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Suppression of Flow Induced Vibrations by Self Excited Oscillations of Two Hinged Plates

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Abstract: The study reported in this paper investigates the feasibility of utilizing the fluid energy,which is causing the vibrations, to suppress the flow induced vibration of a mechanical structure. A cantilever beam with uniform thickness and variable tip mass subjected to harmonic excitation is considered to simulate the oscillating structure (the primary system). Two hinged plates, as a secondary system, are attached at its free end to provide the flow induced dynamic suppression by tuning their characteristic vibration to the oscillation of the primary system. A computer program was developed to evaluate the optimum design parameters of the secondary system (absorber) at any flow speed for a given structure. Experimental verification of the response with the calculated optimum absorber at each flow speed was undertaken. The results show a good correlation between the predicted and the experimentally measured suppressed amplitudes of the primary system.

Nomenclature

X_{rw} resonant vibration amplitude of the main system with the absorber attached
X_m resonant vibration amplitude of the main system without the absorber
M_{es} generalized mass of the structure corresponding to the first mode
M_a total mass of the plates
t plate thickness
H plate height
ρ_{plate} plate material density
ρ_f fluid density
I_t total mass moment of inertia of the plates about the primary axis
R length ratio (L_2 / L_1)
μ mass ratio (M_a / M_{es})

ς damping ratio of the structure (first mode)
f_{ns} natural frequency of the structure (first mode)
X maximum vibration amplitude of the structure
U uniform flow speed
U_{ce} critical speed at which the plates start oscillating
U_r Reduced flow velocity ($U/f_{ns} L$)
f_{er} self excited frequency of the hinged plates on a rigid support
S Strouhal number
L_{1opt}, L_{2opt} optimum length of the primary and the secondary plates respectively

Introduction

Flow induced vibration is encountered in many mechanical structures. This has stimulated many scientists and engineers for the past three decades to explore the main causes and mechanisms that are responsible for the phenomenon. Numerous authors treated the suppression of self excited vibrations caused by winds and ocean waves. Some suggestions have been proposed for controlling the flow field surrounding the structure by streamlining the cross- sectional shape or/and incorporating externally powered rotating cylinders to be placed at specific locations. Others suggested the installation of some devices such as helical strakes, splitter and spoiler plates and guiding vanes on the structure in order to suppress the flow-induced vibration by breaking the organized vortex street in the wake region of the structure. Zdravkovich [1], has summarized the literature related to various aerodynamic and hydrodynamic means for suppressing vortex shedding. These devices, however, can lead to considerable increase of the drag coefficient C_d (up to 50%, for Re = $[10^3 - 10^5]$) compared to the plain cylinder Blevins[2]. Recently, Kubo, Yasuda and Kotsubo [3] developed an active control method for suppressing the aerodynamic vibrations of super-tall structures. They used rotating cylinders at the leading edge to control the boundary layer and hence to improve the flow field around the structure. This, consequently, reduces the drag force and the aerodynamic response of the tall buildings. However, this usually involves complex construction and consequently increases the cost of the system as well as imposing constraints on the architectural design of the structure. A similar approach is suggested by Sandeep and Toshio [4] for drag reduction and vibration control of a D-section using two rotating cylinders at both corners. Another active control of the flutter of a suspension bridge is considered by Kobayashi [5] who used control wings mounted on the bridge deck which are driven so as to produce the aerodynamic forces needed to suppress the bridge motion. It should be noted that a feedback between the bridge oscillations and the control wings has to be established.

Tondl [6], considered attaching a pendulum, which does not interact with the flow, to a cylinder in a cross-flow. The cylinder is excited by the vortex shedding formed behind it. It was shown that a tuning coefficient,defined by $r = \omega_{pendulum} / \omega_{cylinder}$, of about 2 results in diminishing the vibrations of the cylinder.

This paper investigates the feasibility of using the aerodynamically activated two hinged plates as a vibration absorber which can minimize the flow-induced vibration of the structure (the main system).

The first mathematical treatment of the passive linear dynamic vibration absorber attached to an undamped main system was introduced by Ormondroyd and Den Hartog [7]. It was found that an optimum tuning frequency ratio, f, can be described in the following form:

$$f = \frac{1}{1+\mu}$$

where f is the ratio of the uncoupled natural frequencies (frequency of the absorber to the frequency of the main system) and μ is the mass ratio. Note that the optimum tuning frequency of this absorber is a function of the mass ratio. It should be noted that the mass ratio μ is usually constrained by the maximum allowable displacement of the absorber mass and there is no optimum mass ratio. In other words, the larger the mass ratio the better the performance of the absorber, since μ will determine the effective operating speed range. The proposed absorber (the hinged plates) has a mass M_a and variable nonlinear stiffness due to the nonlinear restoring fluid forces. The main system is represented by a mass M_{es}, a linear stiffness, K_s, and viscous damping with a damping ratio ς.

Description of the system

Figure 1 shows a schematic diagram of the proposed absorber (the two hinged plates). In this model the stiffness of the absorber is not constant but is a function of the wind speed, density and the geometry of the plates. It should be mentioned that the vibration of the hinged plates takes place in the horizontal plane; therefore, gravity is not considered as a restoring force in this case.

The uncoupled self excited frequency of the hinged plates (i.e. before attaching the absorber to the system) and the coupled self-excited frequency of the plates attached to the structure are designated as f_{er} and f_{ee} respectively.

