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Vibration reduction of a pipe conveying fluid using the semi-active electromagnetic damper

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Abstract. This paper deals with a uniform cantilever Euler-Bernoulli beam subjected to follower and transversal force at its free end as a model for a pipe conveying fluid under electromagnetic damper force. The electromagnetic damper is composed of a permanent-magnet DC motor, a ball screw and a nut. The main objective of the current work is to reduce the pipe vibration resulting from the fluid velocity and allow it to transform into electric energy. To pursue this goal, the stability and vibration of the beam model was studied using Ritz and Newmark methods. It was observed that increasing the fluid velocity results in a decrease in the motion of the free end of the pipe. The results of simulation showed that the designed semi-active electromagnetic damper controlled by on-off damping control strategy decreased the vibration amplitude of the pipe about 5.9% and regenerated energy nearly 1.9 (mJ/s). It was also revealed that the designed semi-active electromagnetic damper has better performance and more energy regeneration than the passive electromagnetic damper.

Keywords: stability; pipe; fluid; vibration; electromagnetic; damper

1. Introduction

Pipe conveying fluid has significant applications in a wide range of industrial and engineering applications. This subject is utilized for analyzing the fluid-structure interaction which has vast applications in aerospace, medical engineering, oil and gas industries, power plants, chemical plants, hydraulic systems, refrigerators, air-conditioners, heat exchangers and so on. This paper investigates the vibration reduction of a pipe conveying fluid by means of a semi-active electromagnetic damper controlled by on-off damping control strategy. In fact, the structure is modeled by a uniform cantilever Euler-Bernoulli beam subjected to follower and transversal forces at its free end. This is considered as an acceptable model for such structures with the internal fluid flow and controller forces. The latter is the force for the actuator to control the vibration of the pipe. The aim of this study is to analyze the stability of the model and reduce the pipe vibration resulting from the fluid velocity and allow it to transform into electric energy. It will be shown that the semi-active electromagnetic damper controlled by on-off damping control strategy decreases the pipe vibration while the vibration energy can be converted to electric energy. The Ritz method

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is used in the calculations of the system frequencies and the Newmark method is employed for the study of the vibrational properties of the model.

The stability of the beam under the follower force is in the interests of many researchers as it can be applied to many structures. The direction of the follower force is always perpendicular to the cross surface of the beam and changes with the beam deflections. Follower forces have an important influence on the structural natural frequencies. The critical follower force may cause static instability (divergence) or dynamic instability (flutter). Several researchers have published their work on a beam under a follower force (Kim and Kim 2000, Ryu and Sugiyama 2003, Wang 2004, Shvartsman 2007, Djondjorov and Vassilev 2008, Irani and Kavianipour 2009, Kavianipour and Sadati 2009, Kavianipour *et al.* 2013, Caddemi *et al.* 2014, Sohrabian *et al.* 2016). Chellapilla and Simha (2008) studied the problem of vibrations of fluid-conveying pipes resting on a two-parameter foundation model such as the Pasternak-Winkler model. Yi-min *et al.* (2010) investigated the natural frequency of fluid-structure interaction in pipeline conveying fluid by eliminated element-Galerkin method, and obtained the natural frequency equations with different boundary conditions. Gongfa *et al.* (2014) simulated the axial motion equation of the conveying fluid pipe, and studied the response of the system in two aspects of fluid pressure disturbance and the fluid velocity disturbance. Jweeg and Ntayeesh (2015) carried out experimental work by built a rig which was mainly composed of a different boundary condition of pipes conveying fluid and provided with the necessary measurement equipments to fulfill the required investigations. Kokare and Paward (2015) analyzed the natural frequency of straight pipe made of structural steel, conveying turbulent steady water with different boundary conditions. Al-Hilli and Ntayeesh (2015) derived a new analytical model to perform a general study to investigate the dynamic behavior of a pipe under general boundary conditions by considering the supports as compliant material with linear and rotational springs, and consider the effect of foundation and stiffness values on the critical velocity. Jweeg and Ntayeesh (2016) presented a new experimental approach for estimating buckling critical velocities from measuring several natural frequencies at relatively small flow rates. Wen *et al.* (2016) developed a 14-equation model considering the effects of the thickness of pipe wall, fluid pressure and velocity, which used dimensionless terms to describe the fluid-structure interaction behavior of pipelines. Kesimli *et al.* (2016) calculated exact natural frequencies by the solution of the linear problem for the different positions of the support at the center, different longitudinal stiffness, different pipe coefficient, different rate of fluid velocity. Mohamed *et al.* (2016) studied the effects of stiffness addition (linear spring), spring location, and flow velocity on the dynamic stability of the pipe conveying fluid with internal flow. Raminnea *et al*. (2016) indicated that with increasing the stiffness of elastic medium and decreasing volume percent of nanoparticle in fluid, the frequency and critical fluid velocity increase. Zamani Nouri (2017) showed the effects of fluid, volume percent of CNTs, magnetic field and geometrical parameters on the frequency and critical fluid velocity of the system.

