Stiffness-Damping Matching Modelling for Vibration Isolation System of Roadheader ECB

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The electronic control box (ECB) is a key and precise component of a roadheader. The vibration of the ECB is an increasingly prominent issue as the machine capacity grows. In order to promote the isolation effect of the ECB, a whole-body vibration model considering the cutting effect is derived, based upon which the stiffness-damping matching strategy for the ECB isolator is acquired. For engineering application, the tubular constrained damping isolator (TCDI) is developed based on the constrained damping theory and the matching strategy. The theoretical results show that the isolation effect of the ECB isolator strengthens as the stiffness coefficients decline and the damping coefficients increase. The configurations with larger rear stiffness coefficients and larger front damping coefficients could lead to a better vibration control effect. The experiment results indicate that the TCDI exhibits a greater capability of isolating the impact excitation than the traditional E-type isolator, thus verifying the whole-body vibration model for the roadheader and the matching strategy for the ECB isolator. This research can provide theoretical and practical references for the investigation of the dynamic behaviour of complex viscoelastic structures.

NOMENCLATURE

ECB	Electronic control box	c_6	The front damping coefficient of ECB isolator in N·s/m
FEM	Finite element method	l_1	The horizontal distance of the mass centre of the cutting head to the mass centre of the main body in m
TCDI	Tubular constrained damping isolator	l_2	The horizontal distance of the mass centre of the cantilever to the mass centre of the main body in m
UCS	Uniaxial compressive strength	l_3	The horizontal distance of the mass centre of the rear supporting points of the walking mechanism to the mass centre of the main body in m
m_1	The equivalent mass of the cutting head in kg	l_4	The horizontal distance of the mass centre of the front supporting points of the walking mechanism to the mass centre of the main body in m
m_2	The equivalent mass of cantilever in kg	l_5	The horizontal distance of the mass centre of the rear ECB isolator to the mass centre of the main body in m
m_3	The equivalent mass of the main body in kg	l_6	The horizontal distance of the mass centre of the front ECB isolator to the mass centre of the main body in m
m_4	The equivalent mass of the ECB in kg	l_7	The horizontal distance of ECB isolator to the mass center of ECB in m
z_0	The ground excitation in m	θ_1	The rotation angle of the main body in rad
z_1	The vertical displacement of the cutting head in m	θ_2	The rotation angle of the ECB in rad
z_2	The vertical displacement of cantilever in m	t	The time
z_3	The vertical displacement of the main body in m	q	Variable vector
z_4	The vertical displacement of the ECB in m	T	The kinetic energy
J_1	The rotation inertia of the main body in kg·m ²	V	The potential energy
J_2	The rotation inertia of ECB in kg·m ²	D	The dissipation energy
k_1	The stiffness coefficient between m_1 and m_2 in N/m	$\mathbf{M}$	The mass matrix
k_2	The stiffness coefficient between m_2 and m_3 in N/m	$\mathbf{F}$	The force matrix
k_3	The rear stiffness coefficient between the ground and m_3 in N/m	$\mathbf{C}$	The damping matrix
k_4	The front stiffness coefficient between the ground and m_3 in N/m	$\mathbf{K}$	The stiffness matrix
k_5	The rear stiffness coefficient of ECB isolator in N/m	Z_i	the cutting resistance
k_6	The front stiffness coefficient of ECB isolator in N/m		
c_1	The damping coefficient between m_1 and m_2 in N·s/m		
c_2	The damping coefficient between m_2 and m_3 in N·s/m		
c_3	The rear damping coefficient between the ground and m_3 in N·s/m		
c_4	The front damping coefficient between the ground and m_3 in N·s/m		
c_5	The rear damping coefficient of ECB isolator in N·s/m		

