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# INVESTIGATIONS ON DAMPING CHARACTERISTICS OF LIQUID COLUMN VIBRATION ABSORBER

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## ABSTRACT

This work deals with the damping characteristics and optimum performance of a passive liquid column vibration absorber (LCVA) which is effective to decrease the vibration of engineering systems. A numerical technique is applied to solve the nonlinear equations of motion to study the influence of the design parameters such as frequency tuning ratio, length ratio, area ratio, blocking ratio and mass to mitigate excessive vibrations. A constrained multi-objective optimization problem is constructed and solved numerically to minimize the maximum master structure displacement in a broad range of excitation frequency with considering a limit of the maximum liquid displacement in the vertical tube as an extra objective. It is found that the best vibration attenuation can be achieved by using the LCVA with a uniform cross-section area while the non-uniform LCVA offer greater flexibility to modify the non-appropriate total length. The TCVA mass and the structural damping have an influence on all optimum parameters. The optimal values of the blocking ratio are proportional and depend on the intensity of the excitation. The list of necessary optimal parameters and the corresponding performance indices are presented in design tables as a guidance for industrial practices.

## KEYWORDS

Tuned liquid damper, TLCD, LCVA, vibration control, optimization.

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## INTRODUCTION

One of the challenges engineers face today is to find effective and better means of protecting engineering structures such as bridges, wind turbines, floating platforms, etc. from the damaging impacts of undesirable vibrations induced by the dynamic loadings such as earthquakes, ocean waves and wind forces. [1, 3]. One of the pervasive strategies widely utilized to attenuate system vibration is the installation of a dynamic vibration absorber (DVA), which is a device generating a reaction force created from the oscillation of a secondary mass. Several types of passive DVAs have been proposed, one of them is the liquid column vibration absorber (LCVA) which has an effective performance to attenuate the vibration of the dynamic systems [4-6].

The LCVA absorbs the energy from the master structure through the liquid mass in a rigid U-shaped container. This energy is lost through the interacted between the moving liquid and the rigid tube, and head losses created from the liquid motion between the horizontal and vertical portions. The LCVA may also include orifices for extra energy dissipation. LCVAs have unique advantages over other damping devices including low cost, easy installation, economic maintenance and no need to add mass to the structure and using the liquid as a water supply etc. A special class of LCVA, is termed tuned liquid column dampers (TLCD) with uniform sectional area in the horizontal tube and the vertical column. The liquid damping of a LCVA is hard to quantify because of its nonlinearity. Even for a linear master structure, the behavior of the TLVA-master structure is non-linear. In order to overcome this problem, there has been a proliferation of studies to use an equivalent linearization technique to simplify the solution [7-10].

Min et al. [11] derived an analytical formula for the TLCD with an undamped system under Gaussian random excitations for design purposes. Moreover, a detailed studies on the influence of various orifices on damping characteristics of the TLCD at a broad extent of wind intensities were presented to investigate the optimal design of the TLCD. By using the optimal properties suggested in Ref. [1]. Park et al. [12] introduced an optimal design process of a LCVA to mitigate the vibration of a SDOF system under wind excitation based on the statistical linearization technique (SLT), Matteo et al [13-15] proposed and developed new straightforward approximate formulae to determine the optimum parameters of a TLCD attached to a master structure under random loads. Shum et al. [16] explicitly derived the optimal parameters of a LCVA with small structural damping subjected to white-noise induced ground acceleration. Also they noticed that the derived parameters are also usable for the TLCD. Moreover, He [17] derived simple approximate expression for the standard deviation velocity response of tuned vibration absorber to facilitate the estimation of the head loss coefficient of LCVA. Sonmez et al. [18] proposed a new model for semi-active TLCD, which is connected to the master structure using an adaptive spring. Fourier transformation is applied to adjust the spring constant.