The flow-induced vibrations of the main system is assumed to be caused by a vortex shedding mechanism. In practice it is noticed from the signature of the spectrum of an oscillating bluff structure due to external flow, that the first peak in the vibrations is usually caused by vortex shedding and further increase in the wind velocity can produce other types of instabilities, like galloping and flutter. Specially, in the first mode oscillation, large amplitudes of up to 40% of the model length scale (L) can be produced with reduced velocity, U_r, between approximately 1 and 9.

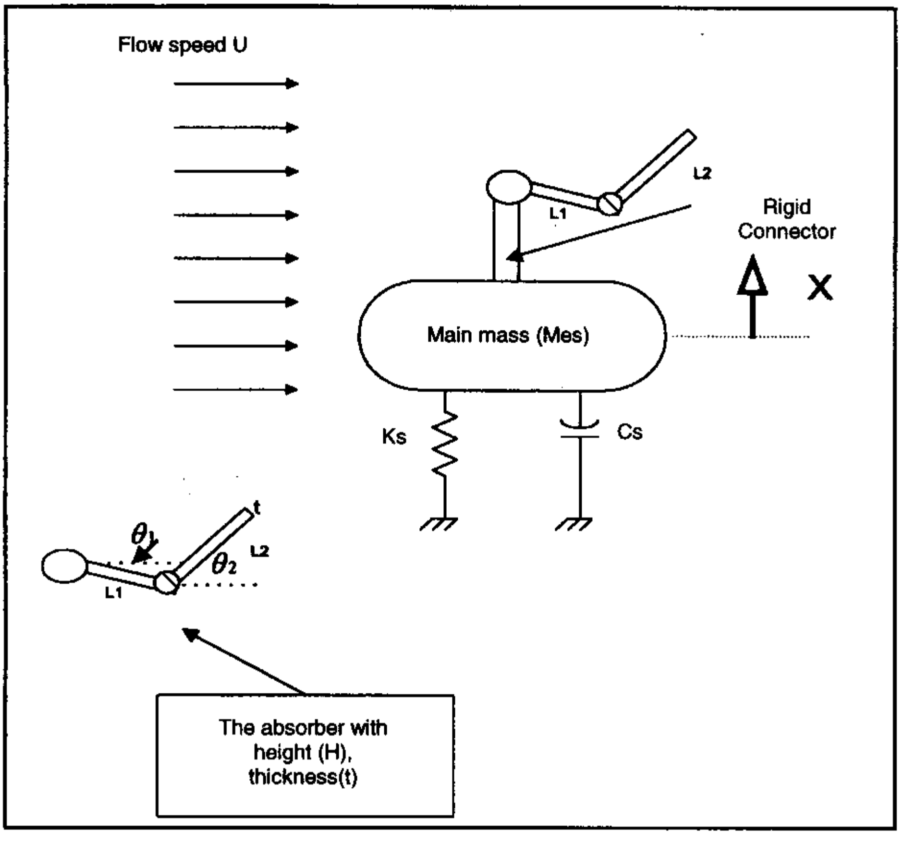

Fig. 1. The hinged plates (as an absorber) attached to the main system.

Experimental Procedure and Results for The Tuning Process

The experimental procedure conducted in this study for tuning the absorber to minimize the flow induced vibrations of the main system can be described as follows:

A given absorber (two hinged plates) was attached to the main system (with a given natural frequency f_{ns}) and the combined system was excited to resonance using an electromagnetic shaker as shown in Fig. 2. The wind speed was then allowed to vary, starting from the critical speed U_{ce}. The maximum amplitude of oscillation of the main system X was then recorded at each wind speed and analyzed in the frequency domain. The value of the wind speed at which the amplitude X is minimum was obtained. This wind speed gives the optimum conditions for the given system. Using this speed, the uncoupled self-excited frequency f_{er} (i.e. the absorber on a rigid support) of the hinged plates can be calculated from the eqdeveloped in Tang [8]. Normalizing this frequency by the natural frequency of the main system, a tuning frequency ratio can be calculated. This frequency ratio was checked for different systems by repeating the same procedure for different mass ratios (μ) and natural frequencies (f_{ns}). In the conducted experiments, the mass ratio was changed using adjustable weights at the tip of the cantilever as shown in Fig. 2. For the case of mass ratio $\mu = 0.2$ and main system's natural frequency $f_{ns} = 0.85$ Hz, the frequency spectrum of the response at different wind speeds was calculated from the time series and the results are given in Figs.3-6. Figure 3 shows the vibration response of the main system x(t) and the hinged plates at wind speed of 6.25 m/s, the amplitude X is approximately one inch (2.54 cm). As the speed increased to 7.45 m/s, the amplitude increased and reached 2.5 in (6.35 cm) as shown in Fig. 4. Further increase of the wind speed showed a reduction of the amplitude X until it reached a very low magnitude of approximately 0.24 inches (0.6 cm) at a wind speed of 10.34 m/s as depicted in Fig. 5. At this wind speed, it is noticed that the coupled self-excited frequency of the hinged plates, f_{ec} = 1.67 Hz, is approximately twice the main system's natural frequency in this case, which is f_{ns} = 0.85 Hz, as can be seen from the spectrum of Fig. 5. The uncoupled self excited frequency of the absorber (f_{er} = 2.5 Hz) is also approximately three times the natural frequency of the main system (f_{ns} = 0.85 Hz). The vibration amplitude X increases again with increasing the wind speed above 10.34 m/s as shown in Fig. 6.

