow is exerted by an intermediate elastic support (Sugiyama *et al.*, 1985), a simple support (Edelstein and Chen, 1985) and also by a system of several springs or a torsion spring (Païdoussis, 1998).

Thus an application of electromagnetic actuators, which generate both elastic and dissipative nonlinear forces, can remarkably affect the stability. The application of such elements for stabilisation of rotating shafts was proposed by Kurnik (1994). It was shown that the magnetic force passively generated by actuators can bring an increase of the critical rotational velocity. The efficiency of magnetic stabilisation was also confirmed in systems of columns subjected to follower load (Kurnik *et al.*, 2009; Szmidt and Przybyłowicz, 2012).

The influence of magnetic forces on nonlinear dynamics of similar structure – a cantilever beam – was investigated both theoretically and empirically (Moon, 1980; Holmes and Moon, 1983). In 1988 Tang and Dowell studied chaotic motion of a cantilevered pipe subjected to external magnetic force, approximated by a cubic function of pipe deflection. Here we confine ourselves to analysing just the stability of the system. On the other hand we apply electromagnets (not magnets), and the magnetic force exerted on the pipe is the effect of the actuators state, which we model in detail. We incorporate realistic magnetisation curve of steel cores and take into account both the effect of magnetic induction and hysteresis. Moreover, we consider aspirating pipes, which dynamics was not understood in the eighties.

Galerkin's multimodal discretisation based on the beam eigenfunctions is applied to the partial differential equation governing the lateral motion of the pipe. The resultant ordinary equations are coupled with ordinary equations of electro-magnetodynamics of actuators. Then, the stability is determined by numerical calculations of eigenvalues of the linear system.

2. Analysed system

Consider a fluid-conveying pipe with electromagnetic actuators attached to it (Fig. 1). The pipe is placed in the direction of the gravitational field g and conveys the stream of water with velocity V. The support is cantilever, simple at both ends or clamped-clamped. The water is pumped downwards or aspirated.

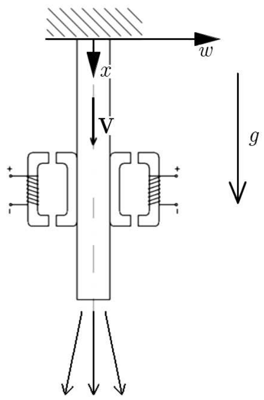

Fig. 1. Pipe with a flow and electromagnetic actuators

Assume the motion of the pipe in the plane of symmetry $x-w$, because for studied systems a two-dimensional model is sufficient for predicting instability (Modarres-Sadeghi *et al.*, 2008). The dimensions of the pipe do not change due to the internal fluid pressure or the flow friction. Lateral deflections are small and the pipe is slender, thus the linear Bernoulli-Euler model is acceptable. The pipe is made of a viscoelastic Kelvin-Voigt material. The vibrations frequency is sufficiently low, so the magnetic induction and its rate of change are approximately constant on the cross-section of the electromagnet core. Therefore, Bertotti's model of magnetic hysteresis is applicable (Przybyłowicz and Szmidt, 2009). We consider a fully-developed turbulent flow, which can be approximated by the so called *plug flow* – an infinite elastic rod moving inside the pipe.

In cases of the pipe supported at both ends and the cantilever pipe conveying fluid towards the free end, the dynamics is governed by the following linear partial equation (Païdoussis and Issid, 1974) coupled with nonlinear equations of electro-magnetodynamics of actuators (Przybyłowicz and Szmidt, 2009)

$$
\begin{gathered}
\Big(1+\beta^{\star}\frac{\partial}{\partial t}\Big)EI\frac{\partial^4 w}{\partial x^4}+[MV^2-(m+M)g(L-x)]\frac{\partial^2 w}{\partial x^2}+2MV\frac{\partial^2 w}{\partial x\partial t}\\
+(m+M)g\frac{\partial w}{\partial x}+\gamma^{\star}\frac{\partial w}{\partial t}+(m+M+M_a)\frac{\partial^2 w}{\partial t^2}=\frac{A}{\mu_0}(B_2^2-B_1^2)\delta_e\\
\Big(\frac{AN^2}{R}+\frac{1}{8}\sigma l a^2\Big)\frac{dB_{1,2}}{dt}+\frac{2(z\pm w(x_e,t))}{\mu_0}B_{1,2}+l\phi^{-1}(B_{1,2})=\frac{NU}{R}
\end{gathered}
\tag{2.1}
$$

