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Active Vibration Suppression of Laminated Composite Structures Integrated with Magnetorheological Fluid Segments

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***Abstract:** This study exhibits synthesis of full-state LQR controllers for suppression of free and forced vibration of a cantilever plate fully and partially treated with the magneto-rheological (MR) fluid. The governing equations of motion of the three layer MR sandwich composite plate are showed in the state variable form containing a function of the control magnetic field. An optimal control elaboration based on the linear quadratic regulator (LQR) developed to conquer the vibration of the plate under limited magnetic field intensity. The free-and forced vibration control performances of LQR control strategies are assessed for the fully as well as partially treated MR-fluid sandwich composite plates. The results demonstrate that the full-state observer-based LQR control can considerably reduce the tip deflection responses and the settling time of the free vibration oscillations. The partially-treated plate with MR-fluid concentration near the free end also yields vibration responses comparable to the fully treated plate, while the natural frequencies of the partially treated beams are considerably higher.*

***Keywords:** composite layered plate, active vibration control, MR fluid*

I. INTRODUCTION

Because of spectacular bending rigidity, low specific weight, high ranking isolating qualities, excellent vibration characteristics and amazing fatigue properties, sandwich construction has become even more interesting due to the introduction of advanced composite materials for the faces and cores. A composite is a structural material that consists of two or more combined constituents that are combined at a macroscopic level and are not soluble in each other. Dissimilar from beams or columns, plate can carry an additional load after buckling without failure in so many structures. [1], [2]. Comparatively few works have been committed to the study of free vibration of composite plates. The modal equations of motion were gained from Lagrange's equation with the strain energy and kinetic energy [3], [4]. Control of vibration in structures through active, semi-active and passive vibration isolation systems extend to be the concentration of many researches. A wide range of active vibration control structures have indicated substantial performance gains, as their executions have been majorly limited because of the high cost and power requirements [5]. In other way, the passive systems are acknowledged to be most dependable, while defined damping parameters take a trade-off between the control of vibration at resonance and the higher frequency isolation performance [6]. Instead, semi-active vibration control structures have indicate to offer the fail-safe and reliable characteristic of the passive systems together with the performance gains in comparison with those of the active devices with lowest power demand[7]. In detail, semi-active devices with smart fluids with controllable rheological properties like Electrorheological (ER) and Magnetorheological (MR) fluids provide remarkable potential for gaining control of vibration over a broad frequency range with only lowest external power [8]. Such fluids are able to offer substantial and quick changes in the damping and stiffness properties of structures with application of an electric or a magnetic field [9]. Even though both the MR and ER fluids normally show similar viscosity in their non-activated or "off" state, the MR fluids exhibit a much greater raise in viscosity when activated and need relatively low power compared to the ER fluids [10]. The yield stress of the MR fluid, MRF 100, is in the range of 2-3 kPa in the without a magnetic field, but it quickly increases 80 kPa under the influence of a magnetic field in the order of 3000 Oe [11]. The mentioned structures have mostly applied lumped ER/MR dampers at selected distinct places of the structures. Such models therefore take various damping elements to control the vibration according to different modes, which would require complicated controller designs. Instead, a few studies have applied ER/MR fluids in simple structure models to accomplish controllable spread properties by embedding ER/MR material layers between two elastic/metal layers. This method can yield important variations in distributed stiffness and damping properties of the structure, and therefore provides superior potential for control of multiple vibration modes. Moreover, the embedded MR/ER fluid treatments could ease up more compact designs compared to the distinct damping treatments offered in previous studies [12]. In Ref. [13] finite-element and Ritz formulations are formulated for a sandwich beam with uniform MR-fluid treatment but several boundary conditions, and indicated their validity through experiments conducted on a cantilever sandwich beam. The research also offered non-linear relationships between the complex shear modulus of the MR fluid and the enforced magnetic field on the basis of the laboratory measured free vibration response.

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The mentioned studies have assumed uniform MR-fluid layer subject to a uniform magnetic field. A few recent studies have indicated that a non-uniform MR-fluid treatment could be promoting in fixing the deflection response for a transverse excitation [14]. The former studies, still, employed distinctly different methods. In Ref. [14] the dynamic responses of a MR sandwich cantilever beam has been experimentally studied subject to a uniform and non-uniform magnetic field. The non-uniform magnetic field was known by distributing five sets of permanent magnets over the whole surface of the beam, as the field intensity of each magnet was independence. It was resolved that the natural frequency of the beam rises down as the permanent magnets are went away from the fixed support. Ref. [13] introduced finite element formulations for a partially-treated MR fluid sandwich beam containing different MR-fluid segments for different boundary conditions. The free and forced vibration responses of different configurations of partially-treated MR-fluid beams were gained for various lengths and number of fluid segments. The article also executed laboratory experiments to demonstrate validity of the analytical formulations and concluded that the location and length of the MR fluid segments have substantial influence on the natural frequencies and the loss factors, plus to the intensity of the magnetic field and the boundary conditions.

The execution of smart structure for active vibration control is powerfully relies on the control algorithm. A study on various control algorithms employed for active vibration control study for smart structures is presented in [16]. The linear quadratic regulator (LQR) is employed for vibration control study for smart beams and plates [17], [18], [19]. The main disadvantage of LQR control is that it involves the measurement of all the states variables. Only limited attempts have been done towards synthesis of semi-active and active controllers for the ER/MR-fluid treated sandwich beams, although the controller design for structures employing piezoelectric actuators have been widely reported [20], [21]. Demonstration of the finite element modeling of the MR sandwich composite plate into state space form as a function of magnetic field have not yet been researched. In the present study, a semi active control synthesis is formulated to control the dynamic characteristics of the fully and partially treated MR sandwich plates. The governing equations of motion of the three layers MR sandwich composite plate derived to the finite element form are presented in the state variable form, and an observer-based linear quadratic regulator (LQR) optimal control is studied.