There are several types of energy regenerative systems already under investigation, such as hydraulic storage system (Chen *et al.* 2007), battery coil induction system (He *et al.* 2005), rack and pinion system (Chen *et al.* 2006), ball screw system (Zhang *et al.* 2008), and linear motion system (Wu and Cao 2007). Developments achieved in power electronics, permanent magnet materials, and microelectronic systems enable the possibility of actual implementation of electromagnetic actuators in order to improve the performance of systems (Martins *et al.* 2006). Due to the several advantages such as high responsiveness, energy saving performance, controllability, etc., electromagnetic damper (ball screw shock absorber) is attracting much interest nowadays. Actually, ball screw mechanism converts linear motion of vibration into input torque

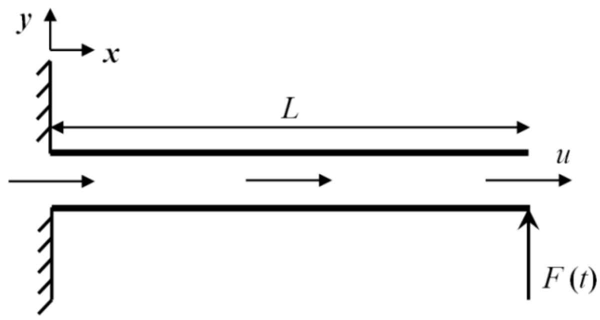


Fig. 1 The simple model of a pipe conveying fluid under electromagnetic damper force

for a DC motor. Nakano *et al.* (2003) studied the self-powered active vibration control using a single actuator. The actuator generated power while the speed of the armature is high. The regenerated power would be applied when the speed of the armature is low. Suda and Shiba (1996) proposed a regenerative system, in which a rotary DC motor was used as an actuator. The rotary motion of DC motor was converted into the linear motion by the rack and pinion mechanism. Suda *et al.* (2000) produced a prototype of an electromagnetic actuator in which ball screw mechanism was used to convert rotary motion into linear motion. Nakano and Suda (2004) applied this self-powered active vibration control system to the cab suspension of a truck. Kawamoto and Suda (2007) analyzed the frequency characteristics of energy balance of the electromagnetic damper. Experimental verification of energy-regenerative feasibility for an automotive electrical suspension system was carried out by Zhang *et al.* (2007). Zhang *et al.* (2009) investigated the application of the permanent-magnet DC motor actuator in automotive active suspension considering energy regeneration. Montazeri-Gh and Kavianipour (2012, 2014a, b) studied the performance of the electromagnetic damper as a passive, semi-active and active actuator. They extracted the nonlinear equations of the electromagnetic damper.

This research inspects the feasibility of an electromagnetic damper in providing adequate damping for isolation of vibration while generates energy from the motion of the pipe. Semi-active vibration isolation involves changing the damping coefficient as a function of time. This work essentially consists of two parts:

Part 1 includes the stability analysis of the pipe conveying fluid as a uniform cantilever Euler-Bernoulli beam with internal fluid flow.