In the equations above, L and EI denote the length and flexural rigidity of the pipe, respectively, m, M, M_a – mass of the pipe, water flowing inside and fluid which surrounds the pipe and is accelerated by its motion, per unit length, $\beta^{\star}$, $\gamma^{\star}$ – coefficients of internal and external damping, l – length of each of the magnetic circuits, a, $A=\pi a^2$ – cross-sectional radius and area of the electromagnet core, N – number of wire coils which are wound on the cores, R – resistance of the electric circuit (copper), x_e – distance from the electromagnets to the upper end of the pipe, z – gap in the magnetic circuits (i.e. between the cores attached to the column and electromagnets) in the middle equilibrium of the column, U – voltage applied to the actuators, σ – electric conductivity of the steel cores, $B=\phi(H)=\arctan(H/400)T$ – primary magnetisation curve of steel, μ_0 – magnetic permeability of vacuum.

The "+" sign at the term expressing the Coriolis effect corresponds to the situation when the fluid moves downwards. In the case of the aspirating pipe, we change the sign to "–", so the flow velocity value V is always positive.

Reversing the flow yields a destabilising effect of the Coriolis force. Furthermore, the reaction at the free end of a pipe becomes complicated (Païdoussis *et al.*, 2005). In case of a pipe discharging the stream, the fluid does not change its velocity at the outlet and follows its lateral motion, so does not generate any reaction there. In case of an aspirating pipe the fluid is accelerated at the inlet, thus it brings tensional reaction, tangential to the free end. According to the Bernoulli equation, it decreases by a half the effect of the inertial force of the fluid inside. However, the lateral reaction of the fluid exerted on the pipe inlet is an open problem. We use the model assuming that the space-averaged direction of inflow is tangential to the deflected pipe and the flow field below the entrance does not follow the lateral motion of the pipe. This model overestimated the critical flow velocity observed in the experiment conducted by Kuiper and Metrikine (2008). The other model the could be applicable differs in that the flow field moves with the inlet, so it does not generate transverse reaction there (this model led to underestimation of the critical flow in above mentioned experiment).

The dynamics of lateral vibrations of the aspirating pipe is given by

$$
\begin{gathered}
\Big(1+\beta^{\star}\frac{\partial}{\partial t}\Big)EI\frac{\partial^4 w}{\partial x^4}+\Big[\frac{1}{2}MV^2-(M+m)g(L-x)\Big]\frac{\partial^2 w}{\partial x^2}-2MV\frac{\partial^2 w}{\partial x\partial t}\\
+(M+m)g\frac{\partial w}{\partial x}+MV\delta_L\frac{\partial w}{\partial t}+\gamma^{\star}\frac{\partial w}{\partial t}+(M+M_a+m)\frac{\partial^2 w}{\partial t^2}=\frac{A}{\mu_0}(B_2^2-B_1^2)\delta_e
\end{gathered}
\tag{2.2}
$$

The parameters of analysed systems are presented in Table 1 (by d and D we denote the internal and external diameter of the pipe). For the plastic pipe, they are the same as in the experimental work by Kuiper and Metrikine. The material from which the pipe is made and dimensions of the pipe determine permissible pressure of the supplied water. This pressure and the pipe geometry determine flow velocity. The parameters were selected so that the critical flow velocity is physically reachable (in the case of an aspirating pipe we took the effect of cavitation into account).