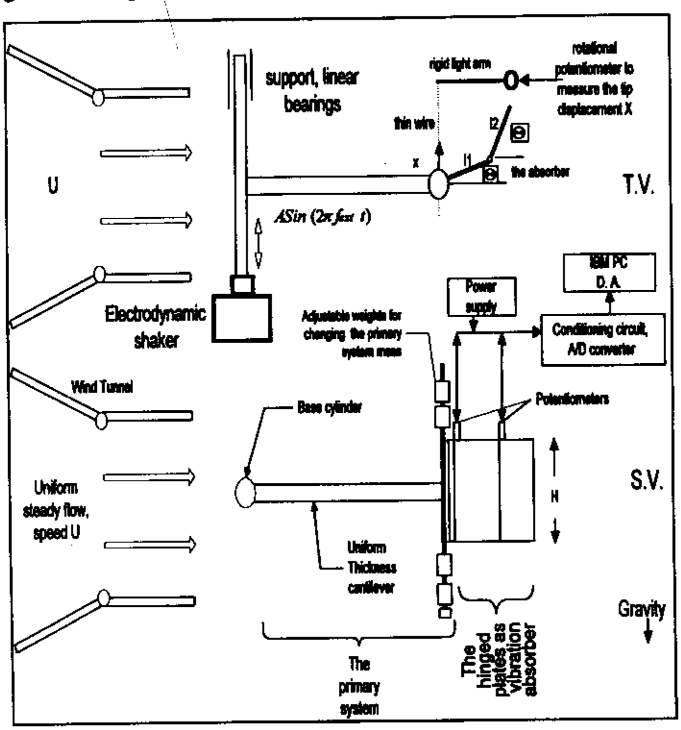

Fig. 2. Experimental setup.

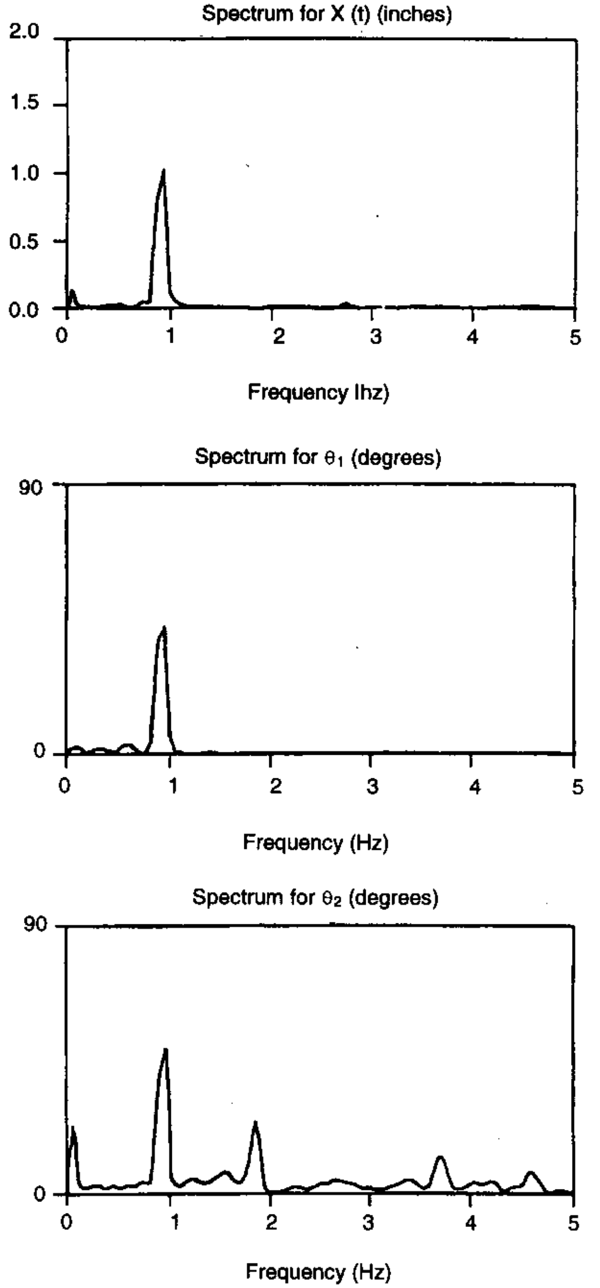

Fig. 3. Frequency spectra of the main system x(t) and the absorber $\theta_1(t)$, $\theta_2(t)$ at U= 6.25 m/s, mass ratio (μ) =.2.