Part 2 makes use of the on-off damping control strategy for improving the performance of the electromagnetic damper. In this part, a simulation study is carried out to show that the semi-active electromagnetic damper has good performance in comparison with the passive electromagnetic damper while more vibration energy can be regenerated and saved in the capacitor.

2. Mathematical modeling

Fig. 1 shows the assumed model for a pipe conveying fluid under electromagnetic damper force. Define the xy-coordinate system as the elastic frame. In this figure, the transversal force $F(t)$ represents the electromagnetic damper force. Also, L and u are the beam length and fluid flow velocity, respectively. The beam is assumed to be axially rigid and is a uniform Euler-Bernoulli beam.

2.1 Energy method

One of the methods to derive the governing equations is the energy method. In fact, by considering all the energies in the system and using the Hamilton's Principle, the governing equations could be derived correctly. The general form of the Hamilton's Principle emerges as (Meirovitch 1997)

$$\delta\int_{t_1}^{t_2}\left(E_k - E_p\right)\mathrm{d}t + \int_{t_1}^{t_2}\delta W_{nc}\,\mathrm{d}t = 0 \tag{1}$$

where δ is the variation, t is the time, E_k is the kinetic energy, E_p is the potential energy, and W_{nc} is the work done by non-conservative forces. For the model shown in Fig. 1, Eq. (1) may be expressed in the form

$$\begin{cases} E_k = \dfrac{1}{2}m_p\displaystyle\int_0^L \dot{y}^2\,\mathrm{d}x + \dfrac{1}{2}m_f\int_0^L\left(u^2 + \dot{y}^2 + 2\dot{y}uy' + u^2y'^2\right)\mathrm{d}x \\ E_p = \dfrac{1}{2}EI\displaystyle\int_0^L y''^2\,\mathrm{d}x \\ \delta W_{nc} = F(t)\delta y(x_F, t) \end{cases} \tag{2}$$

where $\dot{(\ )} = \mathrm{d}(\ )/\mathrm{d}t$, $\ddot{(\ )} = \mathrm{d}^2(\ )/\mathrm{d}t^2$, $(\ )' = \mathrm{d}(\ )/\mathrm{d}x$, and $(\ )'' = \mathrm{d}^2(\ )/\mathrm{d}x^2$.

In Eq. (2), m_p and m_f are the pipe and fluid masses per length, L is the beam length, u is the fluid flow velocity, y is the transverse displacement, EI is the bending stiffness, $F(t)$ is the transverse force, and x_F is the location of the transverse force on the beam. It is to be noted that the rotational inertia of the beam and gravity force are ignored in this paper.

To solve the equation, an approximation method named Ritz method has been employed in this study. In this method, the response is approximated with a series as the following (Hodges and Pierce 2002)

$$y(x,t) = \sum_{i=1}^{N}\psi_i(x)q_i(t) \tag{3}$$

$\psi_i(x)$ is admissible function and $q_i(t)$ is a generalized coordinate.

Replacing Eqs. (2)-(3) in Eq. (1), then by simplifying the results and writing the equation in matrix form, Eq. (4) will result as

$$\lfloor \mathbf{M_{ij}} \rfloor\{\ddot{q}_j\} + \lfloor \mathbf{C_{ij}} \rfloor\{\dot{q}_j\} + \lfloor \mathbf{K_{ij}} \rfloor\{q_j\} = \lfloor \mathbf{Q_j} \rfloor \tag{4}$$

In the above equation, $\lfloor \mathbf{M_{ij}} \rfloor$ is the mass matrix, $\lfloor \mathbf{C_{ij}} \rfloor$ is the damping matrix, $\lfloor \mathbf{K_{ij}} \rfloor$ is the stiffness matrix, and $\lfloor \mathbf{Q_j} \rfloor$ is the generalized force vector which can be described as

$$\begin{cases} \mathbf{M_{ij}} = \left(m_p + m_f\right)\displaystyle\int_0^L \psi_i\psi_j\,\mathrm{d}x \\ \mathbf{C_{ij}} = 2um_f\displaystyle\int_0^L \psi_i'\psi_j\,\mathrm{d}x \\ \mathbf{K_{ij}} = m_f u^2\displaystyle\int_0^L \psi_i''\psi_j\,\mathrm{d}x + EI\int_0^L \psi_i''\psi_j''\,\mathrm{d}x \\ \mathbf{Q_j} = F(t)\psi_j(x_F) \end{cases} \tag{5}$$