Although LCVAs have been studied extensively in literature, most researches are mainly concerned with the determination of their optimal design parameters through (SLT), Due to the inherent nonlinearity of the liquid damping, the optimum parameter for damping, which was derived based on a linearized equation, could not be directly applied to the case of liquid dampers. The LCVAs with their nonlinear behavior are still needed to be developed for practical designs. Furthermore, only few studies have particularly dealt with the influences of the master structure damping on the optimal LCVA parameters. The numerical methods are only way should be performed to compute these parameters when a structural damping is taken into account [19]. Furthermore, the SLT, do not put any constraints on the damping force even during strong excitations, where the maximum oscillation of the liquid may be greater than the vertical pipe. Therefore, it is important consider the restrictions on the peak amplitude of the liquid motion during the optimization approach.

The objective of this study is to reduce the response of the master structure by means of installing a LCVA, and to estimate the influence of different parameters such as mass ratio, area ratio and length ratio on its performance. The nonlinear EOMs for the LCVA-master structure are derived based on the Lagrange's equation and were numerically solved. The optimal design parameters of a LCVA are numerically acquired through the minimization of the maximum response of a master structure equipped with a LCVA. Simultaneously, the constraint criterion is imposed by limiting the peak displacement of the liquid cannot greater than the vertical column height.. Useful design tables are introduced as quick guidance for practical applications. For different values of mass ratio, master structure damping factor and force intensity factor, these design tables provide a list of necessary LCVA design parameters, including frequency tuning ratio, blocking ratio for several values of length and area ratio in addition to the corresponding performance indices.

## MATHEMATICAL MODEL

Figure 1 displays a schematic diagram of a LCVA attached to a single degree of freedom (SDOF) master structure under an external excitation. The mass, the damper constant and the stiffness of the master structure are m, c and k respectively. The master structure displacement is x and y is the vertical displacement of liquid surface. The liquid volume inside the LCVA can be split into two vertical parts and a horizontal part. For incompressible flow and neglecting the sloshing behavior. The kinetic energy KE and the potential energy PE of the LCVA –master structure can be expressed as;

$$\mathrm{KE} = \frac{1}{2}M\dot{x}^2 + \frac{1}{2}\rho A_v(l_v - y)(\dot{y}^2 + \dot{x}^2) + \frac{1}{2}\rho A_v(l_v + y)(\dot{y}^2 + \dot{x}^2) + \frac{1}{2}\rho A_h l_h(\gamma\dot{y} + \dot{x}) \quad (1)$$

and

$$\mathrm{PE} = \frac{1}{2}kx^2 + \frac{1}{2}\rho g A_v\,(l_v - y)^2 + \frac{1}{2}\rho g A_v(l_v + y)^2 \quad (2)$$

where, $A_v$ and $A_h$ are the area of vertical and horizontal columns respectively and $l_h$ and $l_v$ are the horizontal and vertical lengths of the liquid columns, respectively. $\rho$ and g are the liquid density and the gravity acceleration, respectively. The non-conservative force $Q_x$ in x direction and the damping force of liquid motion $Q_y$ in y direction can be defined as:

$$Q_x = -c\dot{x} + f(t) \quad (3)$$

$$Q_y = -\frac{1}{2}\rho A_h\gamma\delta\dot{y}|\gamma\dot{y}| \quad (4)$$

where, δ is the coefficient of overall head loss and $\gamma = \frac{A_v}{A_h}$ is the area ratio.

Using Lagrange's equations, the equations of motion of the LCVA-master structure can be written as:

$$\big(M + \rho A_h(2\gamma l_v + l_h)\big)\ddot{x} + \rho A_h \gamma l_h \ddot{y} + c\dot{x} + kx = f(t) \tag{5}$$

$$\rho A_h \gamma(2l_v + \gamma l_h)\ddot{y} + \rho A_h \gamma l_h \ddot{x} + \frac{1}{2}\rho A_h \gamma^2 \delta \dot{y}|\dot{y}| + 2\rho A_h \gamma g y = 0 \tag{6}$$

Therefore, the natural frequency of the LCVA is $\omega_a = \sqrt{2g/l_e}$ . Where, $l_e$ is the effective length which can be written as:

$$l_e = 2l_v + \gamma l_h \quad \text{and} \quad l = l_e/(1 + \alpha(\gamma - 1)) \tag{7}$$