Y_i	The tractive resistance
X_i	The lateral resistance
p_k	The contact strength of the rock in MPa
k_t	The type coefficient of the cutting pick
k_y	The impact factor of the cutting angle
k_s	The geometric impact coefficient
t_d	the average distance of the cutting lines in mm
h	The average cutting thickness in mm
S_i	The projection area of the cutting head along the cutting direction
C_i	($i = 1, 2, 3$) The impact coefficients of the cutting track
n	The number of the pick involving the cutting process
ϕ_i	The position angle of the i th cutting pick in rad
F_x	The component of the cutting force in the x-direction
F_y	The component of the cutting force in the y-direction
F_z	The component of the cutting force in the z-direction
$a_1 = z_1 - z_2$	
$a_2 = z_2 - z_3$	
$a_3 = a_4 = z_3 - z_0$	
$b_1 = \dot{z}_1 - \dot{z}_2$	
$b_2 = \dot{z}_2 - \dot{z}_3$	
$b_3 = b_4 = \dot{z}_3 - \dot{z}_0$	
$b_5 = b_6 = \dot{z}_4 - \dot{z}_3$	

1. INTRODUCTION

The roadheader was first developed for coal excavation in the early 1950s after the patent disclosure by Dr. Z. Ajtay.¹ With decades of development, the roadheader has become a widely-used mechanized excavation method in the tunnel construction, due to its high mobility, versatility, and low capital cost.^{2,3} The roadheader consists of a cutting system, a crawler frame and a control system, as depicted in Fig. 1.⁴ The cutting system, including a cutting head, a cutting reducer and a cantilever, is mounted at the end of the crawler frame and supported by the hoisting cylinder. The crawler frame contains a power system, a muck gathering system, a moving mechanism, and a conveyor. The control system is the brain of the roadheader, which sends out the action introduction to actuators. Among the control system, the electronic control box (ECB), installed at the platform of the main body, plays a key role since the precise electronic components are placed inside.

In recent years, there has been an inevitable trend to develop the heavy-type roadheader with an overall power of up to 400 kW, a weight of an excessive 100 t, and a capability of cutting hard rock in excess of uniaxial compressive strength (UCS) of approximately 350 MPa.⁶ However, this tendency makes the machine suffer from severe vibration, especially in the complex excavating condition.^{5,7} The electronic components in the ECB are particularly vulnerable to vibration, which arouses the urgent research interest in vibration isolation for the ECB. Huang et al.⁸ and Tao⁹ adopted the steel wire isolator for the ECB of roadheaders. Zhao et al. found that increasing the system damping was an effective way to attenuate the vibration level by establishing the rigid-flexible coupled model for roadheaders.¹⁰ Acaroglu et al. analysed the stability of roadheaders with mini-discs in different cutting modes.¹¹ Zhao et al. calculated the transfer characteristics of the vibra-

tion isolation system for the ECB, based upon which a two-level damping system was proposed.¹² Kang et al.¹³ and Balci et al.¹⁴ carried out cutting experiments under variable cutting modes with the full-scale rock to investigate the vibration characteristics of roadheaders.

It was found by an experiment on a roadheader EBZ 160, a heavy duty roadheader used for coal mine tunnelling and subjected to harsh vibration conditions, that the ECB vibration was affected by the integrative action from the whole roadheader body and the cutting process.¹³ The original vibration control plan adopted the E-type isolator (EI) and failed to meet the requirement, since many new problems occurred as the machine capacity grew. The local vibration model only considering the ECB-isolator system was hard to arrive at a better isolation effect. This forms the object of this research. In this paper, the whole-body vibration model considering the cutting effect is derived, based on which the stiffness-damping matching strategy for the ECB isolator is analysed. On the basis of the matching results, a new viscoelastic isolator is developed and its isolation effect is validated by the experiment.