II. DYNAMIC MODEL OF MULTI-LAYER COMPOSITE PLATE

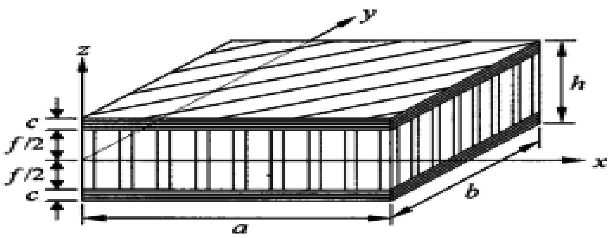

Figure 1. Fully treated MR sandwich composite plate [4]

The dynamic governing equations and uniform boundary condition of the smart composite plate with MR fluids mid-layer, is formulated using extended Hamilton's principle [15].

$$\delta \int_{t_1}^{t_2} (T - U) = 0 \qquad (1)$$

where T is the kinetic energy, U is the strain energy. Various energy terms in equation (3) are set as:

$$T = \int_A \frac{1}{2} \rho \{\dot{u}\}^T \{\dot{u}\} dA \qquad (2)$$

$$U = \int_A \frac{1}{2} \{\sigma\}^T \{\varepsilon\} dA + \int_{A_f} \frac{1}{2} \{\tau_f\}^T \{\gamma_f\} dA \qquad (3)$$

where u is displacement vector. Dot over a variable represents time derivative, ε is strain tensor, σ is stress tensor and A is the plate area of entire plate, γ_f is shear strain tensor of the fluid and τ_f is shear stress tensor of the fluid. Buna-N rubber is assumed as a sealer material on the edges of the MR-fluid layer segment to include the MR fluid within the two composite layers of the sandwich plate and keep consistent thickness. The rubber seal and the MR fluid, yet, are considered as a homogeneous material layer with equivalent shear modulus formulated as [13]:

$$G = G_r \left(\frac{b_r}{b}\right) + G^* \left(1 - \frac{b_r}{b}\right) \qquad (4)$$

where G is the total shear modulus of the homogeneous layer, b_r and b are the widths of the rubber and the entire plate, respectively, and G_r and G^* are the shear modulus of the rubber and MR fluid, accordingly. The shear stress–shear strain properties of MR fluids have been described in many studies [22], [23] and characterized by two distinguished regions, referred to as ‘pre-yield’ and ‘post-yield’ regions, as shown in Figure 2. MR materials show different levels of stress and strain under influence of applied magnetic field and continue a similar pattern in its rheological behavior. In the pre-yield regime, the MR material indicates visco-elastic behavior and is defined as the complex modulus as [22].

$$G^*(B) = G'(B) + iG''(B) \qquad (5)$$

While the storage modulus $G'(B)$ is proportional to the average energy maintained during a cycle of deformation per unit volume of the material, the loss modulus $G''(B)$ is proportional to the energy dissipated per unit volume of the material over a cycle. So, both the modulus are functions of the magnetic field intensity B. The post-yield behavior of MR materials is more or less characterized by Bingham plastic model, such that [23].

$$\tau = \tau_y + \eta\dot{\gamma} \qquad (6)$$

where τ is the shear stress, τ_y is the magnetic field induced dynamic yield stress, η is the plastic viscosity and $\dot{\gamma}$ is the shear strain rate. Because of the application of the magnetic field through MR fluid, the ferrous particle set aside in the viscous fluid become particle chains and yield stress is then promoted [24]. Therefore, both storage and loss modulus will rise up with the increasing of the magnetic field. Accordingly the stiffness and damping characteristics can be designed using the applied magnetic field.

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This modifies a useful mechanism to suppress the vibration of the structural systems [25]. Assume a composite sandwich plate with length a, width b, and total thickness h as shown in Figure 1. The sandwich plate is considered to consist of two balanced laminated face sheets of the same thickness c and and MR fluid core of thickness f. The face sheets are made with $N_f$layer orthotropic laminates. The total Transverse displacement of the plate is considered to be the sum of the displacement due to bending of the plate, w_b, and due to shear deformation of the core, w_s. The displacement field may be formulated as:

$$u(x,y,z) = u_o(x,y) - zw_{b,x} \quad (7)$$

$$v(x,y,z) = v_o(x,y) - zw_{b,y} \quad (8)$$

$$w(x,y,z) = w_b(x,y) + w_s(x,y) \quad (9)$$

where u_0 and v_0 are the uniform in-plane displacement element of the plate. Using the von Karman large deflection as-assumptions, the kinematic relation can be presented as [4]:

$$\begin{Bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma_{xy} \end{Bmatrix} = \begin{Bmatrix} u_{0,x} \\ u_{0,y} \\ u_{0,y} + v_{0,x} \end{Bmatrix} + z\begin{Bmatrix} -w_{b,xx} \\ -w_{b,yy} \\ -2w_{b,xy} \end{Bmatrix} + \begin{Bmatrix} \frac{1}{2}(w_{b,x} + w_{s,x})^2 \\ \frac{1}{2}(w_{b,y} + w_{s,y})^2 \\ (w_{b,x} + w_{s,x})(w_{b,y} + w_{s,y}) \end{Bmatrix} = \{\epsilon_0\} + z\{\kappa\} + \{\delta\} \quad (10a)$$

$$\begin{Bmatrix} \gamma_{yz} \\ \gamma_{zx} \end{Bmatrix} = \begin{Bmatrix} w_{s,y} \\ w_{s,x} \end{Bmatrix} = \{\gamma\} \quad (10b)$$

where $\{\epsilon_0\}$, $\{\kappa\}$, $\{\delta\}$, $\{\gamma\}$ are the midplane strain, plate curvature, large deflection strain and transverse shear strain, accordingly. The stress-strain relation of the sandwich plate is shown by:

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{Bmatrix}_k = \begin{bmatrix} \bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\ \bar{Q}_{12} & \bar{Q}_{22} & \bar{Q}_{26} \\ \bar{Q}_{16} & \bar{Q}_{26} & \bar{Q}_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{Bmatrix} \quad (11)$$

$$\begin{Bmatrix} \tau_{yz} \\ \tau_{zx} \end{Bmatrix} = \begin{bmatrix} \bar{Q}_{44} & \bar{Q}_{45} \\ \bar{Q}_{45} & \bar{Q}_{55} \end{bmatrix} \begin{Bmatrix} \gamma_{yz} \\ \gamma_{zx} \end{Bmatrix} \quad (12)$$

Where $\bar{Q}_{ij}$ is the transformed reduced stiffness and k shows the k_{th} layer of the face. For sandwich plate having MR fluid core, it is fair to consider that the transverse normal stiffness of the core is infinitely large and the core makes no contribution to the bending and membrane stiffness of the sandwich plate. However, the shear strains in the core of the sandwich need to be known due to a low transverse modulus of rigidity of the core. The force and moment outcomes of the sandwich plate are then formulated as:

$$\{N\}^T = [N_x \quad N_y \quad N_{xy}] = \int_{-f/2-c}^{-f/2} [\sigma_x \quad \sigma_y \quad \tau_{xy}]\, dz + \int_{f/2}^{f/2+c} [\sigma_x \quad \sigma_y \quad \tau_{xy}]\, dz \quad (13)$$

$$\{M\}^T = [M_x \quad M_y \quad M_{xy}] = \int_{-f/2-c}^{-f/2} [\sigma_x \quad \sigma_y \quad \tau_{xy}]\, zdz + \int_{f/2}^{f/2+c} [\sigma_x \quad \sigma_y \quad \tau_{xy}]\, zdz \quad (14)$$

$$\{V\}^T = [V_y \quad V_x] = \int_{-f/2}^{f/2} [\tau_{yz} \quad \tau_{zx}]\, dz \quad (15)$$

which goes to:

$$\{N\} = [A]\{\varepsilon^0\} + [A]\{\delta\} \quad (16)$$

$$\{M\} = [D]\{\kappa\} \quad (17)$$

$$\{V\} = [S]\{\gamma\} \quad (18)$$

$$A_{ij} = 2\sum_{k=1}^{N_f} \bar{Q}_{ij}\,(h_k - h_{k-1}) \quad i,j = 1,2,6 \quad (19)$$

$$D_{ij} = \frac{2}{3}\sum_{k=1}^{N_f} \bar{Q}_{ij}\,(h_k^3 - h_{k-1}^3) \quad i,j = 1,2,6 \quad (20)$$

$$S_{ij} = \bar{Q}_{ijC} \quad i,j = 1,2,6 \quad (21)$$

The total strain energy can be indicated as the sum of the energies due to in-plane strain, U_i, bending strain, U_b, shear strainU_s, third-order nonlinear part, U_3, fourth-order nonlinear part, U_4, and MR fluid layer, U_f as:

$$U = U_i + U_b + U_s + U_3 + U_4 + U_f \quad (22)$$

$$U = \frac{1}{2}\int_A \{\varepsilon_0\}^T [A]\{\varepsilon_0\} dA + \frac{1}{2}\int_A \{\kappa\}^T [D]\{\kappa\} dA + \frac{1}{2}\int_A \{\gamma\}^T [S]\{\gamma\} dA + \frac{1}{2}\int_A \{\delta\}^T [A]\{\varepsilon_0\} + \{\varepsilon_0\}^T [D]\{\delta\} dA + \frac{1}{2}\int_A \{\delta\}^T [A]\{\delta\} dA + \frac{1}{2}\int_A \{\gamma_f\}^T [A_f]\{\gamma_f\} dA \quad (23)$$

The kinematic energy [4] may be defined as:

$$T = \frac{\rho_t h}{2}\int_A (w_{b,t}^2 + 2w_{b,t}w_{s,t} + w_{s,t}^2) dA + \frac{I_b}{2}\int_A (w_{b,xt}^2 + w_{b,yt}^2) dA + \frac{I_{bs}}{2}\int_A 2(w_{b,xt}w_{s,xt} + w_{b,yt}w_{s,yt}) dA + \frac{I_s}{2}\int_A (w_{s,xt}^2 + w_{s,yt}^2) dA \quad (24)$$

where $\rho_t h = 2\rho_c c + \rho_f f$, $I_b = 2(I_c + \rho_c f e^2) + I_f$, $I_{bs} = 2I_c + I_f(^c/_f)$, $I_s = 2I_c + I_f\left(\frac{c}{f}\right)^2$, $I_f = \frac{1}{12}\rho_f f^3$, $I_c = \frac{1}{12}\rho_c c^3$, $e = \frac{1}{2}(f + c)$

and ρ_t, $\rho_c$and ρ_f are the densities of the sandwich plate, composite face and MR fluid core, respectively.Finite element formulation of MR sandwich plate

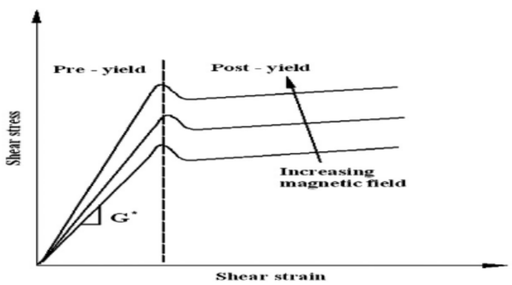

Figure 2. Shear stress–shear strain relationship of MR materials under varying intensities of magnetic field [26]