Introducing the following dimensionless variables:

$$\mu = \frac{m_l}{m}, \alpha = \frac{l_h}{l}, \varphi = \frac{l_h}{l_e}, \beta = \gamma\alpha/(\gamma - \alpha(\gamma - 1)), \ \zeta_s = \frac{c}{2M\omega_n} \tag{8}$$

Defining $f_c = f_o/mg$ as the force intensity factor, where, $f_o$ and $\omega$ are the amplitude and the frequency of the harmonic external excitation, respectively. Therefore, Eqns. (5, 6) can be written into a matrix form as:

$$\begin{bmatrix} (1+\mu) & \mu\beta \\ \varphi & 1 \end{bmatrix}\begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} + \begin{bmatrix} 2\zeta_s\omega_n & 0 \\ 0 & \frac{\gamma\delta|\dot{y}|}{2l_e} \end{bmatrix}\begin{bmatrix} \dot{x} \\ \dot{y} \end{bmatrix} + \begin{bmatrix} \omega_n^2 & 0 \\ 0 & (r_a\omega_n)^2 \end{bmatrix}\begin{bmatrix} x \\ y \end{bmatrix} = \begin{bmatrix} f_c g \sin(\omega t) \\ 0 \end{bmatrix} \tag{9}$$

where; $\zeta_s$ and $\omega_n = \sqrt{k/m}$ are the damping ratio and the natural frequency of the master structure respectively, $\alpha$ is the length ratio, $\mu$ is the mass ratio, $m_l = \rho A_h(2\gamma l_v + l_h)$ is the mass of the liquid and $r_a = \omega_a/\omega_n$ is the frequency tuning ratio

Solving Eqn. (9) by the Adaptive Runge–Kutta method which considers nonlinearities, the response of the master structure attached with LCVA in both the time and the frequency domains can be determined.

## ANALYSIS OF RESULTS AND DISCUSSION

In this section, a parametric analysis is performed in order to show the influence of varying the different parameters mainly the master structure damping factor, the mass ratio, the length ratio, the blocking ratio, the area ratio and the excitation intensity on the vibration attenuation of the master structure. The relative displacement $X_r = \frac{X_{max}}{X_w}$ is determined as the ratio of the peak response of the master structure with a LCVA ($X_{max}$) to its peak response without a LCVA ($X_w$) in forced vibration modes. Furthermore, the relative damping factor $\zeta_r = \frac{\zeta_e}{\zeta_s}$ is defined as the ratio of the damping factor of the master structure equipped with a LCVA ($\zeta_e$) to the damping factor of the master structure without a LCVA ($\zeta_s$) in free vibration modes.

### Effect of Mass Ratio

Figures 2 and 3 show the variation of the relative damping factor $\zeta_r$ with the mass ratio $\mu$ for several values of the damping factor $\zeta_s$ and natural frequency $f_n$ respectively. It can be observed that, the TLCD with a bigger mass can achieve higher relative damping factor (better performance) due to the increase in $m_{12}$ ($m_{12} = \mu\beta$) in Eqn.(9). Furthermore, the TLCD can achieve better performance with a lower damping $\zeta_s$ and higher $f_n$. Therefore, the dynamic effectiveness of the TLCD depends on the master structure characteristics beside the mass ratio.

**Effect of Frequency Tuning Ratio**

It can be noticed from Fig. 4 that, the value of frequency tuning ratio $r_a$ to maximize the damping of the structure- TLCD system in free vibration mode is close to 1, and it became closer to 1 as the mass ratio decreases. The results of forced excitation are shown in Fig. 5. It can be seen that $r_a$ =1 provides the most favorable vibration mitigation at the resonance frequency. For $r_a$ smaller than one gives an acceptable vibration attenuation with the merit of low total length of the TLCD. Decreasing $r_a$ , the total length of the TLCD become smaller due to the decrease of its effective length as expressed in Eqn. (7). This may be useful in the case of very long TLCD.