2. THE WHOLE-BODY VIBRATION MODEL OF ROADHEADER

The roadheader is a multi-degree-of-freedom system. For the whole-body vibration modelling, the following hypotheses were made:⁵ 1) the mass distribution along the longitudinal direction is symmetric, 2) the dominant vibration source is from the rock cutting process. Based on the above hypotheses, the two-dimensional (2-D) mechanical model of the roadheader as shown in Fig. 2 was derived by taking the roadheader as a multi-lumped-mass system connected by spring-damping elements. In the 2-D mechanical model, m_1 , m_2 , m_3 , and m_4 in kg, were the equivalent masses of the cutting head, the cantilever, the main body, and the ECB, respectively. The variables z_1 , z_2 , z_3 , and z_4 in m were the vertical displacements of m_1 , m_2 , m_3 , and m_4 , respectively. The variables J_1 and J_2 in kg·m² were the rotation inertia of the main body and the ECB, respectively. The sets k_i ($i = 1, 2, 3, 4, 5, 6$) in N/m and c_j ($j = 1, 2, 3, 4, 5, 6$) in N·s/m were the stiffness coefficients and damping coefficients between the connected masses. The variables l_1 , l_2 , l_3 , l_4 , l_5 , and l_6 in m were separately the horizontal distance of the mass centres of the cutting head, the cantilever, the rear and front supporting points of the walking mechanism, the rear and front ECB isolators to the mass centre of the main body. The variable l_7 in m was the horizontal distance of ECB isolators to the mass center of the ECB. The sets θ_1 , θ_2 in rad were the rotation angles of the main body and the ECB, respectively. The variable z_0 in m was the ground excitation. The variable F_z was the vertical cutting force.

To establish the whole-body vibration model, the Lagrange equation was adopted, which typically takes the form:¹⁵

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{q}} \right) - \frac{\partial T}{\partial q} + \frac{\partial V}{\partial q} + \frac{\partial D}{\partial \dot{q}} = F_z; \quad (1)$$

where $\frac{d}{dt}$ denoted the derivative with respect to time t , $\frac{\partial}{\partial q}$ denoted the partial derivative with respect to q , $q = [z_1, z_2, z_3, z_4, \theta_1, \theta_2]^T$ was the variable vector, T was the kinetic energy, deduced as follows:

$$T = m_1 \dot{z}_1^2 / 2 + m_2 \dot{z}_2^2 / 2 + m_3 \dot{z}_3^2 / 2 + m_4 \dot{z}_4^2 / 2 + J_1 \dot{\theta}_1^2 / 2 + J_2 \dot{\theta}_2^2 / 2. \quad (2)$$

Figure 1. Diagram of the roadheader.⁵

Figure 2. The mechanical model for the ECB of the roadheader.

The variables V and D were, separately, the potential energy and the dissipation energy, which were written as

$$V = k_1(z_1 - z_2)^2/2 + k_2(z_2 - z_3 + l_3\theta_1)^2/2 + k_3(z_3 + l_1\theta_1)^2/2 + k_4(z_4 - l_2\theta_1)^2/2 + k_5(z_4 - z_3 - l_5\theta_1 + l_6\theta_2)^2/2 + k_6(z_4 - z_3 - l_4\theta_1 + l_6\theta_2)^2/2; \quad (3)$$

$$D = c_1(\dot{z}_1 - \dot{z}_2)^2/2 + c_2(\dot{z}_2 - \dot{z}_3 + l_3\dot{\theta}_1)^2/2 + c_3(\dot{z}_3 + l_1\dot{\theta}_1)^2/2 + c_4(\dot{z}_4 - l_2\dot{\theta}_1)^2/2 + c_5(\dot{z}_4 - \dot{z}_3 - l_5\dot{\theta}_1 + l_6\dot{\theta}_2)^2/2 + c_6(\dot{z}_4 - \dot{z}_3 - l_4\dot{\theta}_1 + l_6\dot{\theta}_2)^2/2. \quad (4)$$

Putting Eqs. (2)-(4) into Eq. (1) yielded the whole-body vibration model of the roadheader:

$$\begin{cases} m_1\ddot{z}_1 + k_1a_1 + c_1b_1 = F_z \\ m_2\ddot{z}_2 + k_1a_1 + k_2(a_2 + l_2\theta_1) + c_1b_1 + c_2(b_2 + l_2\dot{\theta}_1) = 0 \\ m_3\ddot{z}_3 + \sum_{i=2,\dots,6} k_i(a_i + l_i\theta_1) + \sum_{i=5,6} c_i l_i \dot{\theta}_2 = 0 \\ m_4\ddot{z}_4 + \sum_{i=5,6} k_i(a_i + l_i\theta_1 + l_7\theta_2) + \sum_{i=5,6} c_i(b_i + l_i\dot{\theta}_1 + l_7\dot{\theta}_2) = 0 \\ J_1\ddot{\theta}_1 + \sum_{i=2,\dots,6} k_i l_i(a_i + l_i\theta_1) + \sum_{i=5,6} k_i l_i l_7 \theta_2 + \sum_{i=2,\dots,6} c_i l_i(b_i + l_i\dot{\theta}_1) + \sum_{i=5,6} c_i l_i l_7 \dot{\theta}_2 = 0 \\ J_2\ddot{\theta}_2 + \sum_{i=5,6} k_i l_i(a_i + l_i\theta_1 + l_7\theta_2) + \sum_{i=5,6} c_i l_i(b_i + l_i\dot{\theta}_1 + l_7\dot{\theta}_2) = 0 \end{cases} \quad (5)$$

where $a_1 = z_1 - z_2$, $a_2 = z_2 - z_3$, $a_3 = a_4 = z_3 - z_0$, $a_5 = a_6 = z_4 - z_3$, $b_1 = \dot{z}_1 - \dot{z}_2$, $b_2 = \dot{z}_2 - \dot{z}_3$, $b_3 = b_4 = \dot{z}_3 - \dot{z}_0$, $b_5 = b_6 = \dot{z}_4 - \dot{z}_3$. Rewriting Eq. (5) in the matrix form gave:

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{C}\dot{\mathbf{q}} + \mathbf{K}\mathbf{q} = \mathbf{F}; \quad (6)$$

Figure 3. The interaction between the cutting head and the rock.

where $\mathbf{M} = \text{diag}(m_1, m_2, m_3, m_4, J_1, J_2)$ was the mass matrix, $\mathbf{F} = [F_z, 0, 0, 0, 0, 0]$ was the force matrix, $\mathbf{C}$ and $\mathbf{K}$ were separately the damping matrix and the stiffness matrix with their expressions as follows:

$$\mathbf{C} = \begin{bmatrix} c_1 & -c_1 & 0 & 0 & 0 & 0 \\ -c_1 & c_1+c_2 & -c_2 & 0 & c_2 l_2 & 0 \\ 0 & -c_2 & \sum_{i=2,\dots,5} c_i & -c_5-c_6 & \sum_{i=2,\dots,6} (-1)^{i+1} c_i l_i & \sum_{i=5,6} (-1)^i c_i l_7 \\ 0 & 0 & -c_5-c_6 & c_5+c_6 & \sum_{i=5,6} (-1)^i c_i l_i & \sum_{i=5,6} (-1)^{i+1} c_i l_7 \\ 0 & c_2 l_2 & \sum_{i=2,\dots,6} (-1)^{i+1} c_i l_i & \sum_{i=5,6} (-1)^i c_i l_i & \sum_{i=2,\dots,6} c_i l_i^2 & -\sum_{i=5,6} c_i l_i l_7 \\ 0 & 0 & \sum_{i=5,6} (-1)^i c_i l_7 & \sum_{i=5,6} (-1)^{i+1} c_i l_7 & -\sum_{i=5,6} c_i l_i l_7 & \sum_{i=5,6} c_i l_7^2 \end{bmatrix};$$

$$\mathbf{K} = \begin{bmatrix} k_1 & -k_1 & 0 & 0 & 0 & 0 \\ -k_1 & k_1+k_2 & -k_2 & 0 & k_2 l_2 & 0 \\ 0 & -k_2 & \sum_{i=2,\dots,5} k_i & -k_5-k_6 & \sum_{i=2,\dots,6} (-1)^{i+1} k_i l_i & \sum_{i=5,6} (-1)^i k_i l_7 \\ 0 & 0 & -k_5-k_6 & k_5+k_6 & \sum_{i=5,6} (-1)^i k_i l_i & \sum_{i=5,6} (-1)^{i+1} k_i l_7 \\ 0 & k_2 l_2 & \sum_{i=2,\dots,6} (-1)^{i+1} k_i l_i & \sum_{i=5,6} (-1)^i k_i l_i & \sum_{i=2,\dots,6} k_i l_i^2 & -\sum_{i=5,6} k_i l_i l_7 \\ 0 & 0 & \sum_{i=5,6} (-1)^i k_i l_7 & \sum_{i=5,6} (-1)^{i+1} k_i l_7 & -\sum_{i=5,6} k_i l_i l_7 & \sum_{i=5,6} k_i l_7^2 \end{bmatrix}.$$