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Consider a 7-DOF rectangular element with thickness h. The total transverse displacement of the plate is considered to be the sum of the displacement due to bending of the plate and that due to shear deformation of the core. For simplicity, the four displacement functions for the in-plane, bending of the plate and that due to shear deformation are considered to have the same form. These displacement functions for u, v, w_b, and w_s, can be demonstrated in the local $\psi - \varphi$ coordinate system [4].

$$u = \sum_{i=1}^{7} \alpha_i \psi^{m_i} \varphi^{n_i} \quad (25)$$

$$u = \sum_{i=1}^{7} \beta_i \psi^{m_i} \varphi^{n_i} \quad (26)$$

$$w_b = \sum_{i=1}^{7} \gamma_i \psi^{m_i} \varphi^{n_i} \quad (27)$$

$$w_s = \sum_{i=1}^{7} \zeta_i \psi^{m_i} \varphi^{n_i} \quad (28)$$

Where $m_i = (0,2,1,0,3,2,1)$ and $n_i = (0,1,2,0,1,2,3)$ and α_i, β_i, γ_i, ζ_i are constants to be decided by using conditions at nodal point, By continuing the strategy in [17], the displacement functions with reference to the global coordinate can be showed in terms of the element nodal displacement vector $\{q\}$. As substituting (25), (26), (27) and (28) into (22) and (24) and afterwards into Lagrange's equations described as:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial q_i} + \frac{\partial V}{\partial q_i} = Q_i \quad i = 1,2,n \quad (29)$$

The governing equations of motion for the undamped partially or fully treated MR sandwich plate element in the finite element form can be gained as:

$$[m^e]\{\ddot{d}\} + [k^e]\{d\} = \{f^e\} \quad (30)$$

Where n is the total DOF considered in the formulation and Q_i is the generalized force in respect to the i_{th} DOF,$\boldsymbol{m^e}$ and $\boldsymbol{k^e}$ are the element mass and stiffness matrices, accordingly, and $\boldsymbol{f^e}$ is the element force vector. Setting up the mass and the stiffness matrices and the force vector for all the elements, issues the global governing equations of motion of MR sandwich plate which can be shown in the finite element form as:

$$[M]\{\ddot{d}\} + [K]\{d\} = \{F\} \quad (31)$$

where M, K and F are the global system mass and stiffness matrices and global force vector, accordingly. For the partially treated sandwich plate, the matrices M and K are derived by enforcing compatibility conditions which are identical transverse and axial displacement and the slopes at the interfaces of the composite material and MR-fluid segments within the mid-layer of the plate.

III. DESIGN OF AN OPTIMAL CONTROL

Control of vibration in a model is normally concerned with particular modes of vibration. It may then be conquered to express (31) in the modal form using the modal coordinates, which would proceed uncoupled governing equations of motion for the MR sandwich plate. Considering proportional damping, (31) can be shown as the following form [30].

$$\{\ddot{\eta}_i\} + [2\xi_i\omega_i]\{\dot{\eta}_i\} + [\omega_i^2]\{\eta_i\} = \{f_i\}, \ i = 1,2,\dots,n \quad (32)$$

where $\{\eta\}$ is the modal coordinate vector, which is concerned to the modal matrix $[q]$ that $\{d\} = [q]\{\eta\}$, ξ_i is the modal damping ratio for the i_{th} normal mode, ω_i is representing natural frequency of the system without considering the structural damping, and $f_i = [q]^T \{F\}$. Damping factor and natural frequencies of the plate would differ with the applied magnetic field; (32) will then be indicated as a function of the controlled magnetic field u_i:

$$\{\ddot{\eta}_i\} + C'(u_i)\{\dot{\eta}_i\} + K'(u_i)\{\eta_i\} = \{f_i\}, \ i = 1,2,\dots,n \quad (33)$$

where $K'(u_i) = [\omega_i^2]$ and $C'(u_i) = [2\xi_i\omega_i]$.

It has been known that the natural frequencies of the sandwich plate differ linearly with the applied magnetic field [13]. The variation of the factors, $K'(u_i)$ and $'(u_i)$, with the applied magnetic field in range of 0- 4000 Gauss is studied for different modes of vibration and the results demonstrated that both factors vary approximately linearly with the magnetic field. Instead for mode 1, the variation of $K'(u_i)$ and $C'(u_i)$ according to the applied magnetic field is indicated in Figure 2, which clearly shows that, these factors closely, differ linearly according to the applied magnetic field. Note for the interest of simplicity $K'(u_i) = [\omega_i^2]$ and $C'(u_i) = [2\xi_i\omega_i]$, are presumed to be linear functions of the applied magnetic field u_i that:

$$K'(u_i) = \alpha_{ki} + \beta_{ki}u_i \quad ,0 \le u_i \le u_{max} ;$$

$$C'(u_i) = \alpha_{ci} + \beta_{ci}u_i \quad ,0 \le u_i \le u_{max} \quad (34)$$

where u_{max}denotes the maximum magnetic field that can be applied, which is limited because of the saturation limit of the MR fluid used. The coefficients α_{ki} ,β_{ki} and α_{ci} ,β_{ci} are functions of the applied magnetic field refering to each mode, u_i. In this report, a linear quadratic regulator (LQR) based controller synthesis is considered using the full state feedback. The finite element models of the fully- and partially- treated plates are then shown in the state-space form, as:

$$\{\dot{x}_S\} = [A]\{x_S\} + [B]\gamma + \{f\} \ and \ \{y_S\} = [C]\{x_S\} \quad (35)$$

where $[A] = \begin{bmatrix} [0] & [I] \\ [-\alpha_{ki}] & [-\alpha_{ci}] \end{bmatrix}$;

$$[A] = \begin{bmatrix} [0] & [I] \\ [-\beta_{ki}] & [-\beta_{ci}] \end{bmatrix}; [C] = \begin{bmatrix} I & 0 \\ 0 & 0 \end{bmatrix};$$

$$\{\gamma\} = [U]\{x_S\}; \ [U] = \begin{bmatrix} [u_i] & [0] \\ [0] & [u_i] \end{bmatrix}$$

Is the control input matrix; $\{f\} = \begin{bmatrix} [0] \\ [f_i] \end{bmatrix}$; and $\{x_S\} = \{\{\eta\},\{\dot{\eta}\}\}^T$ is the state vector; and $\{y_S\}$is the response vector.

