

Free Transverse Vibration Analysis of an Underwater Launcher Based on Fluid-Structure Interaction

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Abstract: A pneumatic launcher is theoretically investigated to study its natural transverse vibration in water. Considering the mass effect of the sealing cover, the launcher is simplified as a uniform cantilever beam with a top point mass. By introducing the boundary and continuity conditions into the motion equation, the natural frequency equation and the mode shape function are derived. An iterative calculation method for added mass is also presented using the velocity potential function to account for the mass effect of the fluid on the launcher. The first 2 order natural frequencies and mode shapes are discussed in external flow fields and both external and internal flow fields. The results show good agreement with both natural frequencies and mode shapes between the theoretical analysis and the FEM studies. Also, the added mass is found to decrease with the increase of the mode shape orders of the launcher. And because of the larger added mass in both the external and internal flow fields than that in only the external flow field, the corresponding natural frequencies of the former are relatively smaller.

Keywords: underwater launcher; free transverse vibration; natural frequency; mode shape; added mass; fluid-structure interaction

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1 Introduction

A certain form of underwater pneumatic launcher, which uses high-pressure air as the energy source to discharge the weapon out of the launch tube, vibrates in different flow fields during and after the launching process. During the launching process, the launcher vibrates in the external flow field while after the weapon is launched out, the launcher vibrates in both the external and internal flow fields. The study of the free transverse vibration has great significance for further research on the structural response and sound radiation for these types of launchers when subjected to dynamic forces.

Because of the sealing cover at the free end, the vibration

characteristics of the launcher are more difficult to study than those without top structures. Generally, the top component is simplified into a point mass (Auciello, 1996; Posiadala, 1997; White and Heppler, 1995; Wu and Chen, 2003; Salarieha and Ghorashi, 2006). For structures with top point mass which are immersed in water, there is plenty of literature concerned with their vibration. Uscilowska and Kolodziej (1998) studied the free vibration of a partially immersed column with a concentrated mass at the top. Closed form frequencies and eigenfunctions were presented. The parameters' influence on the frequency equations was also discussed. Öz (2003) presented a non-dimensional transverse vibration equation for an Euler-Bernoulli type beam partially immersed and studied the effects of water height, tip mass and water density on the frequencies. Wu and Chen (2005) developed a theory to determine the natural frequencies and the corresponding mode shapes of an immersed and elastically fixed wedge beam. A comparison between the FEM numerical results was made and good agreement was achieved. Zhou (1997) investigated a cantilever beam with a tip mass which was elastically supported. Exact and analytical expressions for eigenfrequencies and mode shapes of a beam were derived. Wu and Hsu (2006 and 2007) studied an immersed uniform beam carrying an eccentric tip mass with elastic and fixed support. Shen (2009) used the finite difference method to investigate the nonlinear free vibration of a compliant beam partially immersed in water.

Little attention has been paid to the structures fully immersed in both external and internal flow fields. In this study, an underwater pneumatic launcher is examined by modeling it into a cantilever beam with a top mass. Not only the external flow field but also the external field and the internal flow field are studied. The natural frequency equation and mode shape function of the transverse vibration are established. An iterative calculation method of added mass is proposed, which gives more precise predictions. Vibration results of the different dimensions are analyzed as examples and the first 2 order natural frequencies and mode shapes are given. In the end, a comparison between the FEM results is made and a good consistency with the natural frequencies and mode shapes is achieved.

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2 Analytical model

Fig.1 shows the sketch of a type of underwater pneumatic launcher. Considering the mass effect of the sealing cover and ignoring its volume influence, the launcher is modeled as a uniform cantilever with a top mass at the free end. Taking into account the fluid-structure interaction with the added mass, the free transverse vibration equation of the beam model may be written in the form:

$$EI \frac{\partial^4 u}{\partial z^4} + m' \frac{\partial^2 u}{\partial t^2} = 0 \quad (1)$$

where EI is the bending stiffness, u is the transverse displacement, m' denotes the effective mass of the beam per unit length in water. $m' = m + \Delta m$, in which m is the structure mass per unit length in air and Δm is the added mass of entrained water per unit length.