**Effect of Length Ratio**

Referring to Fig .6, rising the length ratio $\alpha$ can reduce the maximum master structure response more efficiently. This is due to the mass of the horizontal part of TLCD is the only effective mass of TLCD acting on the master structure as shown from the second term in Eqn. (5). The TLCD with a higher $\alpha$ value has a more effective horizontal liquid mass. This trend is not related to the external force intensity as shown in Fig.7. Referring to Fig.7, for the force intensity factor $F_c$ = 0.05, When $\alpha$ increases from 0.3 to 0.7 the maximum system response is greatly decreasing by about 78 %. While when $\alpha$ increases from 0.7 to 0.9, there is a slightly decrease in the maximum system response by about 4%. However, α is restricted by the liquid displacement in the vertical pipe. So larger length ratio should be avoided in the design of a TLCD, especially for strong external excitation where the large liquid displacement in the vertical tube is expected and a little improvement in the TLCD effectiveness.

**Effect of Cross Section Area Ratio**

Figure 8 shows the influence of the area ratio γ on the relative steady state response $X_r$ for various values of length ratio α. The mass ratio $\mu$ and the damping factor $\zeta_s$ are fixed at the values 0.03 and 0.01, respectively. For small values of $\alpha$ ($\alpha \leq 0.5$), it is found that the best performance occurs for equal cross section area( $\gamma = 1$). In case of a high value of α (α =0.9), we get a small improvement in the system damping by increasing γ. This trend is not related to the mass ratio as shown in Fig.9 for free vibration modes. Therefore, for low natural frequencies of the master structure, the length of liquid column should be relatively long. In this case, it is appropriate to take $\gamma > 1$ to save space because the total length will be shorter than that for $\gamma = 1$ as shows in Eqn. (7). Thus, the effective length is determined only by the natural frequency of the LCVA.

**Effect of Blocking Ratio**

Figure 10 shows the effect of the blocking ratio $\psi$ on the relative steady state response $X_r$ in frequency domain. Blocking ratios $\psi$ from 0 to 80% are used for excitation frequency ratio ranging from 0.8 to 1.2. The smaller blocking ratio TCVA leads to a two-hump curve, and the higher one corresponds to a one-hump curve whose peak response value is larger. The drop of the peaks near resonance frequencies are noticeable, especially for blocking ratios less than 20%. For larger values the blocking ratio, $X_r$ return to increase due to the decrease in the liquid motion.

Figure 11 and 12 show the influence of the blocking ratio on the performance index ($\eta = (1 - X_r) * 100$) and relative liquid displacement $\emptyset = \frac{y_{max}}{l_v}$ , respectively for force intensity $F_c$ = 0.2% and 0.3% and relative length $\alpha$ = 0.5, 0.7 and 0.9. It is seen that for the whole range of $\psi$ the TLCD, using the small blocking ratio value, provide the largest performance index value (good performance). On the other side, as the blocking ratios increase, the vertical oscillations of the liquid column of the TLCDs are suppressed. Furthermore, raising the values of $\alpha$ or $F_c$ increase the relative liquid displacement and therefore, their higher values require a bigger blocking ratio to keep acceptable relative liquid displacement. Fig.13 shows the variation of the performance index and relative liquid displacement $\emptyset$ with the blocking ratio for $\gamma = 0.5, 1, 1.5 \text{ and } 2$. It is observed that increasing the area ratio leads to a decrease in both the performance index $\eta$ and the relative liquid displacement. The larger performance index is obtained when γ=1 with a medium relative liquid displacement.

## PARAMETRIC OPTIMIZATION

To optimize the LCVA parameters for achieving the best vibration attenuation of the master structure a constrained optimization problem is formulated and solved numerically. For the present problem, the mass ratio, length ratio, area ratio and expected force intensity can be specified by the designer. The frequency tuning ratio and the blocking ratio are variables considered to determine their optimal values.

The objective of the current optimization process is to minimize the maximum relative steady- state displacement $\left.X_r\right)_{max}$ within the excitation frequency range. The constraint is imposed by limiting the maximum relative displacement of the liquid to unity ($\phi_{max} \leq 1$) .