In [35] the generalized excitation force vector $\{f\}$ and the matrix $[U]$ define the inputs, while $\{y_s\}$is the output vector.

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In the closed-loop form, the control input vector $\{\gamma\}$ in (35) is referred to state feedback vector, as:

$$\{\gamma\} = -[U]\{x_S\} \quad (36)$$

where the magnetic field matrix $[U]$ does as the control gain matrix, which is assessed according to the desired control law, as negative sign shows externally applied control magnetic field. By substituting for $\{\gamma\}$, the closed loop system could be derived in state space form, as:

$$\{\dot{x}_S\} = ([A] - [B][U])\{x_S\} + \{f\} \quad (37)$$

From [37] it can be stated that the control gain matrix $[U]$ directly relates to the system damping, $C'(u_i)$, and stiffness, $K'(u_i)$, properties. An optimal control is formulated through minimization of a cost function that is proportional to a measure of the system's response and the desired control inputs using the linear quadratic (LQR) approach [27], such that:

$$J = \frac{1}{2}\int_0^{\infty}(\{x_S\}^T[Q]\{x_S\} + \{\gamma\}^T[R]\{\gamma\})\, dt \quad (38)$$

where $[Q]$ and $[R]$ are the symmetric semi-definite and positive-definite weighting matrices, accordingly. The magnitudes of $[Q]$ and $[R]$ are chosen so as to gain an optimal tradeoff between the vibration response and the intensity of the control magnetic field. As $[Q]$ describes the relative weight of each state variable, $[R]$ described the relative weight of control magnetic field [28]. Minimization of (38) yields a linear full-state feedback control law, $\{\gamma\} = -[U]\{x_S\}$, where the control gain matrix, $[U] = [R]^{-1}[B]^T[P]$ is scoped by solving for $[P]$ from the following algebraic Ricatti equation [29]:

$$[A]^T[P] + [P][A] + [Q] - [P][B][R]^{-1}[B]^T[P] = 0 \quad (39)$$

IV. RESULT AND DISCUSSION

The fundamental characteristics of the fully-and partially-treated MR sandwich beam were studied theoretically and experimentally and reported in [13], accordingly. The permanent magnets were utilized in experimental set-up to produce the desired magnetic field over the surface of the MR sandwich beam by varying the gap between the magnets and the structure. The realization of a controllable MR sandwich structure, however, would require designs of compact electro-magnets in order to apply the desired field over the surface of the beam. The designs of compact electro-magnets have been realized for MR fluid dampers employed in vehicle suspensions, where the electro-magnet is integrated within the damper piston [31]. The design of electromagnets for MR sandwich structures, however, takes extra issues considering the larger size of the structure compared to the damper piston. Therefore, further efforts are needed to design compact yet high intensity electromagnets for practical implementation of the controller. This study, however, is limited to simulations to evaluate the effectiveness of the developed full state optimal LQR controllers in suppressing the vibration of the fully- and partially-treated MR sandwich plate with clamped-free boundary conditions under a unit impulse load applied at the tip of the plate. One of the most common forms of fiber-reinforced composite materials is the cross plied laminate, in which the fabricator lays up a sequence of unidirectional reinforced plies. Each ply is typically a thin (approximately 0.1 *mm*) sheet of collimated fibers impregnated with an uncured epoxy or other thermosetting polymer matrix material. The orientation of each ply is arbitrary, and the layup sequence is tailored to achieve the properties desired of the laminate. For the finite element analysis, all of the laminated composite plates have a rigid constraint across the cross-sections at the loading end. The plates are fully constrained on the other end to simulate the symmetric part of loading or built-in end condition (fixed free). A simple plate under a downward load is modeled to find the stiffness of the plate. The plate is chosen to be 100 mm long and 100 mm wide with applying one force of 1 *N* on one end (Fig. 2). The other side of the plate is fully constrained. The stacking sequence of the each layer of the 3-layer plate is [0 0 45 90 90 90 90 45 0 0]. Model for the plate contains 144 elements. Next, the finite element model will assess the performance of different types of laminated plate. Here, the selected fiber is T-300 Carbon while the polymer matrix material is Epoxy 3501-6. The mechanical properties of T-300 Carbon and Epoxy 3501-6 are listed below in Table 1:

Table 1 Mechanical Properties of Composite Plate

Mechanical Property	
Fiber Elasticity (E11)	15 GPa
Fiber Elasticity (E22)	27 Gpa
Fiber Shear modulus (G12)	0.23
Fiber Poisson ratio (v12)	4.30 GPa
Matrix Elasticity (Em)	1.60 GPa
Matrix Shear modulus (Gm)	0.35
Matrix Poisson ratio (vm)	1.27 g/cm3
Composite Density (ρm)	230 GPa

The volume fraction of fiber is assumed as 0.6; hence the volume fraction of polymer matrix is 0.4. Here, the considered each layer of laminated plate has 10 layers of lamina with a thickness-dimension ratio 0.01. The total area of the multi- layer plate is separated into 9 segments of equal area, while the MR fluid treatment is enforced to selected segments of the structure. The remaining segments of the plate are considered to be of composite material. The total length of the MR fluid layer in all the configurations is considered to be constant to ease relative property analyses.
The simulation results are obtained by considering identical baseline thickness of 1 mm of the elastic and fluid layers, while the material and all other properties of the layers are identical. The validity of the developed finite element formulations for the different configurations considered has been demonstrated by comparing the natural frequencies corresponding to the first three modes obtained through FEM under simply supported end conditions without any magnetic field. The location of the fluid segments were chosen on the basis of an earlier study [32] among 6 configurations and denoted as configurations A and configuration B, as shown in Figure3, respectively. The configuration A and B were shown to yield maximum damping factors corresponding to the first four bending modes and suppressing

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the deformation of them, when subjected to a constant magnetic field. Furthermore, the complex shear modulus of the MR fluid was considered to be a function of the storage and loss moduli(5).