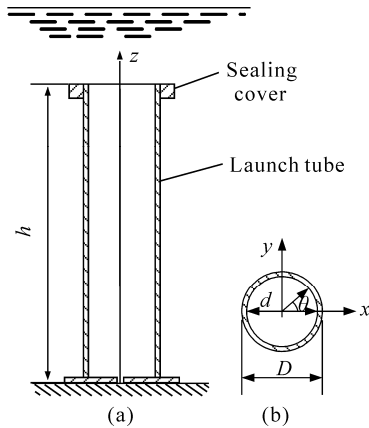
Separating the variables of z and t , the solution of Eq. (1) can be written as:

$$u(z, t) = X(z)T(t) \quad (2)$$

Substituting Eq. (2) into Eq. (1) yields

$$-\frac{\ddot{T}(t)}{T(t)} = a^2 \frac{X''(z)}{X(z)} \quad (3)$$

in which $a^2 = EI / m'$.



(a) front view of the launcher

(b) cross-section of the launch tube

Fig. 1 Sketch of an underwater pneumatic launcher

Noting that the left side of Eq. (3) is a function of t only and the right side is a function of z only, the equation can be satisfied for arbitrary values of t and z only if each side of Eq. (3) is a constant. Supposing the constant is $-\omega^2$, two separate differential equations are derived:

$$\ddot{T}(t) + \omega^2 T(t) = 0 \quad (4.a)$$

$$X''(z) - \frac{a^2}{\omega^2} X(z) = 0 \quad (4.b)$$

The general solutions of Eqs. (4.a) and (4.b) can be expressed in the form:

$$T(t) = A \sin(\omega t + \phi) \quad (5.a)$$

$$X(z) = B_1 \sin(kz) + B_2 \cos(kz) + B_3 \text{sh}(kz) + B_4 \text{ch}(kz) \quad (5.b)$$

where A_1 and $B_1 \sim B_4$ are unknown constants, ω is the natural frequency of the transverse vibration, ϕ is the initial phase angle and k meets the requirements of

$$k^4 = \frac{\omega^2}{a^2} = \frac{\omega^2 m'}{EI}$$

The transverse displacement $u(z, t)$ in Eq. (2) must satisfy the zero displacement and zero slope boundary conditions at the fixed end and the zero moment and shear boundary conditions at the free end, i.e., the conditions:

$$z = 0 : u(z, t) = 0, \frac{\partial u(z, t)}{\partial z} = 0 \quad (6.a)$$

$$z = h : \frac{\partial^2 u(z, t)}{\partial z^2} = 0, M \frac{\partial^2 u(z, t)}{\partial t^2} = EI \frac{\partial^3 u(z, t)}{\partial z^3} \quad (6.b)$$

in which M is the mass of the sealing cover.

Substituting Eqs. (6.a) and (6.b) into Eq. (2) leads to

$$B_2 + B_4 = 0 \quad (7.a)$$

$$B_1 + B_3 = 0 \quad (7.b)$$

$$-B_1 \sin(kl) - B_2 \cos(kl) + B_3 \text{sh}(kl) + B_4 \text{ch}(kl) = 0 \quad (7.c)$$

$$-M\omega^2[B_1 \sin(kl) + B_2 \cos(kl) + B_3 \text{sh}(kl) + B_4 \text{ch}(kl)] = EI k^3 [-B_1 \cos(kl) + B_2 \sin(kl) + B_3 \text{ch}(kl) + B_4 \text{sh}(kl)] \quad (7.d)$$