We assume the approximate solution to the equation of the pipe in a form

$$
\widetilde{w}(x,t)=\sum_{i=1}^{n}\widetilde{W}_i(x)T_i(t)
\tag{2.3}
$$

Table 1. Parameters of studied systems (pipes and actuators)

Parameter	Steel pipe	PVC pipe	Unit
L	2	4.75	m
d	0.01	0.07	m
D	0.0111	0.075	m
a	0.002	0.003	m
l	0.3	0.6	m
N	1000	1667	[–]
R	2.38	2.83	Ω

where $\widetilde{W}_i$, $i = 1,\dots,n$ are the eigenfunctions of a beam with an appropriate support, normalised with respect to integrals of their squares. Applying Galerkin's procedure, we obtain a set of $2n+2$ first-order differential equations for functions T_i, $\dot{T}_i$ and $B_{1,2}$. The stability of the middle equilibrium is determined by numerical calculations of eigenvalues of the linearized system.

The values of critical flow velocity presented in Section 4 were calculated for $n = 2,4,6,8,10$. We found that increasing the number of eigenfunctions from 8 to 10 makes almost no visible change in the shape of stability area, thus 10-modal approximation is accurate enough.

3. Internal and external damping coefficients

We calculate the internal viscous damping coefficient $\beta^\star$ by using the coefficient of dissipation of the material, which is defined as the ratio of energy dissipated as heat due to the internal friction, to the total energy of deformation (Dyląg *et al.*, 2000).

Consider a cantilever pipe of given geometry, without actuators and fluid inside, when the external damping and gravitation are absent. Then, its vibrations are given by the equation

$$\beta^\star EI\frac{\partial^5 w}{\partial x^4 \partial t} + EI\frac{\partial^4 w}{\partial x^4} + m\frac{\partial^2 w}{\partial t^2} = 0 \tag{3.1}$$

Assuming the solution in the form of a product

$$w(x,t) = W(x)T(t) \qquad 0 \leqslant x \leqslant L \qquad t > 0 \tag{3.2}$$

we obtain two independent differential equations. The solution to $W(x)$ is the sequence of eigenfunctions W_i characterising the shape of pipe deflection, with corresponding eigenvalues k_i ($i = 1,\dots,\infty$). We are interested in the first mode, because the values of the coefficients of dissipation are given for the bending trial. The vibrations of this mode are governed by another equation obtained through separation of variables

$$m\ddot{T}_1(t) + \beta EIk_1^4\dot{T}_1(t) + EIk_1^4T(t) = 0 \tag{3.3}$$

This is the equation of harmonic oscillator, for which one can calculate logarithmic decrement of damping. On the other hand, this decrement accounts for the half of the material coefficient of dissipation ψ. From these two relations, we get

$$\beta^\star = \frac{1}{k_1^2}\sqrt{\frac{m}{EI}}\frac{\psi}{\sqrt{4\pi^2 + \psi^2/4}} \tag{3.4}$$

The value of the coefficient of dissipation for steel is between 0.01-0.03, so we assume $\psi = 0.02$. Unfortunately, we have not found the value of ψ for PVC, thus we assume the mean of coefficients

Table 2. Calculated coefficients of the internal and external damping $\beta^\star$, $\gamma^\star$ and the added mass M_a for the studied pipes and surroundings

	Steel pipe		PVC pipe	
	air	water	air	water
$\gamma^\star$ [kg/m/s]	0.000966	0.170864	0.002753	0.477095
M_a [kg/m]	0.000180	0.111576	0.006244	4.644865
$\beta^\star$ [s]	0.000192		0.002646	

for steel and hard rubber, i.e. $\psi = 0.1$. The values of $\beta^\star$ calculated in this way for examined pipes are given in Table 2.