The optimization problem is defined as:
Find $r_{ao}$ and $\psi_o$
To minimize $\left.X_r\right)_{max}$
With
$\phi_{max} \leq 1$ , $lb_1 \leq r_a \leq ub_1$ , $lb_2 \leq \psi \leq ub_2$

where, $r_{ao}$ and $\psi_o$ represent the required optimal values for frequency tuning ratio and blocking ratio respectively. $lb_i$ and $ub_i$ are the lower and upper bound for each parameter (i = 1,2).

Since the above defined multi-objective optimization problem and the variables are of a continuous type thus, Eqn. (10) is a nonlinear complex problem. To find its optimal solution the Global Search algorithm (GSA) is used which an iterative numerical optimization technique. GSA uses the scatter search algorithm to create a set of trial points as start points within a finite bounds and has an efficient nonlinearity constrained multivariable optimization solver. It analyzes the start points and takes only those points with a good probability of achieving a global minimum and runs the solver from that point. The solver uses sequential quadratic programming (SQP) method to optimize the object function. The formularization of SQP is based on Newton's method and Karush-Kuhn-Tucker (KKT) equations. KKT equations based algorithms try to get the Lagrange multipliers directly. Further details of the GSA implementation can be found in Ref. [20].

$$L(x,\lambda) = f(x) + \sum_{i=1}^{m} \lambda_i \, . \, g_i(x)$$

At the kth iteration, the following QP sub program is solved:

$$\min_d \quad \frac{1}{2} d^T H_k \, d + \nabla f(x_k)^T d$$

For new iterate the solution can be expressed as:

$$x_{k+1} = x_k + \beta_k d$$

where d is the direct vector at the given point $x_k$ and $\beta_k$ is the step size

The design tables shown in Tables 1 to 3 are constructed for $\gamma$ = 0.5, 1 and 2. These tables are suitable for structure damping factor and mass ratio ranging from 1% to 3% and 2 to 5%, respectively. Tables 1 to3 give the values of the optimal tuning frequency ratio $r_{ao}$ , optimum blocking ratio $\psi_o$ and the corresponding performance index of the master structure at resonance with an optimized TCVA as well as the maximum relative response liquid $\varphi$ for length ratio $\alpha = 0.5\,, 0.7 \text{ and } 0.9$ and force intinisity factor $F_c$= 0.005 and 0.015.

## CONCLUSION

The effectiveness of a LCVA in controlling peak structural response is studied. The results reveal that a LCVA can efficiently reduce the structural peak responses to imposed harmonic excitations. A uniform LCVA is always the best choice. Increasing the area ratio can greatly reduce the liquid column length requirement to suppress the same level of master structure vibrations. The LCVA with higher mass ratio can be more efficient in suppressing the master structure response and practically it is suitable to use mass ratio $\leq 5\%$. The better performance occurs as the length ratio becomes larger, but the liquid surface displacement tends also become larger. The blocking ratio is effectively controlling the extensive liquid displacement especially for large length ratio and high level excitation. The numerical optimization results show that, the optimum frequency tuning ratio for the system with higher mass ratio and higher master structure damping ratio should be lower. The value between 0.9 to 1.04 was appropriate for moderate to high level external excitations. The optimum Blocking ratio depends proportionally to the intensity of the excitation and it is inversely proportional to the mass ratio.