Writing Eqs. (7.a) ~ (7.d) in a matrix form derives:

$$\begin{bmatrix} C_{11} & C_{12} & C_{13} & C_{14} \\ C_{21} & C_{22} & C_{23} & C_{24} \\ C_{31} & C_{32} & C_{33} & C_{34} \\ C_{41} & C_{42} & C_{43} & C_{44} \end{bmatrix} \begin{Bmatrix} B_1 \\ B_2 \\ B_3 \\ B_4 \end{Bmatrix} = 0 \quad (8)$$

in which $B_1 \sim B_4$ form a column vector. The elements of the coefficients matrix of Eq. (8) are given by:

$$C_{11} = 0, C_{12} = 1, C_{13} = 0, C_{14} = 1, C_{21} = 1, C_{22} = 0, C_{23} = 1, C_{24} = 0$$

$$C_{31} = -\sin(kl), C_{32} = -\cos(kl), C_{33} = \text{sh}(kl), C_{34} = \text{ch}(kl)$$

$$C_{41} = \frac{-Mk}{EI k^3} \sin(kl) + \cos(kl), C_{42} = \frac{-M\omega^2}{EI k^3} \cos(kl) - \sin(kl)$$

$$C_{43} = \frac{-M\omega^2}{EI k^3} \text{sh}(kl) - \text{ch}(kl), C_{44} = \frac{-M\omega^2}{EI k^3} \text{ch}(kl) - \text{sh}(kl)$$

According to the non-zero solution condition and noting $\omega^2 = k^4 EI / m'$, an equation about k is obtained:

$$\frac{M}{m'} = \frac{1}{k} \times \frac{1 + \cos(kl) \text{ch}(kl)}{\sin(kl) \text{ch}(kl) - \cos(kl) \text{sh}(kl)} \quad (9)$$

Considering the relationship between $B_1 \sim B_4$ and solving k from Eq. (9), the mode shape function is also obtained:

$$X(z) = \sin(kz) - \text{sh}(kz) + \frac{\sin(kl) + \text{sh}(kl)}{\cos(kl) + \text{ch}(kl)} [\text{ch}(kz) - \cos(kz)] \quad (10)$$

Eq. (9) is a transcendental equation and has an infinite number of roots. After solving the n -th root of k_n , the n -th natural frequency is then determined:

$$\omega_n = k_n^2 \sqrt{\frac{EI}{m'}} \quad (11)$$

Introducing k_n into Eq. (10), the n -th mode shape of the launcher can also be determined.

3 Added mass

The mass m' in Eqs. (1), (9) and (11) is the mass per unit length considering the effects of the added mass Δm . It is known that Δm is relevant to the parameters including structure geometry, material properties, mode shapes, etc. (Qian *et al.*, 1996; Su and Huang, 2003; Ju and Zeng, 1983). In the previous study of beams with top mass, Δm was usually calculated as the mass of the water displaced by the beam per unit length (Uscilowska and Kolodziej, 1998; Wu and Chen, 2005; Wu and Hsu, 2006 and 2007). This kind of calculation ignores the effects of the differences between mode shapes and cannot be used in the circumstances of structures immersed in both external and internal fluid.

Generally, considering the viscosity of water will make the natural vibration analysis of the structures complicated and there is no significant difference according to non-viscous water. Therefore, the additional inertia effect of the water (i.e., added mass) is mainly taken into account rather than its viscosity. As mentioned in the research done by Ju and Zeng (1983), in irrotational and non-viscous fluid, the velocity potential function ψ of a uniform hollow cylindrical beam satisfies the Laplace equation in cylindrical coordinates.

$$\frac{\partial^2 \psi}{\partial r^2} + \frac{1}{r} \frac{\partial \psi}{\partial r} + \frac{1}{r^2} \frac{\partial^2 \psi}{\partial \theta^2} + \frac{\partial^2 \psi}{\partial z^2} = 0 \quad (12)$$

where r, θ, z are the components in cylindrical coordinates.