For the model of external damping we use the one described by Kirstein *et al.* (1998), which had been known earlier. The coefficient $\gamma^\star$ and the mass of fluid accelerated by vibrations of the pipe M_a are calculated with the formulas obtained from the analytical solution to the Navier-Stokes equation for the surrounding flow outside the pipe. In this model, one assumes that the pipe performs small harmonic vibrations of frequency ω and is placed far away from the walls of the tank with fluid. We use the following formulas

$$\gamma^\star = -\frac{1}{4}\pi\rho D^2\omega\Im\Big(1+\frac{4K_1(\alpha)}{\alpha K_0(\alpha)}\Big) \qquad\qquad M_a = \frac{1}{4}\pi\rho D^2\Re\Big(1+\frac{4K_1(\alpha)}{\alpha K_0(\alpha)}\Big) \tag{3.5}$$

where

$$\alpha = \sqrt{\mathrm{iRe}} \qquad\qquad \mathrm{Re} = \frac{\omega}{\nu}\frac{D^2}{4} \tag{3.6}$$

Re denotes the kinetic Reynolds number for external flow and K_0 and K_1 are modified Bessel functions of the second kind, respectively of the zero and first order.

The problem is what value of frequency ω ought to be used. The goal is the examination of stability, thus it should be the frequency of upcoming flutter vibrations. The external damping affects this frequency, so an iterative procedure is required. However, the fluid-conveying pipe can lose stability in several modes at the same time, what makes such a procedure cumbersome and not necessarily convergent (Kuiper *et al.*, 2007). The authors of that paper proposed a new model of hydrodynamic drag, independent of the frequency of vibrations. However, such a model is applicable only to flows with high values of the Reynolds number.

In this research, we apply a different approach, which is based on the numerically calculated frequency of free vibrations of the cantilever pipe filled by still water. The already estimated internal damping is incorporated, and the effect of actuators is neglected. The analytically calculated frequency of free vibrations of the first mode with the absence of external damping constitutes the starting point. In the next steps the procedure calculates the frequency in the middle of the pipe (basing on the first 10 cycles of free vibrations), and the parameters $\gamma^\star$ and M_a. The computations stop when a relative change of $\gamma^\star$ becomes smaller than 1%, what usually takes place after a few iterations. The obtained values are presented in Table 2.

4. Stability

We studied the influence of magnetic actuators on the stability of: the steel pipe supported at both ends, steel cantilever pipe conveying water downwards and aspirating pipe made of PVC. Points marked in figures denote the critical flow velocity for a given voltage, i.e. the border of stability area. In every figure the right border of the area was found by numerical calculation of V for which the real part of the decisive eigenvalue changed its sign, for a given value of voltage U (changed with an appropriate step). When computing the upper border, the flow velocity V was a parameter and U – a variable.

4.1. Two-side support

We consider the system of a steel pipe with electromagnetic actuators (Table 1). The pipe supported at both ends loses stability by buckling, and the clamped-clamped support raises the critical flow twice comparing to the simple support (Païdoussis and Issid, 1974). Dissipative forces – which depend on the velocity – do not affect the static bifurcation, so the dynamic effect of magnetic damping (Lenz rule) does not appear. Still, each actuator pulls the pipe, thus exerts the effect which destabilises the middle equilibrium. As a result, an increase of voltage worsens the stability of the pipe supported at both ends.

4.2. Cantilever pipe discharging water

In this subsection, we present results for a steel pipe (Table 1) clamped at its upper end. It has been proved that the cantilever pipe pumping the fluid downwards loses stability by flutter (Gregory and Païdoussis, 1966a,b), which let dissipative magnetic forces to appear.

Figures 2 and 3 show the influence of magnetic actuators on the stability area for the pipe placed in air and in water, respectively. Crossing the right-hand border of the area results in flutter, and the upper one yields divergence (buckling). One can see that the effect of magnetic damping is ambiguous. If the actuators are attached at the bottom end of the pipe, they definitely decrease the critical flow. Presumably the pipe is so slender that the static destabilising effect dominates over the damping one. It is difficult to point out the optimal position of actuators. From the practical point of view, one can see that the attachment of actuators in the upper part of the pipe – which is easier in technical implementation – leads to more effective stabilisation.