3. CUTTING FORCE DETERMINATION

The interaction between the cutting head and the rock is illustrated in Fig. 3. For a single cutting pick, Z_i was the cutting resistance, Y_i was the tractive resistance, and X_i was the lateral

Figure 4. Time histories of the cutting forces.

resistance, which were expressed as follows:¹⁶

$$Z_i = p_k [k_t k_s k_y (0.25 + 0.018 t_d h) + 0.1 S_i]; \quad (7)$$

$$Y_i = Z_i (0.15 + 0.00056 p_k) 2.5 / h^{0.4}; \quad (8)$$

$$X_i = Z_i \left(\frac{C_1}{C_2 + h} + C3 \right) \frac{h}{t_d}; \quad (9)$$

where p_k was the contact strength of the rock in MPa, k_t was the type coefficient of the cutting pick, k_y was the impact factor of the cutting angle, k_s was the geometric impact coefficient, t_d was the average distance of the cutting lines in mm, h was the average cutting thickness in mm, S_i was the projection area of the cutting head along the cutting direction, and $C_i (i = 1, 2, 3)$ were the impact coefficients of the cutting track.

Therefore, the components of the cutting force were:

$$F_x = \sum_{i=1}^n X_i; \quad (10)$$

$$F_z = \sum_{i=1}^n (-Y_i \cos \phi_i + Z_i \sin \phi_i); \quad (11)$$

$$F_y = \sum_{i=1}^n (-Y_i \sin \phi_i + Z_i \cos \phi_i); \quad (12)$$

where n was the number of the cutting picks involving the cutting process, ϕ_i was the position angle of the i th cutting pick in rad, and F_x , F_y and F_z were the components of the cutting force in the x -, y - and z -directions, respectively, and their time histories are displayed in Fig. 4. The variable F_z was considered in the proposed model.

4. STIFFNESS-DAMPING MATCHING ANALYSIS

The stiffness-damping matching for the ECB isolator was analysed on the whole-body vibration model by adopting variable magnitudes of k_5 and k_6 in the range of 2×10^5 to 8×10^5 N/m, c_5 and c_6 in the range of 1×10^3 to 14×10^3 N·s/m. For evaluating the isolation effect, the responses of the main body and the ECB were simulated, as well as their RMS (root mean square). The simulating parameters are shown in Table 1. In the cutting process, the moving mechanism of the roadheader stopped and the rear supporting leg was unfolded to relieve the ground specific pressure and prevent the lateral sliding, further to obtain the sufficient supporting strength and improve the stability of the roadheader. On the other hand, it was also experimentally found that the ground impact contributed little to the whole-body vibration when cutting.¹³ Hence, we assumed $z_0 = 0$ in the simulation.

Table 1. Simulating parameters.

Mass parameters / kg	Stiffness parameters / (N/m)	Damping parameters / (N·s/m)	Structural parameters / m
$m_1 = 862$	$k_1 = 3.48 \times 10^5$	$c_1 = 5.3 \times 10^4$	$l_1 = 3.82$
$m_2 = 2824$	$k_2 = 3.02 \times 10^5$	$c_2 = 3.4 \times 10^4$	$l_2 = 2.485$
$m_3 = 31514$	$k_3 = 2 \times 10^6$	$c_3 = 2 \times 10^3$	$l_3 = 1.62$
$m_4 = 800$	$k_4 = 2 \times 10^6$	$c_4 = 2 \times 10^3$	$l_4 = 1.2$
			$l_5 = 3.075$
			$l_6 = 1.61$
			$l_7 = 0.733$

Figure 5. The vibration for stiffness matching analysis: (a) acceleration RMS of the ECB and main body, and the transmissibility for $k_5 = k_6$ and (b) acceleration RMS of the ECB for various stiffness configurations for $k_5 \neq k_6$.