Table 1 Parameters values of the model

Description	Pipe: Steel
Density	ρ_p=7747 (kg/m^3)
Modulus of elasticity	E=207 (GPa)
Length	L=2 (m)
Inner diameter	D_i=0.05 (m)
Outer diameter	D_o=0.054 (m)
Moment of inertia	$I=(\pi/64)\times(D_o^4-D_i^4)$ (m^4)

2.2 Admissible functions

In general the admissible functions should satisfy four conditions (Hodges and Pierce 2002):
- At least must satisfy all geometric boundary conditions.
- Must be continuous and differentiable to highest spatial derivative.
- Should be a complete function.
- Must be linearly independent.

The mode shapes of a clamped-free beam satisfy the above conditions and have been used in this work (Meirovitch 1997).

$$\psi_i(x) = \bar{A}_i \left\{ \begin{array}{l} (\sin(\lambda_i L) - \sinh(\lambda_i L))(\sin(\lambda_i x) - \sinh(\lambda_i x)) \\ + (\cos(\lambda_i L) + \cosh(\lambda_i L))(\cos(\lambda_i x) - \cosh(\lambda_i x)) \end{array} \right\} \qquad i = 1,2,3,\ldots \tag{6}$$

where

$$\begin{cases} \bar{A}_i = \dfrac{A_1}{\sin(\lambda_i L) - \sinh(\lambda_i L)} \\ A_1 = 1 \\ \lambda_i^4 = \dfrac{m_p}{EI}\omega_i^2 \end{cases} \tag{7}$$

and ω_i is the natural frequency of the beam.

2.3 Critical velocity

One of the important objectives of this paper is to calculate the magnitude of the fluid critical velocity leading to instability, i.e., u_{cr}. As seen in Eq. (5), the fluid flow velocity affects the system stiffness and damping matrices and changes the system frequencies. Therefore, to obtain the changes in the system frequencies in terms of the fluid flow velocity, set the right hand side of Eq. (4) to zero (i.e., $F(t)$=0) and suppose the homogeneous response as follows.

$$\{q_j\} = \{\bar{q}_j\}e^{\beta t}, \quad \beta = \alpha \pm j\omega, \quad j = \sqrt{-1} \tag{8}$$

where $\{\bar{q}_j\}$ is a vector with constant elements (eigenvectors), β is the eigenvalues, and ω is the system frequencies. When α takes positive sign or ω becomes zero, the instability of the beam occurs.

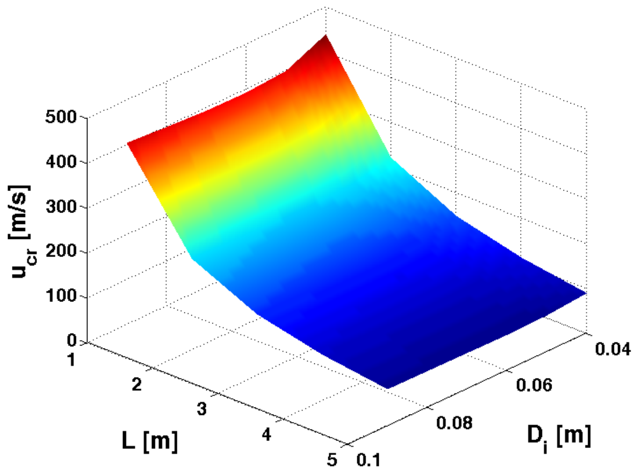


Fig. 2 The effects of the L and D_i on the critical velocity assuming constant thickness