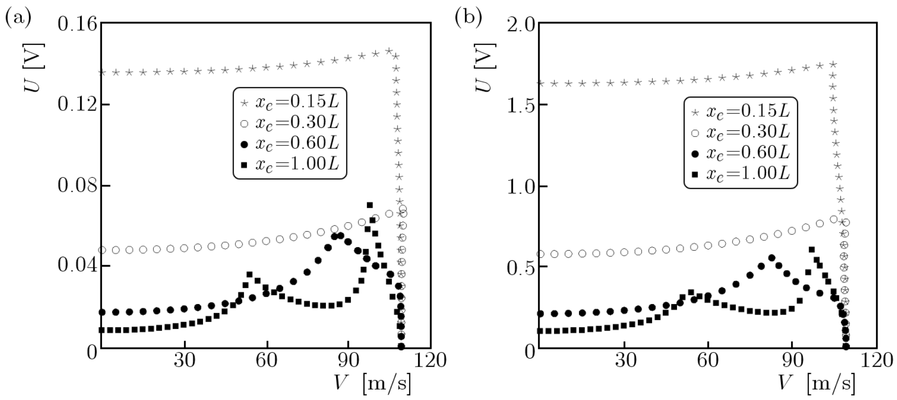

Fig. 2. Stability of the discharging pipe in air: (a) $z = 0.00013\,\mathrm{m}$, (b) $z = 0.001\,\mathrm{m}$

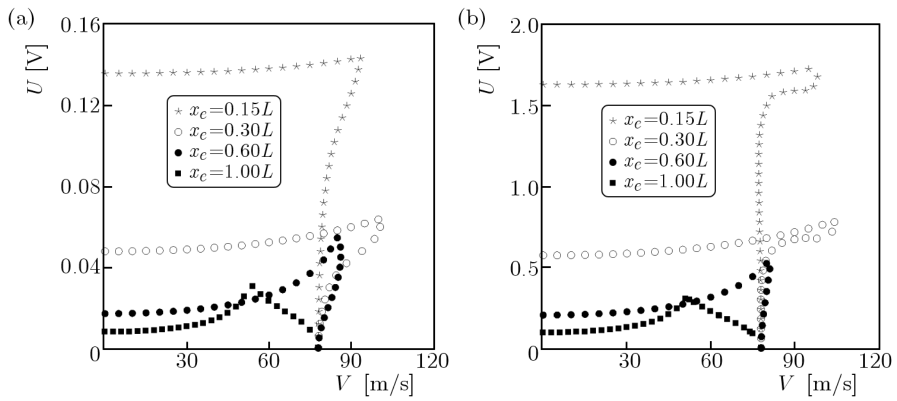

Fig. 3. Stability of the discharging pipe in water: (a) $z = 0.00013\,\mathrm{m}$, (b) $z = 0.001\,\mathrm{m}$

At the zero voltage on actuators, the critical flow velocity for the system immersed in water is lower than in air (for the given mass ratio of the fluid and the pipe, the external damping exerts a destabilising effect), and only in that environment the stabilising effect of magnetic damping can be observed. If the pipe is put in air, the actuators cannot increase the critical flow velocity.

Decreasing the gap in magnetic circuits improves the efficiency of magnetic damping, because the system can be stabilised at much lower values of U. This is why we study stiff pipes made of steel or PVC rather than rubber pipes, which lose stability at significantly lower flow velocities, but are susceptible to large-amplitude flutter vibrations.

4.3. Aspirating cantilever pipe

Finally, we present results for the electromagnetically damped pipe made of PVC (Table 1), through which the water is conveyed upwards. The stability of such a pipe was recently investigated by Kuiper and Metrikine (2008), both theoretically and experimentally. The observed critical flow velocity for the pipe partially – at 1.95 m depth – submerged in water amounted to 2.1-2.4 m/s, and destabilisation was due to flutter. The model with conventional boundary conditions (lack of transverse reaction at the free end) predicted the critical velocity of 0.73 m/s, whereas an incorporation of the transverse reaction (as assumed in our work) resulted in overestimation – 6.8 m/s. Both models correctly predicted oscillatory instability.