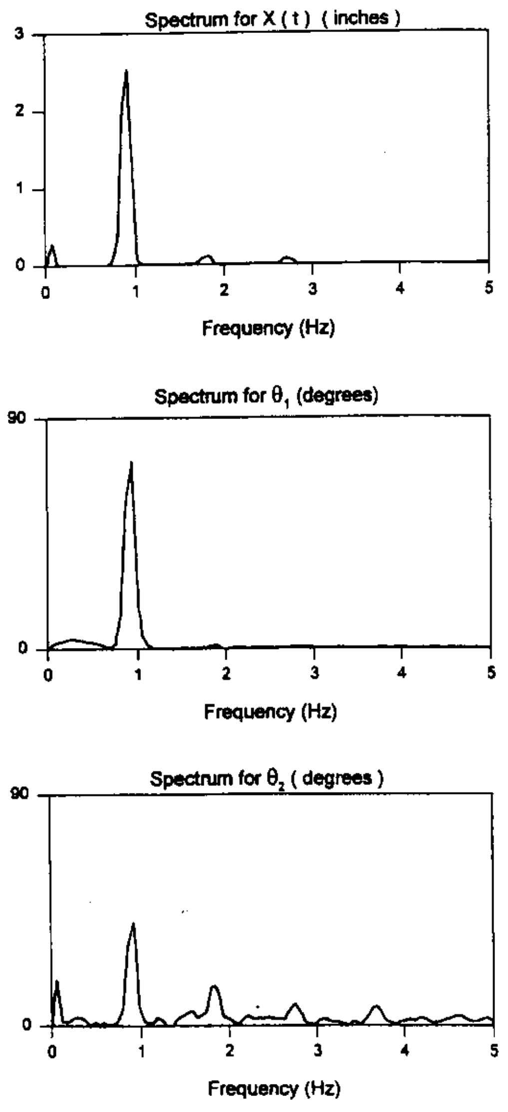

Fig. 4 Frequency spectra of the main system x(t) and the absorber $\theta_1(t)$, $\theta_2(t)$ at U= 7.45 m/s, mass ratio (μ) =.2.

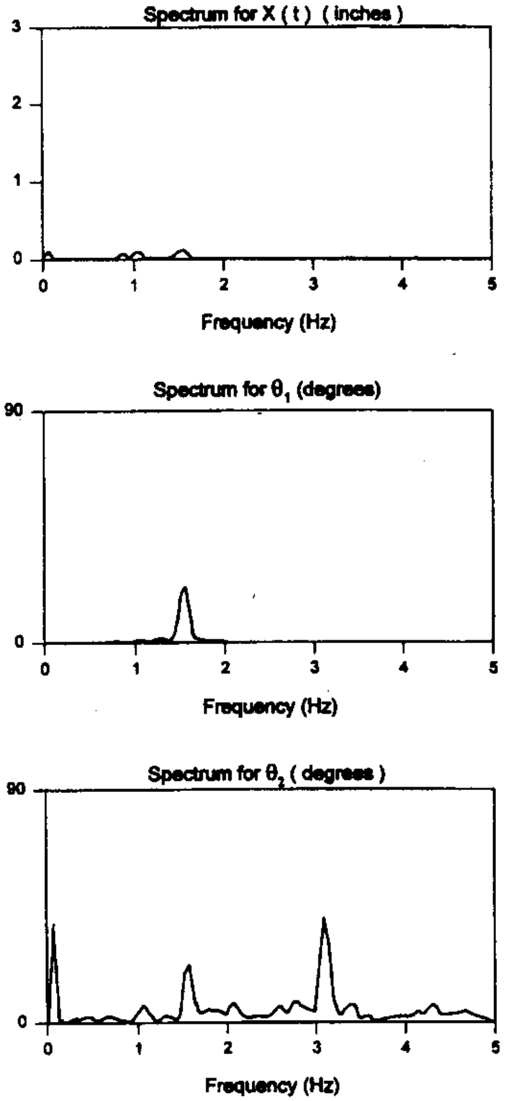

Fig. 5. Frequency spectra of the main system x(t) and the absorber $\theta_1(t)$, $\theta_2(t)$ at U = 10.34 m/s, mass ratio (μ) = 0.2

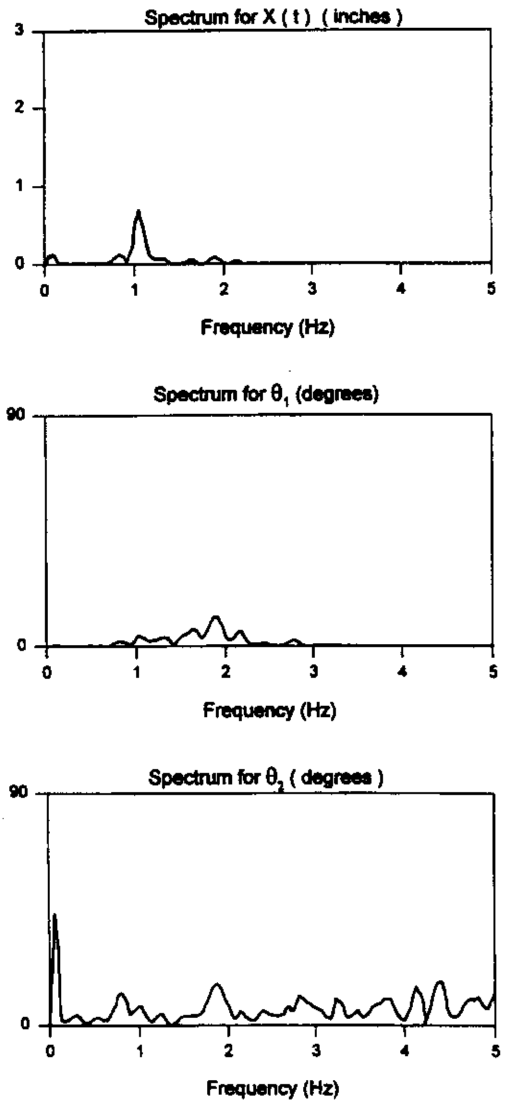

Fig. 6. Frequency spectra of the main system x(t) and the absorber $\theta_1(t)$, $\theta_2(t)$ at U= 12.5 m/s, mass ratio (μ) =.2

Consequently, the results suggest that for each mass ratio and natural frequency of the main system there is a specific wind speed that will bring the vibrations of the structure or the main system to minimum and that there is a certain-tuned frequency ratio (f_{er} / f_{ns}) at which the maximum vibration amplitude (X) will be minimum.

