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THE CONSTRUCTIVE, DYNAMIC AND TECHNOLOGICAL PERFORMANCE OF ANCHORS WITH STRANDS FOR STRENGTHENING THE FOUNDATION SYSTEM OF BUILDINGS

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ABSTRACT

The paper presents the special joint systems type anchor - concrete. The study was conducted with the hypothesis according to which the steel anchors are provided through longitudinal cuts in the form of "fringe" as segments evenly arranged at the end of the joint.

These are expanded by the initial assembly through an inner cone extracted during the fixation process. Steel anchors have linear elastic behavior, both on axial direction and the lateral direction the vertical direction of the commit request. The anchor-concrete interface separation strength is provided only by axial pull friction force without a slipping or sliding trend.

KEYWORDS: anchor, strands, dynamic, concrete, elastic behavior

1. INTRODUCTION

For fastening, assembling and installing of industrial, power and processing equipments operation in dynamic mode, there are often used highly efficient expandable steel anchors systems

It achieves longitudinal expansion of systems located at one end of the anchor, when the fastening bolt pulls out one truncated end. This one is larger in diameter than the cylindrical anchor inside. Segmented elements by expanding and plastic deformation get a secure mounting hole formed in the hardened concrete.

Fixing at the anchor end is based on sufficient increase of frictional force $Ff \leq \mu N$, where μ is the coefficient of frictional force at slip, and N is the resultant of all the pressing forces Ni on the surface of the concrete segments, i.e. $N = \Sigma Ni$.

Dynamic and fatigue behavior of fastening systems with anchors that are either elastically axial or transversely deformed show clearly the dry friction process.[1]

This study aims to highlight the behavior of a joint system with a steel anchor, fixed in concrete by expanding, under unidirectional dynamic loads with force $F(t)=m_0 r\omega^2 \sin\omega t$.

Based on the dynamic model and the experimental results, the characteristics expressed by the fastening forces $Q = kx + \mu N \mathrm{sign}(\dot{x})$ and operating modes can be evaluated.

2. EXPERIMENTAL SYSTEM SCHEMATIZATION

Figure 1 is a schematic drawing of the expanded steel anchors one end 1, with a central bolt M20, length 25 cm, and the free end 2 is provided with a fastening system of the vibrator 3 provided with two eccentric masses involved and which generate a disruptive force only in the axial direction. The values of physical quantities are: E = 2.1 · 105 N/mm2, l = 25 cm, d = 2 cm for bolt.

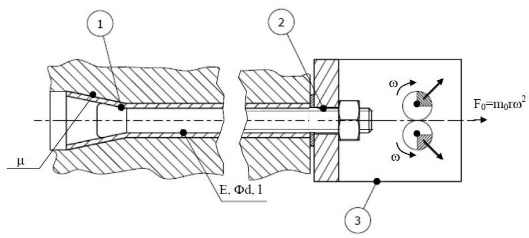


Fig 1 Expanded steel anchor

The full dynamic linear elastic model with parameters m, k, c_{ech} is driven by the force $F = F_0 \sin \omega t$, where the amplitude of the force is dependent on the angular velocity ω so $F_0(\omega) = m_0 r \omega^2$, where $m_0 r$ is the total static moment of unbalanced masses in rotation with ω. Figure 2 shows the model with standard notation.

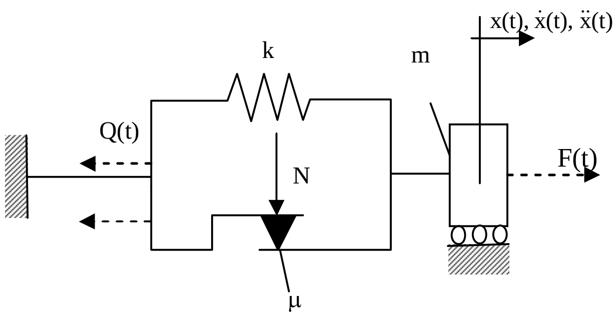


Fig 2 Dynamic model

3. DYNAMIC BEHAVIOUR OF HARMONIC FORCE APPLIED AXIALLY

The differential equation of motion can be written as:

$$m\ddot{x} + \mu N \operatorname{sign}\dot{x} + kx = F_0 \sin\omega t \qquad (1)$$

with stationary vibration solution as the right member, of the form:

$$x = A\sin(\omega t - \varphi) \qquad (2)$$

The dry friction force $F_0 = -\mu n = \mu N \operatorname{sign}(\dot{x})$ has the ability to dissipate energy and compared with equivalent linear viscous damping it can be replaced by equivalent force $F_v = c_{ech}\dot{x}$. In this case the energy dissipated per cycle in both cases has to be the same [2], ie:

$$\Delta W_d^f = \Delta W_d^v \qquad (3)$$

The energy dissipated by dry friction ΔW_d^f can be determined as follows:

$$\Delta W_d^f = \int_0^t \mu N \dot{x} \operatorname{sign}\dot{x}\, dt \qquad (4)$$

where:

$$\dot{x} = \omega A \cos(\omega t - \varphi) = \omega A \cos[(\omega t - \varphi) - \pi].$$

Let us mark $\omega t - \varphi = u$, so $du = \omega dt$, so that $\dot{x} = \omega A \cos(u - \pi)$.