At the free surface and the surface adjacent to the stationary structures, the boundary conditions must be satisfied. At the boundary surface between the beam and fluid, the continuity condition is required as follows:

$$r = \frac{D}{2}: \frac{\partial \psi}{\partial r} = \frac{\partial u}{\partial t} \cos \theta; r = \frac{d}{2}: \frac{\partial \psi}{\partial r} = \frac{\partial u}{\partial t} \cos \theta \quad (13)$$

One form of the solution of Eq. (12) can be obtained by assuming ψ in the following form:

$$\psi(r, z, \theta, t) = \cos \theta \sum_{n=1}^{\infty} \dot{q}_n \sum_{s=1,3,5}^{\infty} f(r) \cos \frac{\pi s}{2h} z \quad (14)$$

in which $f(r)$ is a function of radial coordinate r , \dot{q}_n is the generalized velocity. Substituting Eq. (14) into Eq. (12) leads to a modified Bessel function of the first kind:

$$\frac{\partial^2 f(r)}{\partial r^2} + \frac{1}{r} \frac{\partial f(r)}{\partial r} - (1 + \frac{1}{r^2}) f(r) = 0 \quad (15)$$

where $r_1 = \frac{\pi s}{2h} r$.

Noting the velocity continuity indicated as Eq. (13), the function of $f(r_1)$ could be expressed as follows:

$$f(r_1) = B_{ns} I_{1s}(r_1) + C_{ns} K_{1s}(r_1) \quad (16)$$

in which B_{ns} and C_{ns} are constants determined by the cylindrical boundary conditions, $I_{1s}(r_1)$ and $K_{1s}(r_1)$ are modified Bessel functions of the first kind and second kind respectively. According to the characteristics of the modified Bessel function, two series of $I_{1s}(r_1)$ and $K_{1s}(r_1)$ are used to express the velocity potential functions of the internal and external flow field respectively. After superposition, the unified potential function becomes:

$$\psi(r, z, \theta, t) = \cos \theta \sum_{n=1}^{\infty} \dot{q}_n \sum_{s=1,3,5}^{\infty} [B_{ns} I_{1s}(r_1) + C_{ns} K_{1s}(r_1)] \cos \frac{\pi s}{2h} z \quad (17)$$

Expanding the transverse displacement $u(z, t)$ into a series of the product of the underwater mode shape X_n and the generalized coordinate q_n , one obtains

$$u(z, t) = \sum_{n=1}^{\infty} q_n(t) X_n(z) \quad (18)$$

Substituting Eqs. (18) and (17) into Eq. (13), the constants of B_{ns} and C_{ns} are found to be:

$$B_{ns} = \frac{4 \int_0^h X_n \cos \frac{\pi s}{2h} z dz}{\pi s I'_{1s}(\frac{\pi s}{2h} a_2)} \quad (19.a)$$

$$C_{ns} = \frac{4 \int_0^h X_n \cos \frac{\pi s}{2h} z dz}{\pi s K'_{1s}(\frac{\pi s}{2h} a_1)} \quad (19.b)$$

where I'_{1s} and K'_{1s} are first order derivations of the modified Bessel function of the first kind and second kind respectively, $a_1 = D/2$ and $a_2 = d/2$ are the outer and inner radius of the launch tube section respectively.

Substituting Eqs. (19.a) and (19.b) into Eq. (17), the velocity potential function ψ becomes finally

$$\begin{aligned} \psi(r, z, \theta, t) = \cos \theta \sum_{n=1}^{\infty} \dot{q}_n \sum_{s=1,3,5}^{\infty} \left[\left(\frac{I_{1s}(r_1)}{I'_{1s}(\frac{\pi s}{2h} a_2)} + \frac{K_{1s}(r_1)}{K'_{1s}(\frac{\pi s}{2h} a_1)} \right) \times \right. \\ \left. \frac{4}{\pi s} \cos \frac{\pi s}{2h} z \int_0^h X_n \cos \frac{\pi s}{2h} \xi d\xi \right] \end{aligned} \quad (20)$$

Suppose the beam vibrates under a certain frequency ω_n , according to the theorem of kinetic energy of fluid, one obtains

$$T = \frac{1}{2} \rho_w \int_S \psi \frac{\partial \psi}{\partial r} dS \quad (21)$$

in which T is the kinetic energy of the fluid around the beam, ρ_w is water density, S is the fluid boundary.