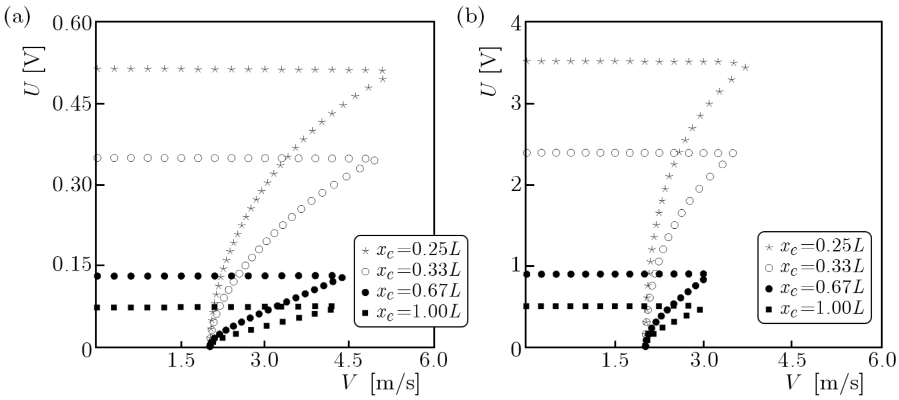

Fig. 4. Stability of the aspirating pipe in air: (a) $z = 0.001$ m, (b) $z = 0.004$ m

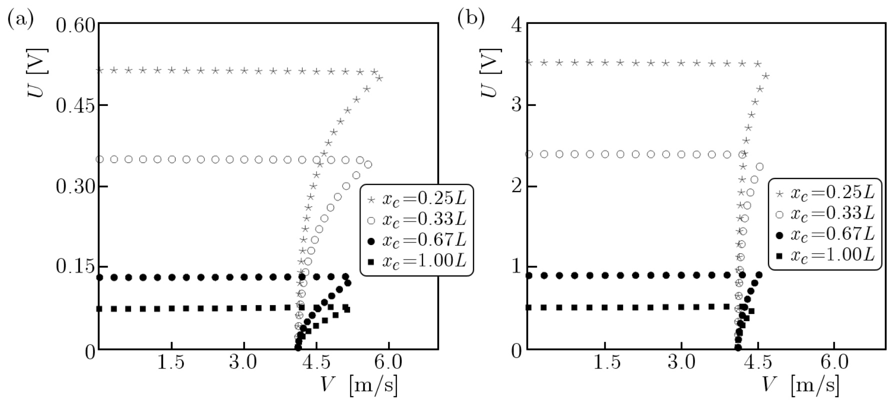

Fig. 5. Stability of the aspirating pipe in water: (a) $z = 0.001$ m, (b) $z = 0.004$ m

Figures 4 and 5 present stability areas for the pipe put, respectively, in air and in water, for two values of the gap in magnetic circuits: $z = 0.001$ m and $z = 0.004$ m. At the zero voltage, the critical flow velocity amounts to $\sim$2 m/s (in air) and 4.1 m/s (water). Thus the critical flow for the pipe partially submerged – as in the experiment by Kuiper and Metrikine – would be around 3-3.5 m/s, which is lower than the value calculated in their paper. The reason lies in the difference between the assumed models of hydrodynamic drag. Kuiper and Metrikine used a linear-quadratic model developed earlier by themselves and Battjes (2007), with coefficients independent of the frequency of vibrations and estimated from auxiliary experiments. In our research, we use a more widely known linear model, whose parameters have to be estimated iteratively, as they depend on the frequency.