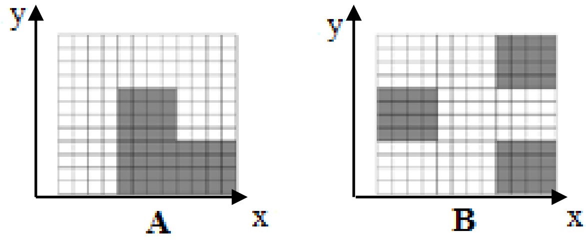

Figure 3. Selected configurations of partially treated MR sandwich beams with maximum modal damping factors corresponding to first four modes under a constant magnetic field

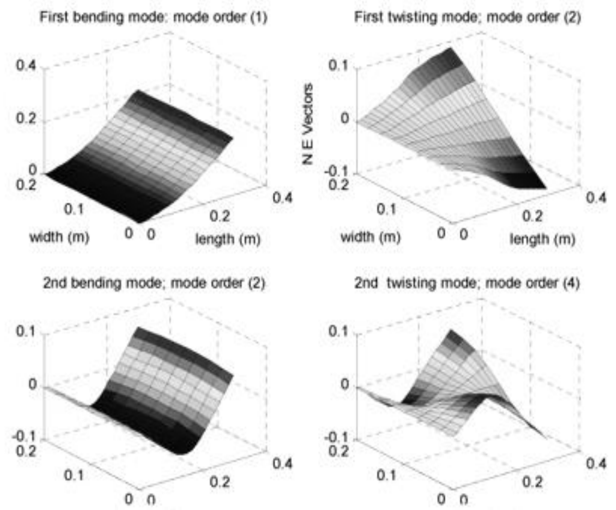

Figure 4. First four mode shapes of passive composite [15]

V. RESPONSE TO IMPULSE DISTURBANCE

The free vibration response of the closed-loop fully and partially-treated plates were designed under an impulse force applied at the free end of the clamped-free plate. The results are presented to show the use of the LQR controller design formulated in (38). The weighting matrices $[Q]$ and $[R]$ of the controller were defined through minimization of a composite performance function of the peak displacement along the z-axis and the settling time, such that

$$\text{Minimize } f(x) = \alpha_1 t_s + \alpha_2 d_z(l) \tag{40}$$

Where t_s is the settling time, $d_z(l)$ is the deflection at the tip along the z-axis (transverse deflection), and α_1 and α_2 are the constant weighting factors. The settling time was identified as the displacement responses of the plate. Limit constraint were also imposed on $[R]$ and $[Q]$, as

$$[R_l \leq [R] \leq R_u]; \text{ and } [Q_l \leq [Q] \leq Q_u]; \tag{41}$$

The limiting values were indentified trough a parametric study involving the effects of variations in $[R]$ and $[Q]$ on t_s and $d_z(l)$ together with required field intensity. It should be noted that the minimization problem was solved assuming different weighting factors ranging from 0 to 1 $(\alpha_1 + \alpha_2) = 1$ and different values of the starting vectors. The time history and the frequency amplitude of the tip deflection responses of the controlled fully-treated MR sandwich plate to an impulse excitation are compared with those of the passive plate ($B = 0$) Figure 5 and 6. From the results, it is proven that the LQR control algorithm can substantially suppressed the tip displacement with the permissible intensity of the control magnetic field.

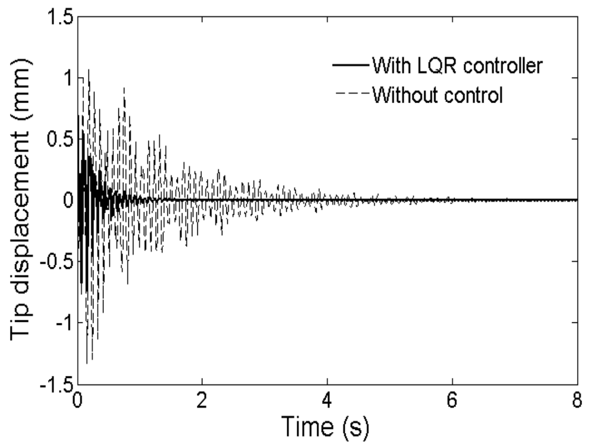

Figure 5. The tip deflection responses MR sandwich plate for fully treated plate

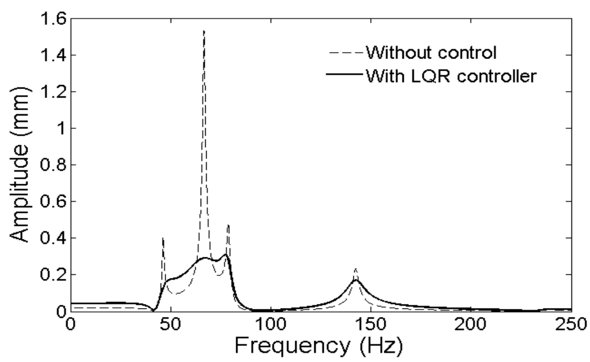

Figure 6. Amplitude spectrum of the fully treated plate

The results show that the settling time of the controlled plate is in order of 1.2 s, which is significantly lower than 4.94 s of the passive plate. The amplitude spectra of the tip displacement response, illustrated in figure 5 and 6 also shows substantially lower deflections of the controlled plate corresponding to all of the modes observed up to 250 Hz. The results show that the amplitudes corresponding to the first two natural frequencies of the controlled fully-treated plate are, lower than those of the uncontrolled plate.