4.1. Stiffness Matching Analysis

The isolation effect of the ECB isolator, for fixed damping coefficients i.e. $c_5 = c_6 = 7 \times 10^3$ N·s/m and variable stiffness coefficients, was analysed in the following cases.

(1) $k_5 = k_6$

In this case, the rear and front stiffness coefficients of the ECB isolator held the same simulating steps at interval of 1×10^5 N/m in the range of 2×10^5 to 8×10^5 N/m, which gave a 7-point numerical sequence. The isolation effect of the ECB isolator was evaluated through the acceleration RMS and the transmissibility defined as the ratio of the acceleration RMS of the ECB to the acceleration RMS of the main body, as shown in Fig. 5(a). It was found that the acceleration RMS of the ECB and the transmissibility increased as the stiffness coefficients rose. This indicated a negative relationship between the stiffness coefficients and the isolation effect of the ECB isolator. Although the lower stiffness coefficients gave a better isolation effect, it also led to a soft support effect, which reduced the stability of the ECB.

(2) $k_5 \neq k_6$

In this case, the rear and front stiffness coefficients of the ECB isolator held different simulating configurations, as shown in Table 2. These five stiffness configurations were selected to simulate different total stiffness coefficients, i.e., $k_5 + k_6 = 3 \times 10^5, 4 \times 10^5, 5 \times 10^5, 6 \times 10^5, 7 \times 10^5, 8 \times 10^5, 9 \times 10^5$ N/m. The values of k_5 and k_6 are shown in Table A1 in

Table 2. Rear and front stiffness configurations.

Configuration	1	2	3	4	5
Relation	$k_5 = k_6$	$k_5 > k_6$	$k_5 > k_6$	$k_5 < k_6$	$k_5 < k_6$
$ k_5 - k_6 $ $/ (\times 10^5 \text{ N/m})$	0	1	2	1	2

Table 3. Rear and front damping configurations.

Configuration	1	2	3	4	5
Relation	$c_5 = c_6$	$c_5 > c_6$	$c_5 > c_6$	$c_5 < c_6$	$c_5 < c_6$
$ c_5 - c_6 $ $/ (\times 10^3 \text{ N}\cdot\text{s/m})$	0	1	2	1	2

Figure 6. The vibration for damping matching analysis: (a) acceleration RMS of the ECB and main body, and the transmissibility for $c_5 = c_6$ and (b) acceleration RMS of the ECB for various stiffness configurations for $c_5 \neq c_6$.

Appendix A. The results, as shown in Fig. 5(b), indicated that the acceleration RMS of the ECB exhibited valleys at $k_5 + k_6 = 4 \times 10^5 \text{ N/m}$, with the implication that the total stiffness of the ECB isolator should be taken at around $4 \times 10^5 \text{ N/m}$ for a better isolation effect. Meanwhile, the configurations with larger rear stiffness, i.e. $k_5 > k_6$, contributed more to the isolation effect and the greater difference between the rear stiffness and the front stiffness could enhance this effect.

4.2. Damping Matching Analysis

The isolation effect of the ECB isolator, for fixed stiffness coefficients i.e. $k_5 = k_6 = 5 \times 10^5 \text{ N/m}$ and variable damping coefficients, was analysed in the following cases.

(1) $c_5 = c_6$

In this case, the rear and front damping coefficients of the ECB isolator held the same simulating steps at interval of $2 \times 10^3 \text{ N}\cdot\text{s/m}$ in the range of 2×10^3 to $14 \times 10^3 \text{ N}\cdot\text{s/m}$, which also gave a 7-point numerical sequence. The acceleration RMS and the transmissibility of the ECB are illustrated in Fig. 6(a). It was evident that the acceleration RMS of the ECB decreased as the damping coefficients rose, accompanied by a slight decline in transmissibility. This implied a weak positive correlation between the damping coefficients and the isolation effect of the ECB isolator for the equal-damping case.