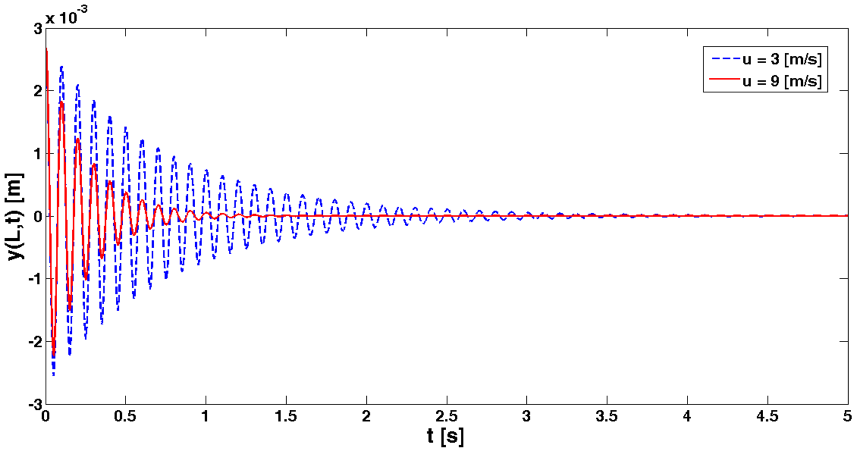


Fig. 3 The decrease in the displacement of the free end of the pipe due to the increase in the fluid flow velocity

To assure the validity of the computer code first, the values of the u and m_f parameters were set to zero and the others were considered in accordance with Table 1 (Faal and Derakhshan 2011). The inner fluid is assumed to be water and its density (ρ_f) is equal to 997 (kg/m^3). It was observed that the resulting frequencies from Eq. (4) were equal to the uniform cantilever Euler-Bernoulli pipe. It is to be noted that to solve Eq. (4) in the present work, the first three mode shapes of the model (N=3) are considered.

By increasing the value of u, the values of ω are lessened (as cited by Jweeg and Ntayeesh 2015) until the first system frequency becomes zero. Indeed, the instability of the pipe occurs at this value of the fluid velocity, i.e., u_{cr}. Since this value of u makes the first system frequency become zero, the resulting instability is called static instability or divergence. To analyze the displacement of the free end of the pipe (i.e., $y(L,t)$) due to the fluid velocity, the Newmark method (Craig and Kurdila 2006) is used in the present work.

Fig. 2 depicts the effects of the L and D_i on the fluid critical velocity assuming constant

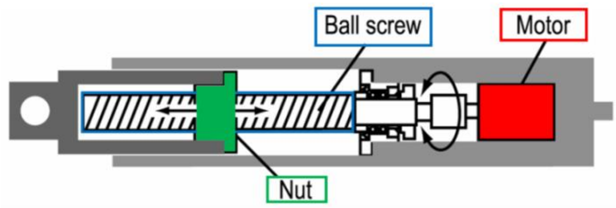


Fig. 4 The passive electromagnetic damper

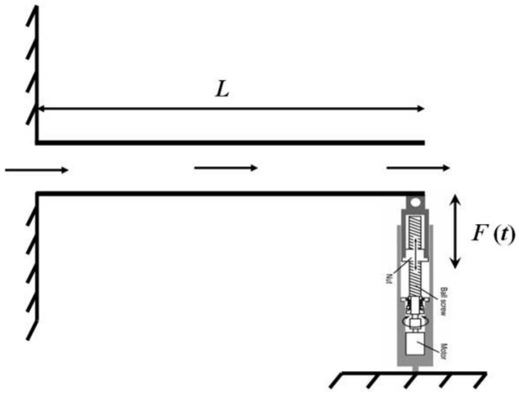


Fig. 5 The electromagnetic damper force at the free end of the pipe

thickness. It is quite clear from this figure that the critical velocity decreases with the increase in the length of the pipe. Moreover, an increase in the D_i will at first lead to a decrease of the value of the critical velocity, and brings about a rise in this parameter later.

Fig. 3 illustrates the vibrational properties with the x=L=2 (m) as a function of time for the two different cases of the fluid flow velocity assuming a given initial condition. It is observed that increasing the fluid velocity results in a decrease in the motion of the free end of the pipe. Furthermore, the results of Fig. 3 indicate that the displacement of the free end of the pipe tends to eliminate in time due to the presence of damping effects, i.e., Coriolis force.