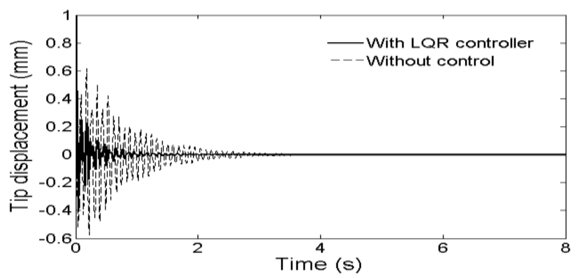

Figure 7. Tip deflection responses MR sandwich plate configuration A

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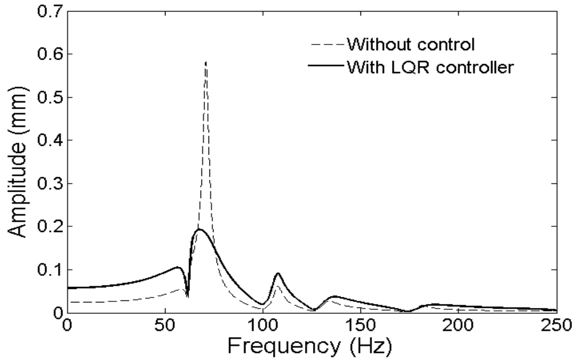

Figure 8. Amplitude spectrum configuration A

The time response and amplitude of tip displacement responses of configuration B of the controlled as well as passive partially-treated sandwich plates are observed in Figures 7 and 8 respectively. The greater treatment near the free end goes larger reduction in the tip displacement, while the settling time inclines to be quietly higher for both the controlled as well as passive plates compared to the fully-treated and configuration A of the partially-treated plates. The execution of the LQR control algorithm proceeds peak tip displacement of 0.42 mm and settling time of 0.87 s. The relative values for the passive plate are 0.65 mm and 3.1 s, respectively. The amplitudes referring to the first three natural frequencies of the controlled structure are 19%, 21% and 68%, respectively, lower than those of the passive plate. The deflection amplitudes of the controlled configuration B are significantly lower than those of the controlled configuration A. This is especially traceable to relative higher modal damping factors of configuration B.

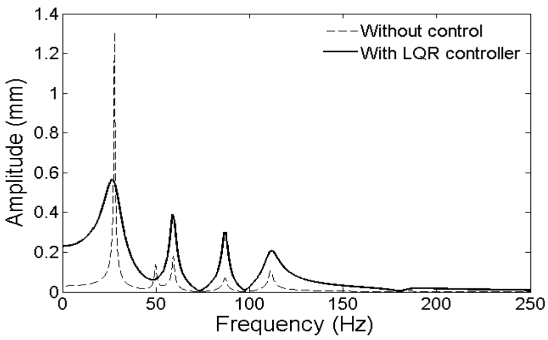

Figure 9. Tip deflection responses MR sandwich plate configuration B

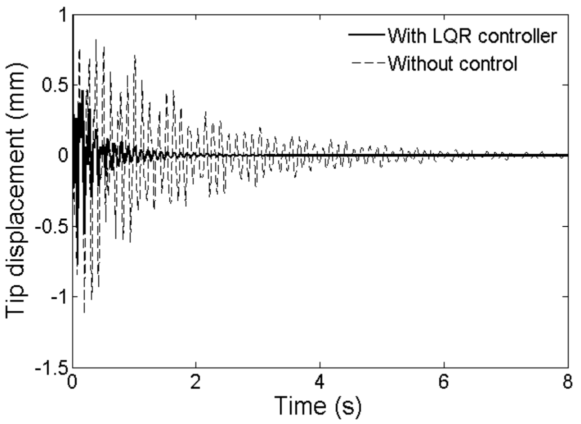

Figure 10. Amplitude spectrum configuration B

Furthermore, the peak displacement response at the tip of the passive partially treated plate (configuration B) is almost in comparison to that of the fully treated passive plate, as seen from the time-history of the responses, figure 9 and 10. The results also indicate that the execution of the LQR control to configuration B proceeds substantially higher reductions in deflection amplitudes corresponding to the first three modes compared to those of observed for the fully treated and configuration A of the partially treated plate.

VI. CONCLUSION

In this article, the semi-active vibration control of a multilayer composite plate fully-and partially –treated with MR fluid segments has been studied. The results of the analysis indicate that the full-state observer based LQR control can ease up significant reductions in the settling time and free-and forced vibration responses of the plates treated either fully or partially with MR-fluid segments under an impulse excitation. The LQR control of the fully and partially treated plates resulted nearly 67% reduction in the free-vibration settling time, and 50% reduction in the tip deflection in comparison with that of the passive plates. The execution of the controller, however, would require promotion in compact electromagnets to apply the controlled magnetic field on the surfaces of the treated structures.