To investigate this observation further, a set of experiments was performed for six different main systems (with different mass ratios and frequencies f_{ns}) with the same absorber parameters. The numerical values of the mass ratios, the natural frequencies and the measured damping ratios of the considered systems are listed in Table 1. The damping ratio of the main system was used to calculate its resonant amplitude value without the absorber X_m. This value will be used to normalize the suppressed amplitude of the main system X_{rw} due to the absorber.

Table 1. Mass ratios, natural frequencies and damping ratios of the tested systems

(μ)	.05	.15	.2	.27	.45	.85
f_{ns} (first mode)	.6 hz	.75 hz	.85 hz	.95 hz	1.12 hz	1.52 hz
Damping Ratio ς	.0043	.0049	.0058	.0063	.0078	.0096

For each main system, an external excitation was applied at the base of the foundation with a frequency close to the natural frequency of the system. This was done to simulate the oscillation at the natural frequency of the main system due to organized trailing vortices in the wake region. The wind speed was then varied until the minimum X was attained.

A sample result is shown in Fig. 7 for the mass ratio 0.15 and natural frequency 0.75 Hz. It can be seen that the minimum amplitude of x(t) is achieved at the frequency ratio of approximately three.

This ratio can be explained by the nonlinearity of the system which is clearly manifested in the self excited response of the plates. The self-excited absorber acts as a nonharmonic disturbing force with significant subharmonic component of order 1/3 f_{ee} generated by the flow excitation. When the absorber and the main system are then combined, the self excited frequency of the absorber should be three times the external excitation frequency, $f_{ee} = 3 f_{ns}$, such that the 1/3 f_{ee} component of the absorber response will produce an internal resonance that will contribute in diminishing the vibration of the main system.

A summary of the results for the considered cases is depicted in Fig. 8 which shows a straight line with slope equal to three. This suggests that the tuning frequency ratio seems to be constant for the different tested systems. However, the minimum amplitude ratio is found to be a function of the mass ratio as summarized in Fig 9. Every point in this figure represents the best tuned dynamic amplitude reduction of the main system for that particular mass ratio. The figure shows that there is an optimum mass ratio ($\mu \sim 0.05$) at which a reduction of approximately 97% of the vibration amplitude of the main mass can be achieved. For mass ratio below 0.05 the amplitude ratio (X_{rw}/X_{rn}) will increase sharply until it reaches one at zero mass ratio (i.e. no absorber is attached).

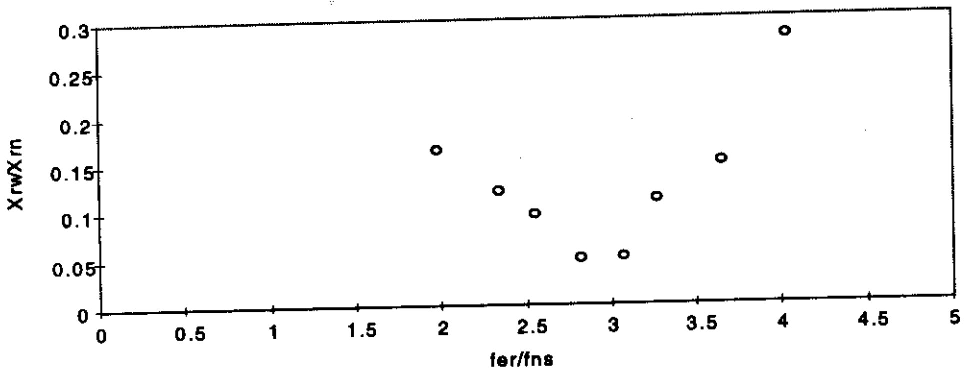

Fig. 7. Amplitude ratio (Xrw/Xrn) vs. frequency ratio (fer / fns), mass ratio =.15, Fns =.75 hz.

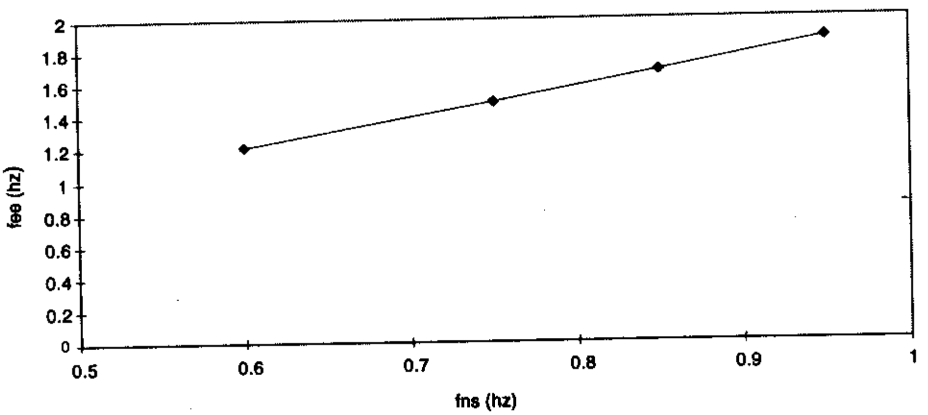

Fig. 8. Self excited frequency (fee) vs natural frequency of the primary system (fns) at suppression condition.