3. Electromagnetic damper

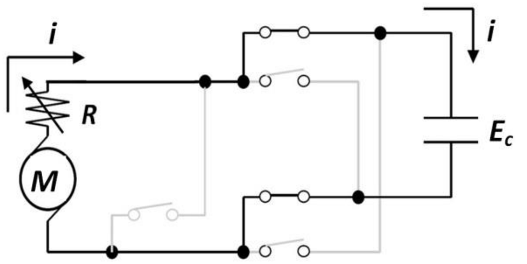
The electromagnetic damper consists of DC motor and the ball screw mechanism as shown in Fig. 4. It converts linear motion of vibration into input torque for DC motor. The electromagnetic damper applies electromagnetic force as the damping force. In the passive application, it has almost linear damping characteristics. The modeling of the electromagnetic damper was presented by Kawamoto *et al.* (2007) and Montazeri-Gh and Kavianipour (2012). For this, the two primary assumptions made were:

• Motor rotor and ball screw are supposed to be rigid.

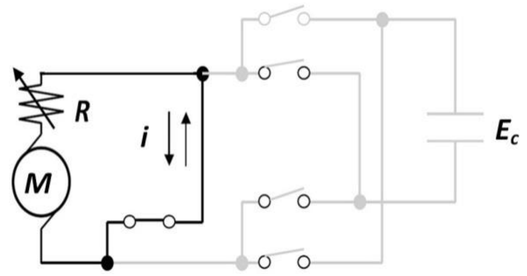
• Backlash and torsion of ball screw are ignored.

The output force of the electromagnetic damper shown in Fig. 5 is in the form

$$F(t) = f_e - I_e \ddot{y}(x_F, t) - c_e \dot{y}(x_F, t) \tag{9}$$



(a) Electric circuit of the electromagnetic damper in regeneration mode when $\dot{y}(x_F,t)>0$



(b) Electric circuit of the electromagnetic damper in dissipation mode

Fig. 6 Electric circuit of the electromagnetic damper

where

$$\begin{cases} f_e = -\phi i, \quad \phi = \dfrac{2\pi}{l_b} k_m \\ I_e = \left(\dfrac{2\pi}{l_b}\right)^2 (J_m + J_b) \end{cases} \tag{10}$$

In Eqs. (9)-(10), f_e is the force, I_e is the equivalent inertia, c_e is the equivalent damper coefficient, Φ is the motor constant, i is the current, l_b is the lead of the ball screw, k_m is the torque constant, J_m is the moment of inertia of motor rotor and J_b is the moment of inertia of ball screw.

3.1 Electric circuit of the electromagnetic damper

Due to environmental and economical concerns, clean energies are of vital importance and are of interest to many researchers. In this direction, the use of electrical components and energy storage systems has created new opportunities.

In this section, energy storage in the electromagnetic damper is investigated by numerically evaluation. For this purpose, the motor circuit of electromagnetic damper is modeled as the equivalent DC motor circuit. Since there is a condenser in the electric circuit for storage of energy, the current is given by

$$i = \frac{e - E_c}{R}, \quad e = \phi \dot{y}(x_F, t) \tag{11}$$

where e is the induced voltage, E_c is the voltage of the condenser, and R includes both the internal resistance of the motor and the additional (constant or variable) resistance placed in the circuit. Actually, R is one of the most effective parameters of the electromagnetic damper on its performance.

It can be shown that the relationship between the variation of the condenser voltage and the current is expressed as follows

$$\dot{E}_c = \frac{1}{c_{con}} i \tag{12}$$

where c_{con} is the capacity of the condenser.