Noting that $T = \Delta m V^2 / 2$, in which V is the velocity, added mass Δm can be determined as follows:

$$\Delta m_n = \frac{4 \rho_w}{X_n(z)} \sum_{s=1,3,5}^{\infty} \bar{H}_s \cos \frac{\pi s}{2h} z \int_0^h X_n(\xi) \cos \frac{\pi s}{2h} \xi d\xi \quad (22)$$

in which Δm_n is the added mass per unit length of the n -th

free transverse vibration, ξ is the axial coordinate. \bar{H}_s is expressed as follows:

$$\bar{H}_s = \frac{1}{s} \left[a_2 \left(\frac{I_{1s}(r)}{I_{1s}(r)} \right)_{r=a_2} + a_1 \left(\frac{K_{1s}(r)}{K_{1s}(r)} \right)_{r=a_1} \right] \quad (23)$$

Δm_n could not be directly determined yet by Eq.(22) because the wet mode shape function X_n is still unknown. A feasible method to solve the problem is to substitute the wet mode shape with the dry mode shape (Ju and Zeng, 1983). It can be deduced that this type of substitution method for beams without point mass could be extended to beams with top mass. Further more, if an iterative method is used more precisely, a prediction will be reached. As for the launcher model studied in this work, first, a dry mode shape is used to give "rough" results of Δm_n , ω_n and X_n through Eqs. (22), (11) and (10) respectively. Then, with the "rough" wet mode shape X_n , a relatively more precise Δm_n can be calculated, as well as ω_n and X_n . By repeatedly using this iteration, a precise transverse response can be achieved. The flowchart of the iterative method is shown in Fig.2. The analytical computation mentioned above is realized with MATLAB.

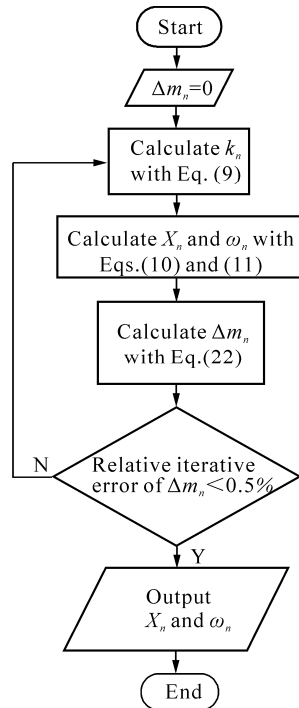


Fig. 2 Flowchart of the iterative calculation

4 Results and discussion

In order to check the validation of the theoretical analysis derived in this work, a 3D FEM calculation is conducted using ANSYS. The axisymmetric plane of the FSI (fluid structure interaction) model is shown in Fig.3. The element type SOLID45 is used to model the structure and FLUID30 is used to simulate the fluid medium. Interfaces between the structure and fluid are specified as the FSI coupling surfaces

in ANSYS. The fluid field in the simulation is cylinder shaped, with its radius 6 times as large as the outer diameter of the launch tube and its length 2 times as long as that of the launcher. The structure model is meshed into 5 elements in thickness direction and 100 elements in length direction. Such fluid region size and element mesh density are considered as meeting the requirements of calculation accuracy (Huang *et al.*, 2001; Yang *et al.*, 2009). The main parameters used in the calculation are as follows: the fluid density is 1,000 kg/m³, the sonic velocity underwater is 1,400 m/s, the structure density is 7,850 kg/m³, Yong's modulus and Poisson ratio are 200 GPa and 0.3 respectively. Nodes at the bottom surface of the launcher are fixed in displacements. The pressure of the outer surface of the fluid is set to zero. The normal displacement of the fluid at the bottom surface is also set to zero.

Different dimensions of h , D and d are discussed as examples in ANSYS, with the sealing cover's diameter 4 cm larger than D and the height 4 cm in length.