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**FIGURES**

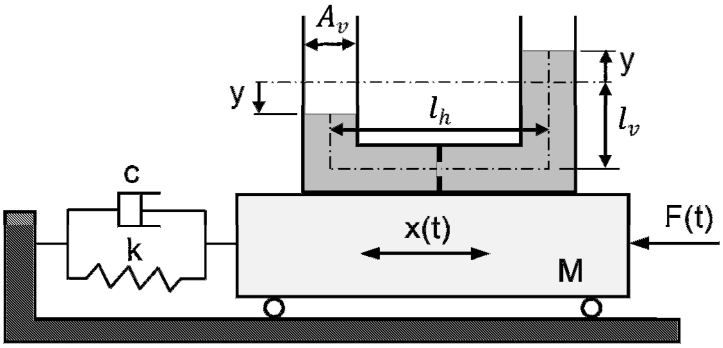


**Fig.1.** Schematic diagram of the master structure equipped with LCVA.

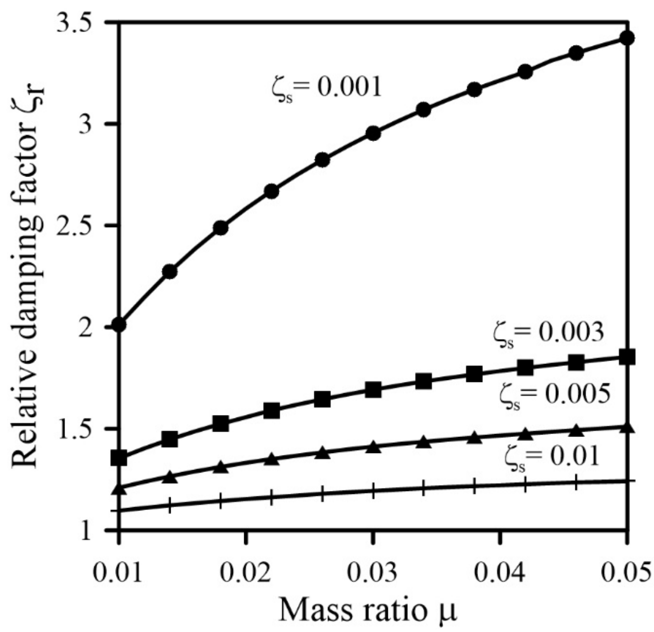


**Fig. 2.** Influence of the mass ratio on the damping factor for different master structure damping factors.

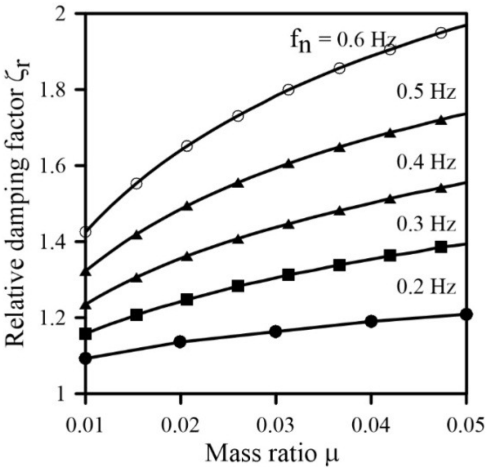


**Fig. 3.** Influence of the mass ratio on the relative damping factor for different natural frequencies of the master structure.