We change variable $u - \pi = \alpha$ and we have $du = d\alpha$, and the limits of integration become $u_1 = \omega \cdot 0 - \varphi = -\varphi$, for t=0, in which case $\alpha_1 = -\varphi - \pi$, $u_2 = \omega T - \varphi = 2\pi - \varphi$ for t=T, and $\alpha_2 = 2\pi - \varphi - \pi$ or $\alpha_2 = \pi - \varphi$. In this case we have:

$$\Delta W_d^f = \int_0^{2\pi-\varphi} \pi N \dot{x} \operatorname{sign}[\cos(\omega t - \varphi)]dt \qquad (5)$$

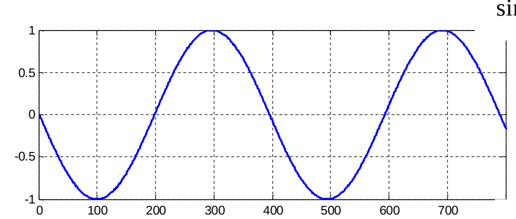
or

$$\Delta W_d^f = \int_{-\varphi}^{2\pi-\varphi} \mu N \omega A \cos u \operatorname{sign}[\cos u] \frac{du}{\omega} \qquad (6)$$

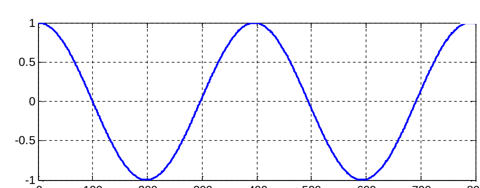
In Figure 3 there are presented the curves x, $\dot{x}$ and F_f time domain and the function sign [cosu] signifies the opposite change of speed $\dot{x} \cdot \operatorname{sign}\dot{x}$ with the frictional force μN. This could be written:

$$\cos u \cdot \operatorname{sign}[\cos u] = \cos(u - \pi) = -\cos u$$

sin u(t)



cos u(t)



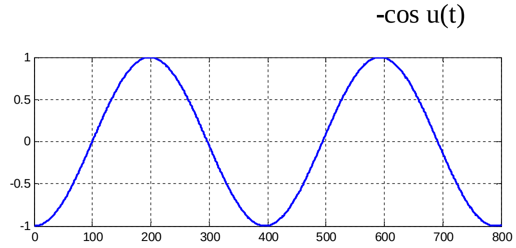


Fig 3 Curves x, $\dot{x}$ and F_f

So relation (6) can be written:

$$\Delta W_d^f = \int_{-\varphi}^{2\pi-\varphi} \mu NA\cos(u-\pi)du = \\ =4\int_{-\varphi}^{\frac{\pi}{2}-\varphi} \mu NA\cos(u-\pi)du \quad (7)$$

$$\Delta W_d^f = \int_{-\varphi}^{2\pi-\varphi} \mu NA\cos(u-\pi)du = 4\int_{-\varphi}^{\frac{\pi}{2}-\varphi} \mu NA\cos(u-\pi)du$$

Passing to the variable $\alpha = u - \pi$, we have:

$$\Delta W_d^f = 4\mu NA\int_{-\pi-\varphi}^{-\frac{\pi}{2}-\varphi} \cos\alpha d\alpha = 4\mu NA\sin\alpha$$

or

$$\Delta W_d^f = 4\mu NA\left[\sin\left(-\frac{\pi}{2}-\varphi\right) - \sin(-\pi-\varphi)\right] \quad (8)$$

where finally we obtain:

$$\Delta W_d^f = -4\mu NA(\sin\varphi + \cos\varphi \quad (9)$$

where φ is the phase difference between the instant movement x(t) and force F(t).

The dissipated energy in viscous element with constant c, for a stationary stable regime with pulsation ω of the forced regime, with $\omega \neq \omega_n$, is [4]:

$$\Delta W_d^v = \pi c\omega A^2 \quad (10)$$

so the relation (3) becomes:

$$4\mu NA(\sin\varphi + \cos\varphi) = \pi c\omega A^2$$

from where

$$c = \frac{4}{\pi}\mu N(\sin\varphi + \cos\varphi)\frac{1}{\omega A(\omega)} \quad (11)$$

Taking into account that only for resonance condition:

$$\omega = \omega_n = \sqrt{\frac{k}{m}},\ A_{max} = \frac{F_0}{c\omega_n},\ \varphi = \frac{\pi}{2}$$

viscous force equilibrium is showed with disruptive force, we have:

$$c = \frac{4\mu N(1+0)}{\pi\omega A_{max}} = \frac{4}{\pi}\frac{\mu N}{\omega A_{max}} \quad (12)$$

where $A_{rez} = A_{max}(\omega)$ with $\omega = \omega_n \pm \varepsilon$, i.e. near the resonance.