Table 2 Parameters values of the electromagnetic damper

Parameter	Value
ϕ	11 (N/A)
$R_{internal}$	11.8 (Ω)
c_{con}	0.08 (F)
c_e	0 (Ns/m)
I_e	0.6 (kg)

The electromagnetic damper rectifies induced voltage and transfers the energy to the condenser. The mode in which the electromagnetic damper regenerates vibration energy is called the regeneration mode (Nakano *et al*. 2001). However, when the magnitude of the induced voltage is less than the voltage of the condenser, the electromagnetic damper can NOT transfer it to the condenser. In this mode, the electromagnetic damper stops regeneration. This mode is called the dissipation mode. The Circumstance of operating the electric circuit of the electromagnetic damper is demonstrated in Fig. 6. To compute the variation of the condenser voltage, the nonlinear equations of the electromagnetic damper electric circuit must be solved in accordance with Fig. 6.

By substituting Eq. (11) in Eqs. (10) and (12), the electromagnetic damper force and the equations of its electric circuit is expressed as

when $\phi|\dot{y}(x_F,t)| \geq E_c$ (regeneration mode)

$$\begin{cases} f_e = -\dfrac{\phi^2}{R}\dot{y}(x_F,t) + \dfrac{\phi}{R}E_c \operatorname{sign}(\dot{y}(x_F,t)) \\ \dot{E}_c = \dfrac{1}{c_{con}}\left(\dfrac{\phi|\dot{y}(x_F,t)| - E_c}{R}\right) \end{cases} \tag{13}$$

when $\phi|\dot{y}(x_F,t)| < E_c$ (dissipation mode)

$$\begin{cases} f_e = -\dfrac{\phi^2}{R}\dot{y}(x_F,t) \\ \dot{E}_c = 0 \end{cases} \tag{14}$$

Comparing Eq. (13) with Eq. (14), it can be discovered that the electromagnetic damper reduces its damping force when it regenerates vibration energy.

The characteristics of the electromagnetic damper are considered in accordance with Nakano *et al.* (2001) and represented in Table 2.

3.2 On-off damping control strategy

The resistance, *R*, plays an important role in the electromagnetic damper performance. If the value of the resistance is constant, the electromagnetic damper will act as a passive damper. In this study, on-off damping control strategy is considered for the semi-active electromagnetic damper. The semi-active damper is switched off when the position and velocity have the same sign, and is switched on when the position and velocity are in the opposite direction. This control strategy is termed "on-off balance control" (Liu 2004). In practice, the corresponding semi-active damping considering non-zero off-state damping is given by

Table 3 Comparison of the results for the three cases

	First Peak (m)	E_c (v)
Without Electromagnetic Damper	-2.55×10^{-3}	0
With Passive Electromagnetic Damper	-2.44×10^{-3}	0.20
With Semi-active Electromagnetic Damper	-2.40×10^{-3}	0.22

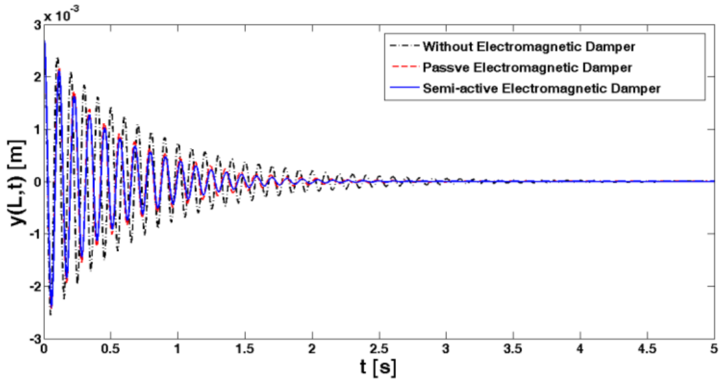


Fig. 7 The vibration of the free end of the pipe with and without electromagnetic damper