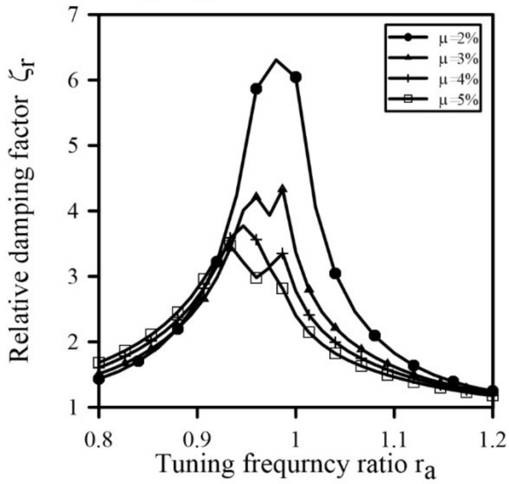


**Fig. 4.** Influence of tuning frequency ratio on the relative damping factor for different mass ratios.

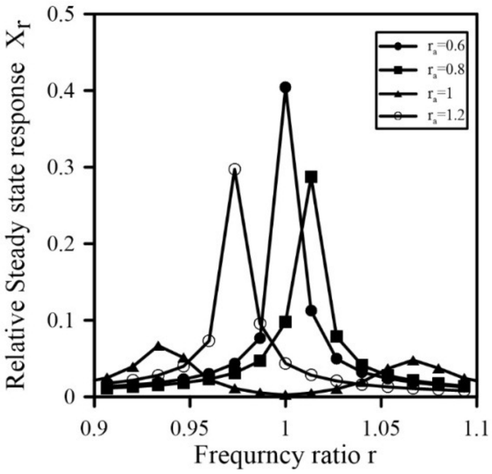


**Fig. 5.** Frequency ratio effect on the relative system response for different tuning frequency ratios.

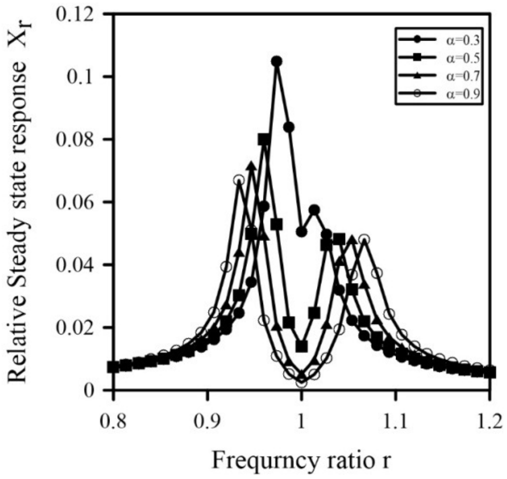


**Fig. 6.** Frequency ratio effect on the relative system response for different length ratios.

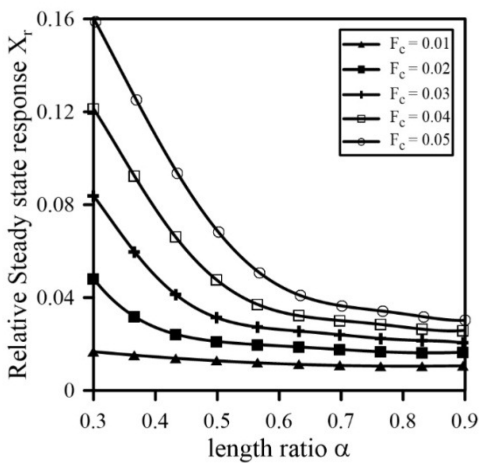


**Fig. 7.** Variation of the relative system response with relative length for different excitation intensities.

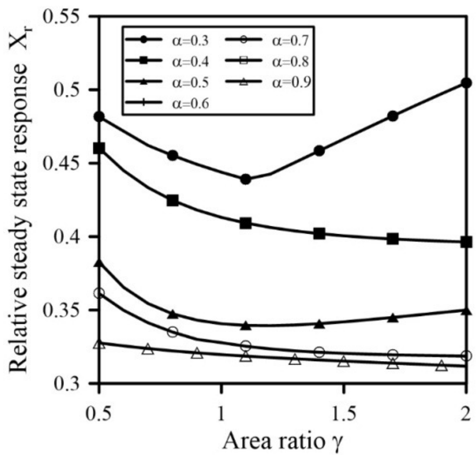