REFERENCES

1. Sh. D. Pawar, A. Utpat, “Analysis of Composite laminate for maximum stiffness, IJRTE May 2014, Vol 3, pages 60-66.
2. A.N. Bambole, Y.M. Desai, “Hybrid-interface finite element for laminated composite and sandwich beams”. Finite Elements in Analysis and Design, 2007, Vol. 43, pages 1023-36.
3. H. Charles, L.M. Rochel Accorsi, “A new finite element for global modeling of delaminations in laminated beams”. Finite Elements in Analysis and Design 1998, Vol.31, pages 160-177.
4. L. Shiau, Sh. Kuo, “Free vibration of thermally buckled composite sandwich plates”. Journal of vibration and acoustics 2006, Vol. 128, pages. 1-7
5. U. Lee, J. Kim, “Spectral element modeling for the beams treated with active constrained layer damping”, Intl. J. of Solids and Struct. 2001, Vol. 38, pages. 5679-5702.
6. G. Wang, N. M. Wereley, “Spectral finite element analysis of sandwich beams with passive constrained layer damping”, J. of Vib. and Acoustics, 2002 Vol. 124(3), pages 376-386.
7. Jr. B F Spencer, S. Nagarajaiah, “State of the art of structural control”, Journal of Struct. Eng., 2003 , Vol. 129(7), pages. 845-856.
8. Y. Liu, T. P. Water and M. J. Brennan, “A comparison of semi-active damping controlstrategies for vibration isolation of harmonic disturbances”, J. of Sound and Vib.,2005, Vol. 280(1-2), pages. 21-39.
9. H See, “Advances in electro-rheological fluids: Materials, modeling and applications”,Journal of Indust. and Eng. Chemistry, 2004,Vol. 10(7), pages1132-1145.
10. G. Z. Yao, F. F. Yap, G. Chen, W. H. Li and S. H. Yeo, “MR damper and its application for semi-active control of vehicle suspension system, Mechatronics”, 2002, Vol. 12(7), pages. 963-973.
11. Y . Ahn, B. Yang, M. Ahmadian and S. Morishita, 2005, “A Small sized variable damping mount using magnetorheological fluid” Journal Intell. Mater. Sys. Struct., 2005, Vol. 16(2), pages. 127-133.
12. T. Pranoto, K. Nagaya, and A. Hosoda, “Vibration suppression of plate using linear MR fluid passive damper”, J. of Sound and Vib., 2004, Vol. 276 (3-5), pages. 919-932.
13. V. Rajamohan, S. Rakheja and R. Sedaghati, “Vibration analysis of a partially treated multi- layer beam with magnetorheological fluid”, Journal of Sound and Vib., 2010, Vol. 329(17), pages 3451-3469.

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14. V. N. Lara-Prieto, R. Parkin, M. Jackson, V. Silberschmidt and Z. Kesy, "Vibration characteristics of MR cantilever sandwich beams: Experimental study", Smart Mater. And struct., 2010, Vol. 19(1), pages 015-005 (9pp).
15. M. Yaqoob Yasin, N. Ahmad, M. Naushad Alam, "Finite Element analysis of actively controlled smart plates with patched actuators and sensors". Latin Journal of Solids and Structures, 2010, Vol. 7.
16. R. Alkhatib and M. F. Golnaraghi. "Active structural vibration control: a review". Shock and Vibration Digest, 2003, Vol. 35(5), pages 367–383
17. K. R. Kumar and S. Narayanan, "The optimal location of piezoelectric actuators and sensors for vibration control of plates" Smart Materials and Structures, 2007, Vol.16 pages 2680–2691
18. S. Narayanan and V. Balamurugan, "Finite element modelling of piezo laminated smart structures for active vibration control with distributed sensors and actuators", Journal of Sound and Vibration, 2003, Vol. 262, pages 529–562
19. A. Zabihollah, R. Sedagahti, and R. Ganesan, "Active vibration suppression of smart laminated beams using layerwise theory and an optimal control strategy", Smart Materials and Structures, 2007, Vol.16, pages 2190–2201.
20. A. M. Sadri, J. R. Wright and R. J. Wynne, "Modelling and optimal placement of piezoelectric actuators in isotropic plates using genetic algorithm", Smart Mater. and Struct., 1999, Vol. 8(4), pages 490-498.
21. Q. Hu and G. Ma, "Variable structure control and active vibration suppression of flexible spacecraft during attitude maneuver", Aerospace Science and Tech.,2005, Vol. 9(4), pages 307-317.
22. W.H. Li, G. Chen, S. Yeo, "Viscoelastic properties of MR fluids", Smart Materials and Structures, 1999, Vol. 8 pages 460–468
23. Y. T. Choi, N. M. Wereley and Y. S. Jeon, "Semi-active vibration isolation using magnetorheological isolators", Journal of Aircraft, 2005, Vol. 42 (5), pages1244-1251.
24. F.D.Goncalves, J.H.Koo, M.Ahmadian, "A review of the state of the art in magnetorheological fluid technologies—part I: MR fluid and MR fluid models", The Shock and Vibration Digest, 2006, Vol. 38 pages 203–219.
25. M, Yalcintas and H. Dai, "Magnetorheological and electrorheological materials in adaptive structures and their performance comparison", 1999, Smart Mater. and Struct., Vol 8(5), pages 560–573.
26. D.J.Mead, S.Markus,"The forced vibration of a three-layer, damped sandwich beam with arbitrary boundary conditions", Journal of Sound and Vibration, 1969, Pages163–175.
27. V. Sethi and G. Song, "Optimal vibration control of a model frame structure using piezoelectric sensors and actuators, J. of Vib. and Contl", 2005, Vol. 11(5), pages 671-684.
28. W. H. Liao and D. H. Wang, "Semi-active vibration control of train suspension systems via magnetorheological dampers", 2003, Journal of Intell. Mater. Sys. and Struct. , Vol.14 pages161-172.
29. A. G. O. Mutambara, 1999, Design and analysis of control systems, CRC press, Florida.
30. V. Rajamohan, R. Sedaghati, S. Rakheja, "Optimal vibration control of beams with total and partial-treated MR fluid treatments". Journal of Smart. Mater. Struct., 2011, Vol 20
31. S.J. Dyke, B.F. Spencer, M.K. Sain and J.D. Calrson, "An Experimental Study of MR Dampers on Seismic Protection", Smart Materials and Structures, 1998, Vol. 7(5) pages 693-703.
32. Z. S. Fattahi, A. Zabihollah, "Vibration of Laminated Composite Structures Integrated with Magnetorheological Fluid segments". Journal of Modares Mechanical Engineering, 2013, Vol. 13, pages 156-160

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