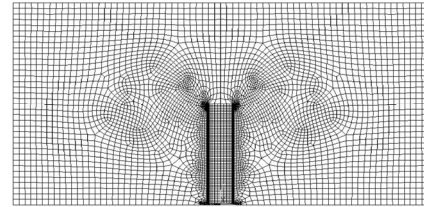


Fig. 3 FSI model of the launcher

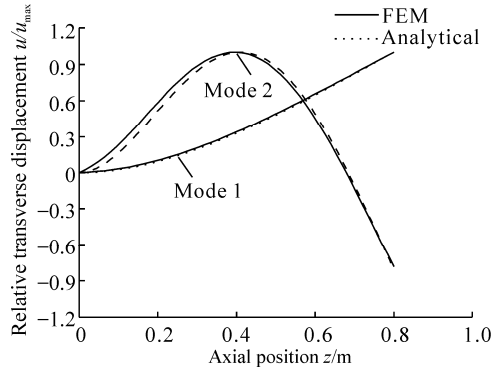
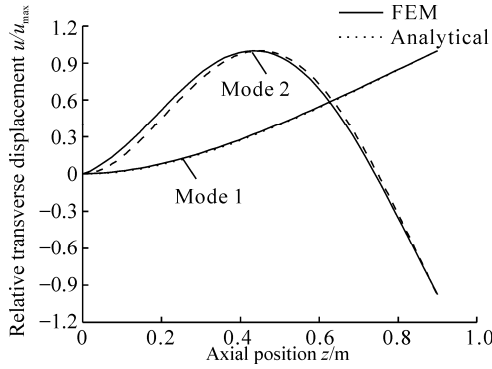
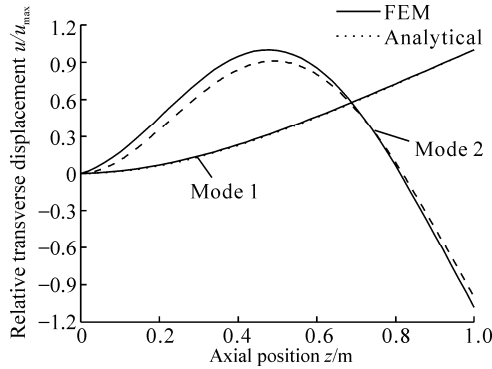
4.1 External flow field

The first 2 order numerical natural frequencies are given in Table 1, and compared with the analytical results and the traditional method (i.e., added mass is regarded as the mass of water displaced by the structure per unit length).

Table 1 First 2 order natural frequencies in external flow field

| h/m | D/m | d/m | Analytical | | FEM | | Traditional | |
|-------|-------|-------|------------|----------|----------|----------|-------------|----------|
| | | | f_1/Hz | f_2/Hz | f_1/Hz | f_2/Hz | f_1/Hz | f_2/Hz |
| 0.8 | 0.10 | 0.07 | 107.6 | 639.7 | 108.5 | 667.7 | 103.6 | 599.7 |
| 0.8 | 0.10 | 0.08 | 105.5 | 631.6 | 105.7 | 660.0 | 100.4 | 588.3 |
| 0.8 | 0.10 | 0.09 | 94.1 | 581.3 | 93.2 | 596.0 | 87.8 | 529.2 |
| 0.9 | 0.11 | 0.07 | 95.4 | 563.4 | 96.3 | 591.1 | 92.9 | 532.8 |
| 0.9 | 0.11 | 0.08 | 95.8 | 565.2 | 96.5 | 586.5 | 93.0 | 535.7 |
| 0.9 | 0.11 | 0.09 | 93.6 | 554.6 | 93.8 | 578.5 | 90.0 | 524.0 |
| 1.0 | 0.12 | 0.07 | 84.8 | 500.5 | 85.8 | 523.1 | 83.4 | 475.9 |
| 1.0 | 0.12 | 0.08 | 86.0 | 505.3 | 86.9 | 530.5 | 84.4 | 504.3 |
| 1.0 | 0.12 | 0.09 | 86.2 | 505.5 | 86.8 | 524.0 | 84.4 | 503.8 |

It can be seen from Table 1 that the analytical frequencies have a better consistency with the FEM results, with a maximum relative error of -4.75% , less than -6.53% of the traditional results. The mode shapes of the first 2 order natural frequencies are shown in Fig.4(a)-(c). Good agreements are also achieved with the curves between the analytical and FEM results.