**Fig. 8.** Area ratio effect on the relative system response for different length ratios.

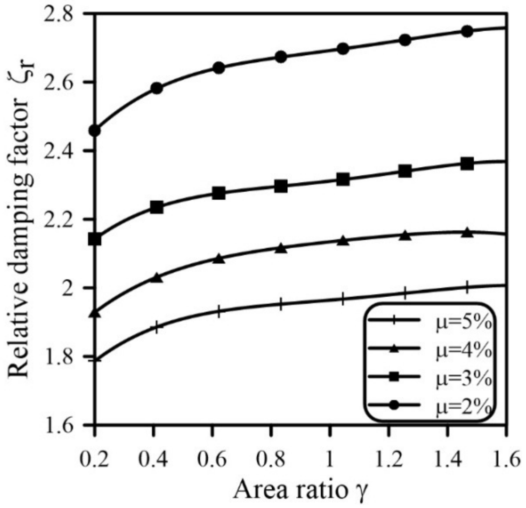


**Fig. 9.** Area ratio effect on the relative damping factor for different mass ratios.

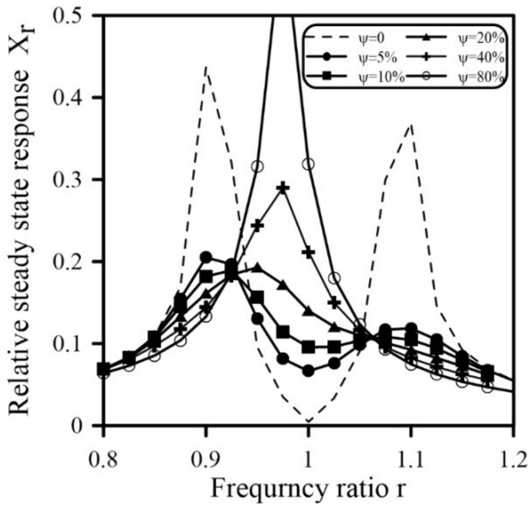


**Fig. 10.** Effect of blocking ratio on the system response for different relative lengths.

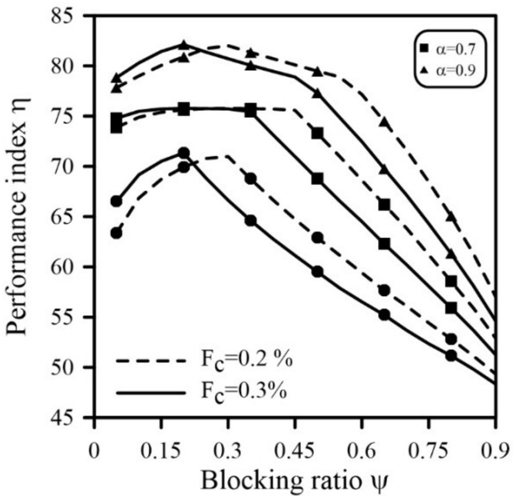


**Fig. 11.** Effect of blocking ratio on the system response for different relative lengths.

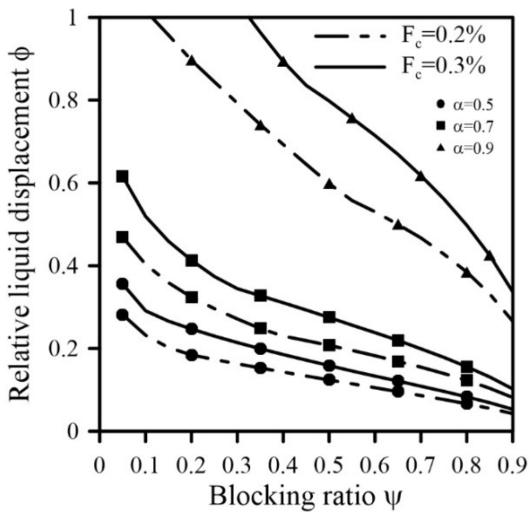


**Fig. 12.** Effect of blocking ratio on the performance index for different relative lengths.

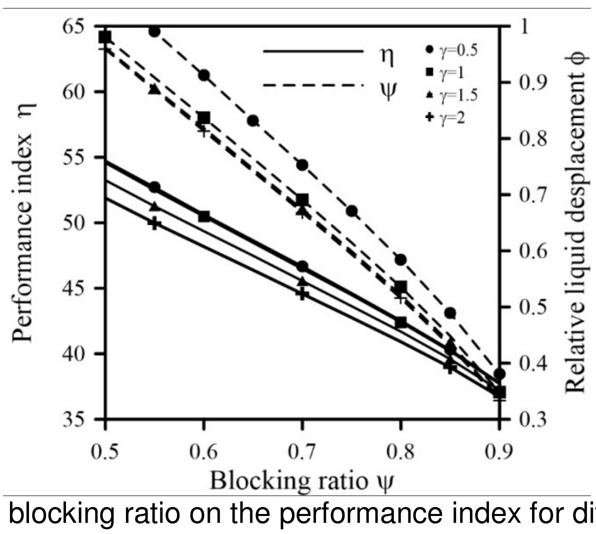


**Fig. 13.** Effect of blocking ratio on the performance index for different area ratios.