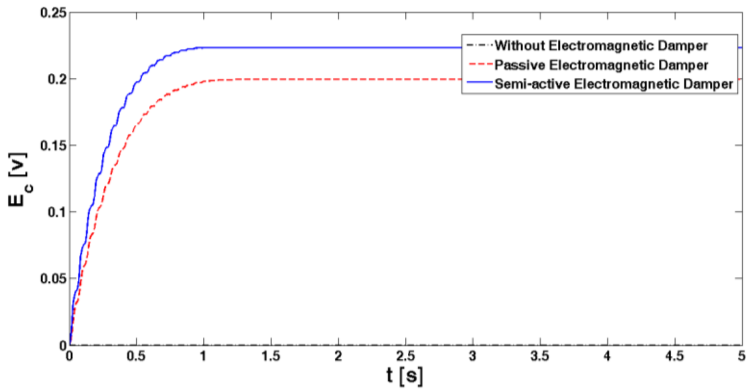


Fig. 8 The variation of the condenser voltage

$$R = \begin{cases} R_{\min} & y(x_F, t)\dot{y}(x_F, t) \le 0 \\ R_{\max} & y(x_F, t)\dot{y}(x_F, t) > 0 \end{cases} \tag{15}$$

where R_{min} and R_{max} are the minimum and maximum values of the resistance in the circuit.

4. Results of the use of the semi-active electromagnetic damper

In order to study the effects of the semi-active electromagnetic damper on the pipe vibration and the amount of the energy saving, the pipe vibration and electromagnetic damper equations must be solved simultaneously. To evaluate the energy storage per time in the condenser, the equation below can be used.

$$E_{s/t} = \left(\frac{1}{2} c_{con} E_{c_2}^{\;2} - \frac{1}{2} c_{con} E_{c_1}^{\;2} \right) \Big/ (t_2 - t_1) \tag{16}$$

where E_{c1} and E_{c2} are the voltage of the condenser at the times t_1 and t_2, respectively.

The results of simulation are presented in Figs. 7-8 for u=3 (m/s), x_F=L=2 (m), R_{min}=$R_{internal}$=11.8 (Ω), and R_{max}=31.8 (Ω). Also, the value of R is supposed to be 21.8 (Ω) for the passive electromagnetic damper. It can be generally deduced from Figs. 7-8 that the semi-active electromagnetic damper has better performance and more energy regeneration than the passive electromagnetic damper.

The vibration of the free end of the pipe is shown in Fig. 7 in three cases (without electromagnetic damper, with passive and semi-active electromagnetic damper). The vibration is obviously reduced when the electromagnetic damper is applied. Fig. 8 demonstrates the variation of the condenser voltage with time. As seen in this figure, the condenser voltage is at first increased evidently because of the vibration of the free end of the pipe. Afterward, the condenser voltage is constant due to a decrease in the vibrational amplitude.

The results have been compared in Table 3 for the three cases. The first peak of the vibration amplitude is reduced about 4.3% when the passive electromagnetic damper is utilized, whereas this amount is equal to 5.9% for the semi-active case. The value of the energy storage is approximately 1.6 (mJ/s) and 1.9 (mJ/s) for the passive and semi-active electromagnetic damper, respectively.

5. Conclusions

In this work, the performance of the semi-active electromagnetic damper on a uniform cantilever pipe with internal fluid flow was investigated and analyzed. It was assumed that the inner fluid is water and the pipe is made of steel.

• At first the stability of the pipe was studied using Ritz method and the dependence of the fluid critical velocity on the length and inner diameter of the pipe was considered. It was distinguished that the critical velocity decreases with the increase in the length of the pipe assuming constant thickness. Moreover, an increase in the inner diameter of the pipe leads to a decrease in the value of the critical velocity at first, and then causes a slightly rise in this parameter later.

• The vibrational properties of the model were computed using Newmark method. It was observed that increasing the fluid velocity results in a decrease in the motion of the free end of the pipe and then, the divergence takes place.

• Afterward, the effects of the semi-active electromagnetic damper on the pipe vibration and the energy storage were evaluated. By numerical assessment, it was revealed that the vibration of the free end of the pipe is reduced about 5.9% when the designed semi-active electromagnetic damper is applied. Besides, the findings indicated that the vibration energy of the pipe can be converted to electric energy and stored in the condenser nearly 1.9 (mJ/s).

• Likewise, it was shown that the designed semi-active electromagnetic damper controlled by on-off balance control strategy has better performance and more energy regeneration than the passive electromagnetic damper.

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