(a) $h = 0.8$ m, $D = 0.10$ m, $d = 0.09$ m(b) $h = 0.9$ m, $D = 0.11$ m, $d = 0.09$ m(c) $h = 1.0$ m, $D = 0.12$ m, $d = 0.09$ m**Fig. 4 The first 2 order mode shapes in the external flow field obtained from analytical and FEM solutions**

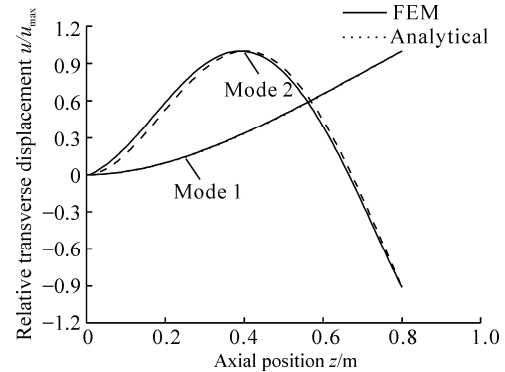
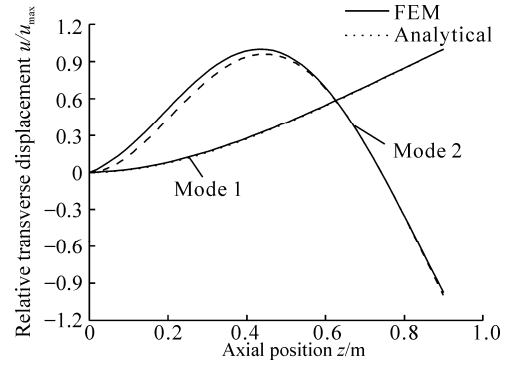
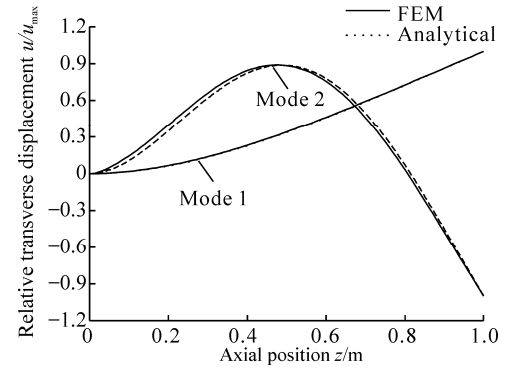
4.2 External and internal flow field

The flow of the fluid in both external and internal cases is also considered in ANSYS by setting both the outer and inner surface of the launcher as FSI surfaces. Natural frequencies, as well as the mode shapes, are given in Table 2 and Fig. 5(a)-(c) respectively, from which good agreements can be seen. The maximum relative error rate of the first 2 order natural frequencies is about 6.81%.

Due to the relatively larger added mass of the launcher in both the external and internal flow field, the natural frequencies are generally smaller than those obtained in only the internal flow field.

Table 2 The first 2 order natural frequencies in both the external and internal flow field

| h/m | D/m | d/m | Analytical | | FEM | |
|-------|-------|-------|------------|----------|----------|----------|
| | | | f_1/Hz | f_2/Hz | f_1/Hz | f_2/Hz |
| 0.8 | 0.10 | 0.07 | 102.2 | 634.1 | 103.3 | 610.0 |
| 0.8 | 0.10 | 0.08 | 97.6 | 607.5 | 99.0 | 585.1 |
| 0.8 | 0.10 | 0.09 | 83.2 | 513.7 | 85.1 | 508.0 |
| 0.9 | 0.11 | 0.07 | 91.3 | 565.2 | 92.1 | 542.4 |
| 0.9 | 0.11 | 0.08 | 90.3 | 564.1 | 91.2 | 534.8 |
| 0.9 | 0.11 | 0.09 | 85.9 | 528.1 | 87.0 | 509.8 |
| 1.0 | 0.12 | 0.07 | 81.8 | 514.6 | 82.3 | 484.9 |
| 1.0 | 0.12 | 0.08 | 82.0 | 516.7 | 82.5 | 483.8 |
| 1.0 | 0.12 | 0.09 | 80.7 | 495.8 | 81.4 | 475.1 |