Knowing $c_{cr} = 2m\omega_n$, we obtain the equivalent critical damping fraction ξ_{eq}, so

$$\xi_{eq}^{max} = \frac{c}{c_r} = \frac{4\mu N}{\pi\omega_n A_{max} 2m\omega_n}$$

or

$$\xi_{eq}^{max} = \frac{2}{\pi}\mu\frac{N}{k}\frac{1}{A_{max}} \quad (13)$$

If friction forces μN generate the deformation Δ of elastic element k, then $\Delta = \frac{\mu N}{k}$, and (13) can be written:

$$\xi_{eq}^{max} = \frac{2}{\pi}\frac{\Delta}{A_{max}} \quad (14)$$

Outside of resonance regime, i.e. for $\omega \neq \omega_n$, amplitude $A = A(\omega) \neq A_{rez}$, and critical damping fraction ξ depends on the excitation pulsation ω, as follows:

$$\xi_{eq}(\omega) = \frac{c}{c_{cr}} = \frac{4\mu N}{\pi\omega A}\frac{1}{2m\omega_n} \quad (15)$$

or

$$\xi_{eq}(\omega) = \frac{2}{\pi}\frac{\mu N}{\sqrt{km}}\frac{1}{\omega A(\omega)} \quad (16)$$

The differential equation equivalent to the equation (1) is of the form:

$$m\ddot{x} + c_{eq}\dot{x} + kx = F_0\sin\omega t \quad (17)$$

With solution of forced vibration:

$$x(t) = A\sin(\omega t - \varphi) \qquad (18)$$

where A and φ is determined from the condition of checking the differential equation (16). Thus, by replacing x, (x) and x ̈ in (17) we obtain:

$$A = F_0 \frac{1}{\sqrt{(k - m\omega^2)^2 + c^2\omega^2}}$$

or

$$A = \frac{F_0}{k} \frac{1}{\sqrt{(1-\omega^2)^2 + (2\xi\frac{\omega}{\omega_n})^2}} \qquad (19)$$

where

$$2\xi\frac{\omega}{\omega_n} = 2\frac{2}{\pi}\frac{\mu N}{\omega\omega_n mA(\omega)} \cdot \frac{\omega}{\omega_n} = \frac{4\mu N}{\pi\omega_n^2 mA(\omega)}$$

or

$$2\xi\frac{\omega}{\omega_n} = \frac{4\mu N}{\pi k A(\omega)} = \frac{\Delta_0}{A(\omega)} \qquad (20)$$

where $\xi\frac{\omega}{\omega_n} = \xi_{eq}$, iar $\Delta_0 = \frac{4\mu N}{\pi k}$, so we have

$$\xi_{eq} = \frac{1}{2}\frac{\Delta_0}{A(\omega)}$$

where Δ_0 is the equivalent deformation given by the dry friction force $\frac{4}{\pi}\mu N$.

Equation (19) can be written as:

$$A(\omega) = \frac{F_0}{k} \frac{1}{\sqrt{(1-\omega^2)^2 + \frac{\Delta_0^2}{A^2(\omega)}}} \qquad (21)$$

where we finally obtain:

$$A(\omega) = \frac{F_0}{k}\frac{1}{1-\Omega^2}\sqrt{1 - \frac{\Delta_0^2 k^2}{F_0^2}} \qquad (22)$$

with motion condition A>0, i.e.

$$H = 1 - \frac{\Delta_0^2 k^2}{F_0^2} > 0$$

Where $\frac{\Delta_0}{F_0} = \pm 1$, and function H>0 corresponds to $-1 < \frac{\Delta_0}{F_0} < +1$.

The physical sense corresponds only for $0 < \frac{\Delta_0 k}{F_0} < +1$, where from $\Delta_0 k < F_0$. Finally, the condition of physical sense is:

$$\frac{4\mu N}{\pi k} \cdot k < F_0$$

or

$$\mu N < \frac{\pi}{4} F_0 \qquad (23)$$

Phase difference φ results:

$$\varphi = \mathrm{arctg}\frac{1}{1-\Omega^2}\frac{4\mu N}{\pi k A(\omega)}$$

It replaces $A(\omega)$ with relation (22) and we obtain:

$$\varphi = \mathrm{arctg}\frac{\pm\frac{\Delta_0 k}{F_0}}{\sqrt{1-\left(\frac{\Delta_0 k}{F_0}\right)^2}} \qquad (24)$$

Under condition $H = 1 - \left(\frac{\Delta_0 k}{F_0}\right)^2 > 0$, where from $\Delta_0 k < F_0$ or $\mu N < \frac{\pi}{4} F_0$.

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