Once the voltage is applied, the positive effect of electromagnetic damping reveals, and is much more evident than in a pipe through which the water is pumped downwards. As previously, stabilisation is also stronger for the system with a lower critical flow velocity – but here it occurs in air, not in water. Smaller gap in magnetic circuits allows not only for significant reduction of the voltage applied to actuators, but also results in higher critical flow. Actuators are most effective when placed in the upper part of the pipe. However, unlike the discharging case, they also exert stabilising action when attached at the free end.

5. Summary

We studied the influence of electromagnetic actuators on the dynamics of a fluid-conveying pipe. Simply supported, clamped-clamped and cantilever (discharging and aspirating) pipes were investigated. Assumed physical parameters of the systems will enable future experimental verification of the results. We proposed simple methods of calculation of the internal and external damping coefficients, which do not require experiments, and are based on known models.

The governing equation was discretised with 10-modal Galerkin's procedure based on the beam eigenfunctions. Such number of eigenmodes has proved to be sufficiently convergent. Then the stability of the resultant dynamical system was checked with eigenvalues of its linearization.

The actuators destabilise the pipe supported at both ends, but can improve the stability of a cantilever one. The efficiency of magnetic damping depends on the external damping of environment in which the system is immersed. Moreover, it depends on the position at which the actuators are attached to the pipe. The optimal position of the actuators is an open problem, but important from the practical point of view position in the upper part of the pipe brings stabilisation. Decreasing the gap in magnetic circuits improves the efficiency of the method.

Destabilisation of all studied pipes occurs at very high flow velocities. Such extreme flows are rarely encountered in every-day life and are difficult (yet possible) to achieve.

References

1. Dyląg Z., Jakubowicz A., Orłoś Z., 2000, *Strength of Materials*, Vol. II, 2nd edition, WNT, Warsaw, 222-228 [in Polish]

2. Edelstein W.S., Chen S.S., 1985, Flow-induced instability of an elastic tube with a variable support, *Nuclear Engineering and Design*, 84, 1-11

3. Elishakoff I., Impollonia N., 2001, Does a partial elastic foundation increase the flutter velocity of a pipe conveying fluid? *Journal of Applied Mechanics*, **68**, 206-212

4. Feodos'ev V.P., 1951, Vibrations and stability of a pipe when liquid flows through it, *Inzhenernyi Sbornik*, **10**, 169-170

5. Gregory R.W., Païdoussis M.P., 1966a, Unstable oscillation of tubular cantilevers conveying fluid. I. Theory, *Proceedings of the Royal Society (London) A*, **293**, 512-527

6. Gregory R.W., Païdoussis M.P., 1966, Unstable oscillation of tubular cantilevers conveying fluid. II. Experiments, *Proceedings of the Royal Society (London) A*, **293**, 528-542

7. Holmes P.J., Moon F.C., 1983, Strange attractors and chaos in nonlinear mechanics, *Journal of Applied Mechanics*, **50**, 4b, 1021-1032

8. Housner G.W., 1952, Bending vibrations of a pipe line containing flowing fluid, *Journal of Applied Mechanics*, **19**, 205-208

9. Impollonia N., Elishakoff I., 2000, Effect of elastic foundation on divergence and flutter of an articulated pipe conveying fluid, *Journal of Fluids and Structures*, **14**, 559-573

10. Kirstein S., Mertesdorf M., Schönhoff M., 1998, The influence of a viscous fluid on the vibration dynamics of scanning near-field optical microscopy fiber probes and atomic force microscopy cantilevers, *Journal of Applied Physics*, **84**, 4, 1782-1790

11. Kuiper G.L., Metrikine A.V., 2008, Experimental investigation of dynamic stability of a cantilever pipe aspirating fluid, *Journal of Fluids and Structures*, **24**, 541-558

12. Kuiper G.L., Metrikine A.V., Battjes J.A., 2007, A new time-domain drag description and its influence on the dynamic behaviour of a cantilever pipe conveying fluid, *Journal of Fluid and Structures*, **23**, 429-445