(a) $h = 0.8$ m, $D = 0.10$ m, $d = 0.09$ m(b) $h = 0.9$ m, $D = 0.11$ m, $d = 0.09$ m(c) $h = 1.0$ m, $D = 0.12$ m, $d = 0.09$ m**Fig. 5 The first 2 order mode shapes in both the external and internal flow field obtained from analytical and FEM solutions**

An examination of Fig.4 and Fig.5 indicates that the consistency of the mode shapes of the first order is better than the second order. This is probably due to the greater similarity of the first order dry mode shape to the real wet mode shape in the iteration calculation for the first time.

4.3 Effects of modes on added mass

According to the proceeding derivation of added mass, Δm_n is not a constant but a function determined by the

orders of vibration modes and the geometry of the launcher. Table 3 gives the added mass of the first 3 modes for the launcher in different flow fields.

It can be seen from Table 3 that the added mass decreases with the increasing of the mode orders of the launcher. For the same order of vibration mode, the added mass in both the external and internal flow field is relatively larger than that of only the external flow field.

Table 3 Added mass per unit length in different flow fields of the launcher

| l/m | D/m | d/m | External flow field | | | External and internal flow field | | |
|-----|------|------|------------------------|------------------------|------------------------|----------------------------------|------------------------|------------------------|
| | | | $\Delta m_1/\text{kg}$ | $\Delta m_2/\text{kg}$ | $\Delta m_3/\text{kg}$ | $\Delta m_1/\text{kg}$ | $\Delta m_2/\text{kg}$ | $\Delta m_3/\text{kg}$ |
| 0.8 | 0.10 | 0.07 | 11.46 | 9.15 | 7.18 | 15.56 | 12.83 | 10.89 |
| 0.8 | 0.10 | 0.08 | 11.48 | 8.80 | 7.39 | 16.85 | 13.72 | 12.29 |
| 0.8 | 0.10 | 0.09 | 11.53 | 8.91 | 7.54 | 18.36 | 15.11 | 13.65 |
| 0.9 | 0.11 | 0.07 | 13.72 | 10.46 | 9.13 | 17.82 | 14.27 | 12.90 |
| 0.9 | 0.11 | 0.08 | 13.74 | 10.78 | 9.86 | 19.08 | 16.53 | 13.73 |
| 0.9 | 0.11 | 0.09 | 13.76 | 10.73 | 7.92 | 20.54 | 16.94 | 13.89 |
| 1.0 | 0.12 | 0.07 | 16.20 | 12.60 | 11.61 | 20.30 | 16.38 | 13.50 |
| 1.0 | 0.12 | 0.08 | 16.21 | 12.52 | 11.38 | 21.55 | 17.47 | 16.40 |
| 1.0 | 0.12 | 0.09 | 16.22 | 12.25 | 10.71 | 22.98 | 18.61 | 16.89 |

5 Conclusions

In this research, the characteristics of the transverse vibration of an underwater launcher in two different flow fields are presented. First, the launcher is modeled as a cantilever beam with a point mass at the top by simplifying the sealing cover into a concentrated mass. The natural frequency equation and the mode shape function are then derived by taking the boundary conditions into account in the motion equation. Then, an iterative calculation method of added mass is proposed to give a more precise prediction of the vibration response. Two cases of flow field are discussed. The numerical results given by FEM are used to validate the analytical theory and results. Good agreements are obtained with the natural frequencies and mode shapes. For the flow field in both the external and internal cases, due to the relatively larger added mass, the natural frequencies are relatively smaller than with those only in the external flow field. The research involved with this work may be helpful in future studies of the dynamic response and sound radiation for underwater launchers when subjected to transient actions.

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30th Symposium on Naval Hydrodynamics

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