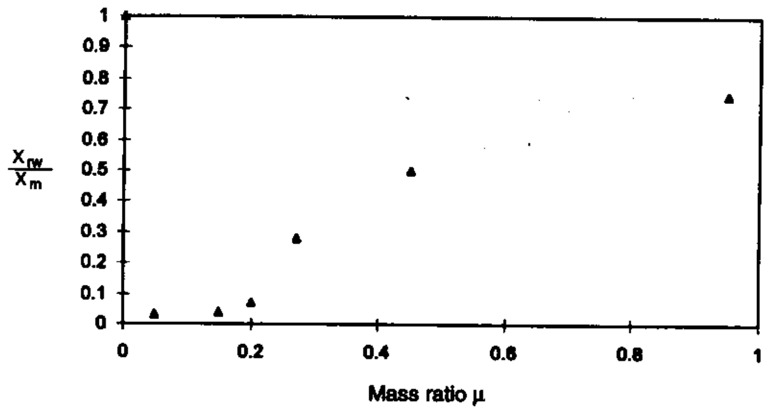

Fig. 9. The minimum amplitude ratio versus the mass ratio at the optimum frequency ratio (fer/fns = 3). [note: The optimum amplitude ratio Xr_w/Xr_n =.035 at mass ratio μ =.05]

At the optimum mass ratio, the optimum amplitude ratio, X_{rw}/X_m, is approximately 0.035 on the average, which can be used to predict the suppressed amplitude of the main system X_{rw} given its resonant vibration magnitude X_m.

In summary, the hinged plates can be used to suppress the vibrations of a given main system (up to approximately 97% reduction) that is in state of flow induced natural oscillation at a specific wind speed, as long as the plates are designed to execute self-excited oscillations at that wind speed with uncoupled frequency (f_{er}) equal to three times the natural frequency of the main system, when the total mass of the plates is about 5% of the effective modal mass of the system.

Optimum Design of the Absorber

A computer program was developed based on the previously outlined experimental results to determine the absorber control parameters (L_1, L_2, H, ρ_{plate}) at each approaching flow speed such that the suppression of the main system vibrations was maintained throughout the entire flow speed range. The optimization problem can be stated as follows.

Given: U_{max},U_{min}, f_{ns} , M_{es} and ρ_f
Find: L_{1max} , L_{2max}, H, L_{1opt} and L_{2opt} at each wind speed in the given range

To minimize the Maximum Dynamic Response Ratio $\{ \frac{X_{rw}}{X_m} \}$ or, Minimize: The Merit Function $= \left\{ 1 - \frac{f_{er}/f_{ns}}{TC} \right\}$ (1)

Subjected to the constraint : mass ratio $\mu = M_a / M_{es} = .05$.

TC is the frequency tuning coefficient, L_{1max} and L_{2max} are the maximum designed lengths of the primary and secondary plates, respectively, $\{U_{min} - U_{max}\}$ is the fluid speed range expected to cause flow induced vibrations in the main system. The frequency ratio f_{er}/f_{ns} should be kept at three for optimal conditions, as shown in the preceding section.

The self-excited frequency of the hinged plates on a rigid support f_{er} is empirically developed and can be written as [8]:

$$f_{er} = \frac{1}{2\pi} \frac{.75 + \left(-2 + 4\sqrt{2}Rn - \sqrt{3}Rn^2\right)}{(1+Rn)^3} \sqrt{\frac{\rho_f (U - U_o)^2 H L_t^2}{I_t}}$$

For R < Rm

$$f_{er} = \frac{1}{2\pi} \frac{.707 + \left(-1 + .4\sqrt{2}Rn - 2Rn^2\right)}{(1+Rn)^3} \sqrt{\frac{\rho_f (U - U_o)^2 H L_t^2}{I_t}}$$

For R > Rm

$$Rn = R / Rm \qquad Rm = .1 + \left(2\sqrt{3}\rho_{rs}\right)^{-1}$$

$$U_o = \sqrt{\frac{M_a g}{\rho_f L_t H}} \left[2e^{-Rn} - e^{-.25Rn}\right] / 4 \qquad Rn = R / R_\Phi$$

$$R_\Phi = 0.707 \sqrt{\frac{1}{\rho_{rs}}} \qquad \rho_{rs} = \frac{\rho_{plate}}{1000\rho_f} \qquad (2)$$

A code OPTIMABS was developed that solves for the parameters (L_1, L_2, ρ_{plate}) which minimize the merit function and satisfies the given constraint at each fluid speed for a given primary system. The code implements the Univariate search technique procedure and the details of the design procedure is given in [9].

The inputs to the code are the fluid speed range, the density, the natural frequency of the main system f_{ns} and its effective mass M_{es}. The outputs of the program are the maximum total length of the plates L_{tmax} and the lengths L_{1opt}, L_{2opt} at each fluid speed (U).

As an example, the code is used to design an absorber for a given main system with the following parameters: $f_{ns} = 0.6$ Hz, $M_{es}= 2.45$ kg, damping ratio $\varsigma = 0.0043$ and flow speed range of [6.25, 12.5 m/s].

The results for the optimum absorber parameters obtained by the program are shown in Table 2. To check the performance of the calculated optimum absorber with lengths ($L_{1opt.}$ and L_{2opt}), a set of experiments were conducted to verify the optimum results obtained by the program at each wind speed. The results are presented in Fig. 10. The figure shows that, using the optimum parameters obtained from the code at each wind speed, the vibration amplitude of the main system can be kept at less than four percent of its resonant amplitude without the absorber. This occurs in the entire range of the flow speeds.

Table 2. The optimum values of L_1 and L_2 at each approaching wind speed, fn_s =.6 hz

U (m/s)	L_{1opt} cm (in)	L_{2opt} cm (in)
6.25	10.16 (4)	10.16 (4)
7.45	10.16 (4)	10.16 (4)
8.1	10.16 (4)	10.16 (4)
8.96	10.16 (4)	10.16 (4)
9.74	10.16 (4)	22.86 (9)
10.34	10.16 (4)	22.86 (9)
11.58	20.32 (8)	7.62 (3)
12.5	20.32 (8)	7.62 (3)

A sample of time domain signal showing the amplitude of vibration of the main system without the absorber for the first 7 seconds then with the absorber is depicted in Fig.11.