13. Kurnik W., 1994, Magnetic stabilization of a rotor with hydrodynamic bearings, *Machine Dynamics Problems*, **7**, 117-133

14. Kurnik W., Przybyłowicz P., Szmidt T., 2009, An application of electromagnetic phenomena to damping of mechanical vibrations, [In:] *Influence of Parameter Modifications on Structure Vibrations*, Tylikowski A. (Edit.), Warsaw University of Technology, 53-87

15. Lottati I., Kornecki A., 1986, The effect of elastic foundation and dissipative forces on the stability of fluid conveying pipes, *Journal of Sound and Vibration*, **109**, 327-338

16. Modarres-Sadeghi Y., Païdoussis M.P., Semler C., 2008, Three-dimensional oscillations of a cantilever pipe conveying fluid, *International Journal of Non-Linear Mechanics*, **43**, 18-25

17. Moon F.C., 1980, Experiments on chaotic motions of a forced nonlinear oscillator: strange attractors, *Journal of Applied Mechanics*, **47**, 3, 638-644

18. Païdoussis M.P., 1998, *FluidStructure Interactions: Slender Structures and Axial Flow*, Vol. 1, Academic Press, London

19. Païdoussis M.P., 2008, The canonical problem of the fluid-conveying pipe and radiation of the knowledge gained to other dynamics problems across applied mechanics, *Journal of Sound and Vibration*, **310**, 462-492

20. Païdoussis M.P., Issid N.T., 1974, Dynamic stability of pipes conveying fluid, *Journal of Sound and Vibration*, **33**, 3, 267-294

21. Païdoussis M.P., Semler C., Wadham-Gagnon M., 2005, A reappraisal of why cantilever pipes do not flutter at infinitesimal flow, *Journal of Fluid and Structures*, **20**, 147-156

22. Przybyłowicz P., Szmidt T., 2009, Electromagnetic damping of a mechanical harmonic oscillator with the effect of magnetic hysteresis, *Journal of Theoretical and Applied Mechanics*, **47**, 2, 259-273

23. Sugiyama Y., Tanaka Y., Kishi T., Kawagoe H., 1985, Effect of a spring support on the stability of pipes conveying fluid, *Journal of Sound and Vibration*, **100**, 257-270

24. Szmidt T., Przybyłowicz P., 2012, An active electromagnetic stabilisation of the Leipholz column, *Archives of Control Sciences*, **22**, 2, 125138

25. Tang D.M., Dowell E.H., 1988, Chaotic oscillations of a cantilevered pipe conveying fluid, *Journal of Fluids and Structures*, **2**, 3, 263-283

Krytyczna prędkość przepływu w przewodzie stabilizowanym aktuatorami elektromagnetycznymi

Streszczenie

W pracy przeprowadzono badania wpływu tłumienia elektromagnetycznego na właściwości dynamiczne przewodu z przepływającym płynem. Rozważono przewody podparte obustronnie swobodnie oraz zamocowane wspornikowo (dla obydwu przypadków ruchu płynu, tj. rury wyrzucającej czynnik na zewnątrz oraz rury ssącej). Założono takie parametry fizyczne układu, które pozwalają na weryfikację eksperymentalną uzyskanych wyników. Zaproponowano proste metody pozwalające na oszacowanie wartości tłumienia zewnętrznego i wewnętrznego oparte na znanych modelach i nie wymagających przeprowadzania doświadczeń. Równanie różniczkowe cząstkowe ruchu przewodu z przepływem zdyskretyzowano metodą Galerkina, a po wyznaczeniu wartości własnych układu zlinearyzowanego, określono jego stateczność. W wyniku przeprowadzonych analiz zaobserwowano, że aktuatory elektromagnetyczne destabilizują przewód obustronnie podparty, ale znacząco poprawiają stateczność rury wspornikowej, przy czym efekt tłumienia magnetycznego silnie zależy od położenia aktuatorów względem miejsca zamocowania takiego przewodu.

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