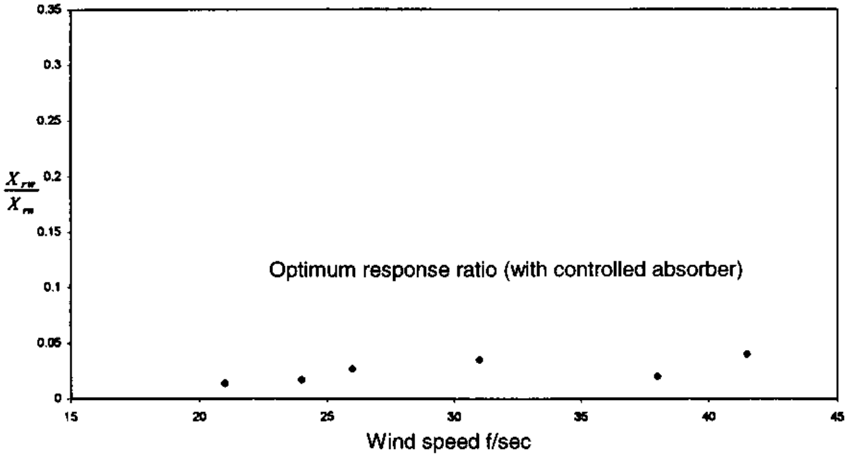

Fig. 10. Experimental verification of the calculated optimum absorber for the main system (I) with fns =.6 hz.

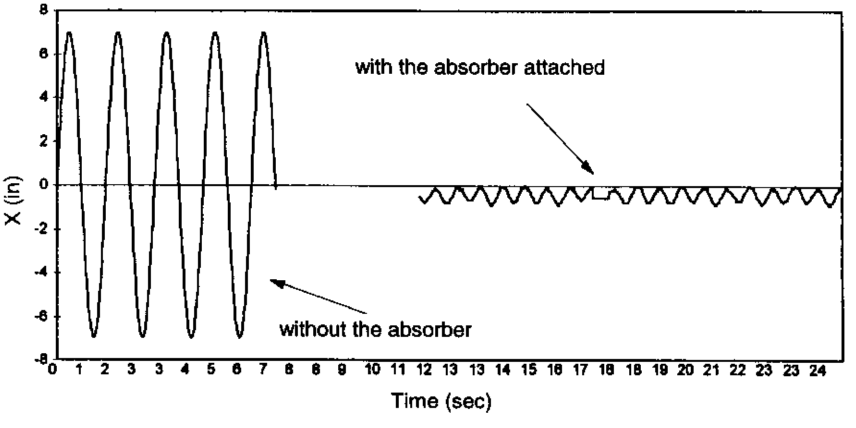

Fig. 11. Time history of the main system (fns =.6 hz) with and without the optimum absorber a speed U = 31.6 f/s.

Implementation of Variable Length Absorber

Form the last section, it is found that for each flow speed there is an optimum absorber that can suppress the flow-induced vibration of the main system. In practice, the active absorber should change its lengths automatically according to the magnitude of the incoming flow speed. Therefore for a given absorber mass, the density has to change as the plate lengths (L_1, L_2) change. Figure 12 illustrates the configuration of the absorber with variable length. Note that the gap between the plates is maximum at full extension and minimum,approximately solid, at the minimum length. The density of the absorber can be readily calculated for any length condition.

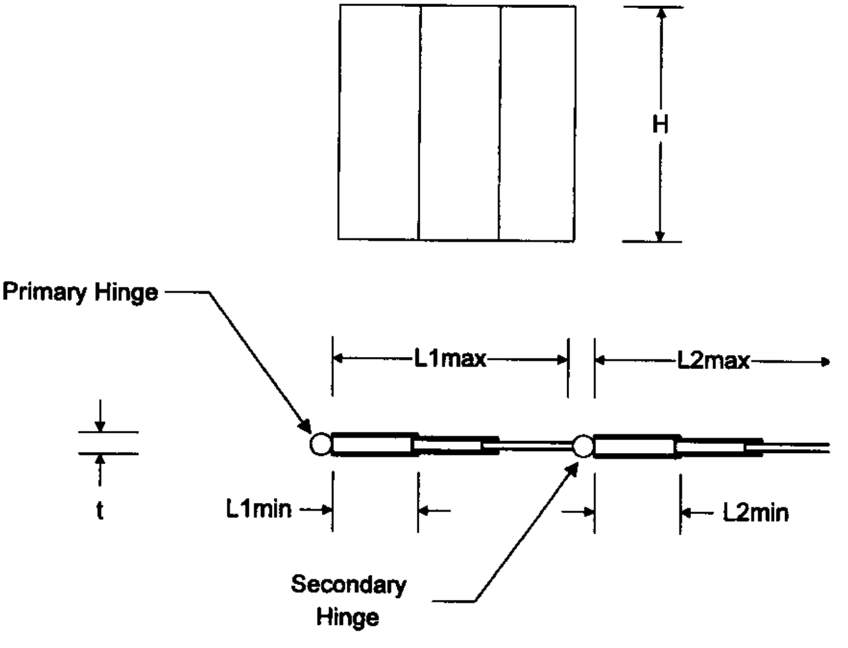

Fig. 12. The variable length absorber.

The relationship between the plate lengths and the approaching flow speed represents the control law that adjusts the absorber dimensions to maintain the amplitude of vibrations X of the main system minimum at all times.

Since the dynamics of the controlled system are known, a feed-forward algorithm, that contains the optimum plates geometry as function of flow speed, can be incorporated in the control system.

Some Applications of the Proposed Absorber

Illustrative examples

Any slender structure such as a tower, chimney, mast or high rise light structure is considered a good candidate for this application. For instance, the antenna base of telecommunication towers should not oscillate beyond an allowable limit. These towers usually are prone to wind loading more than any other source of excitation.

Figure 13 shows a typical chimney structure made of steel subjected to uniform wind loading. Part of the data in this example is adapted from [10]. The damping ratio is estimated to be approximately (0.005), the total mass is about 1.01×10^5 kg (2.22×10^5 lb), the fundamental mode is close to (f_{ns}= 0.792 hz), the diameter (D) of the cross-section is about 3 meters (9.75 ft) and height of 92.5 meters (300 feet). The critical wind speed is estimated from the Strouhal number as follows:

$$U_c = \frac{f_{ns}\, D}{S} = \frac{.792\,(3)}{.22} = 10.8\, m/s$$

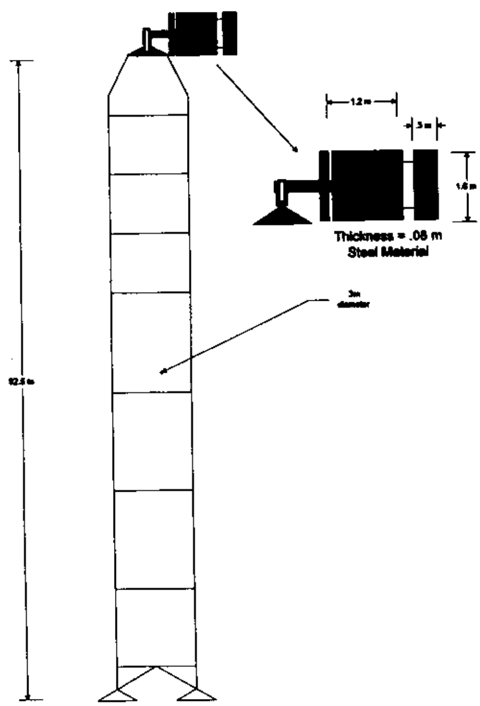

Fig. 13. Typical chimney structure and the optimum absorber.

To design an absorber for this main system at this specific critical wind speed, the data are entered as an input to the code to solve for $L_{1opt.}$ and L_{2opt} of the absorber. The optimum results are shown in Fig. 13. This absorber is expected to suppress the flow induced vibration of the chimney such that the vibration amplitude ratio X_{rw}/X_m can be maintained at approximately 0.035. From the results in Fig. 13, It can be seen that the dimensions of the absorber and its total mass are compatible with the size of the main system. Care must be exercised when attaching the absorber at the top of the chimney to avoid any interference with its plume. Furthermore, circular steel plate can be fixed beneath the oscillating absorber to protect the pedestrians below in case of absorber failure.

Other detailed examples including the design of optimum absorbers for the suppression of galloping vibrations of transmission lines and the vortex induced vibration of underwater risers are given in reference [9].

Conclusions

Based on the experimental investigation carried out in this study, it is found that a uniform flow-induced vibration of a structure can be suppressed significantly by attaching two hinged plates at or near the location where the largest displacement is expected. The hinged plates should have the following characteristics:

- the flow induced frequency of the plates f_{er} should be three times the natural frequency of the structure;
- the total mass of the plates should be equal to 5% of the modal mass corresponding to the first mode of vibration of the structure.

Since the total force required to perform the suppression process is composed of the induced aerodynamic force from the absorber as well as the inertia effect of the absorber one would expect to achieve the desired effect by a smaller mass for the absorber as opposed to the conventional absorber (the absorber does not interact with the flow) where the inertia of the absorber plays the major role in the suppression process.

The proposed absorber is relatively easy to construct and can align itself in the direction of the flow continually. Accordingly, one absorber would be capable of suppressing the cross-flow vibrations in any direction.

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كبح الاهتزازات الميكانيكية الناتجة عن سريان مائع بواسطة ذبذبة ذاتية لكابح يتكون من لوحين بمفاصل

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(أستلم في ٩٨/٨/١٩ ؛ وقبل للنشر في ٩٩/٤/٥)

ملخص البحث. في هذا البحث، دراسة لإمكانية استخدام نفس طاقة المائع المسببة للاهتزازات لكبحها وذلك بإضافة كابح للمنشأة يستمد حركته من سريان المائع عليه. يتكون هذا الكابح من لوحين بينهما مفصل. استخدمت كمرة ذات سماكة منتظمة وكتلة متغيرة في أحد طرفيها، لتمثل المنشأة الميكانيكية. عند هذا الطرف، ثبت لوحين بمفصل للقيام بعملية الكبح للاهتزازات عن طريق سريان الهواء عليه.

أعد برنامج يقوم بتصميم أبعاد الكابح بحيث يعمل بشكل أمثل. كما تم القيام بتجارب للتحقق من نتائج التصميم التي تم الحصول عليها من البرنامج وأعدت مقارنة بين النتائج التجريبية والمتوقعة لمقدار الاهتزازات الميكانيكية الناتجة عن سريان الهواء على المنشأة.