(2) $c_5 \neq c_6$

In this case, the rear and front damping coefficients of the ECB isolator held different simulating configurations, as shown in Table 3. These five damping configurations were selected to simulate different total damping, i.e., $c_5 + c_6 = 4 \times 10^3$,

Table 4. Study cases.

Type	Load	Stiffness $/ (10^5 \text{ N/m})$		Damping $/ (10^3 \text{ N}\cdot\text{s/m})$	
		k_5	k_6	c_5	c_6
I	Continuous	5.5	5.5	6	6
II	Continuous	4	2	8	11
III	Impact	4	2	8	11

8×10^3 , 12×10^3 , 16×10^3 , 20×10^3 , 24×10^3 , $28 \times 10^3 \text{ N}\cdot\text{s/m}$. The values of c_5 and c_6 are shown in Table A2 in Appendix A. The results, as shown in Fig. 6(b), demonstrated that the vibration level of the ECB was effectively isolated in the greater total damping situation. For a specific total damping case, the ($c_5 < c_6$) configurations gave rise to a low vibration level of the ECB, and the greater value of ($c_6 - c_5$) strengthened the isolation effect. This indicated that the configurations with larger front damping corresponded to the better isolation effect for the ECB.

4.3. Vibration Control Effect

Based on the stiffness-damping matching results, the comprehensive vibration control effect of the ECB was contrastively studied. The parameters of the study cases are shown in Table 4. Type I was the equal-damping-stiffness case from the current isolator, types II and III were the matching cases with greater rear stiffness and front damping. Types I and II were subjected to the continuous cutting load with time histories depicted in Fig. 4, and Type III was subjected to the impact cutting load with a magnitude of $8.86 \times 10^4 \text{ N}$. The results are displayed in Fig. 7 and Table 5.

As seen in Table 5, the ECB response of Type II performed greater attenuation than Type I by 40.7% in Max, 36.7% in Min, and 21.9% in RMS. Accordingly, type II gave a smaller transmissibility. It was also noted in Figs. 7(a) and 7(b) that the ECB response tended to be stable in 3s for Type I and 1s for Type II. Figure 7(c) shows that the impact vibration of the main body was notably filtered out, thus largely hindering the vibration component passing to the ECB. These results validated the effectiveness of the stiffness-damping matching results, both for continuous and impact loads. The engineering application of the matching results to a new ECB isolator will be discussed in the following section.

5. ENGINEERING APPLICATION

The stiffness-damping matching results were applied to the isolation improvement of the ECB for the roadheader EBZ 160. The tubular constrained damping isolator (TCDI) was developed based on the constrained damping theory.¹⁷ The TCDI consisted of the inner and outer tubes made of 45# carbon steel and the viscoelastic damping layer made of modified natural rubber. The tubes constrained the damping layer by an interference fit to ensure the adequate damping-supporting effect.

Table 5. Statistic results for Types I and II.

Vibration level		Type I	Type II
ECB $/ (\text{m/s}^2)$	Max	11.194	6.636
	Min	-8.753	-5.541
	RMS	4.428	3.455
Transmissibility $/ \%$		39.0	35.3

Figure 7. The responses of the ECB and the main body for (a) type I, (b) type II and (c) type III.

Figure 8. Structure and configuration of TCDIs.

Six TCDIs, whose stiffness-damping parameters were configured according to the matching results, were installed to support the ECB by special fixtures. The structure and mounting arrangement of TCDIs are illustrated in Fig. 8.

5.1. FEM Analysis

The finite element method (FEM) was adopted to check the static strength of the TCDI prior to the fabrication and installation. The extreme load with a magnitude of 6650 N, five times as much as the weight of the ECB, was applied to the inside surface of the inner tube, whereas the outside surface of the outer tube was completely constrained. The deformation, stress, and strain of the damping layer and the tube were analysed, and the results are shown in Fig. 9. It was noted from the FEM results that the maximum stresses were 2.03 MPa for the damping layer and 61.2 MPa for the outer tube. They were much less than the allowable of 20 MPa for modified natural rubber and 370 MPa for 45# carbon steel. The maximum strain of the damping layer was 32.17%, indicating that the corresponding zone produced a great energy strain accompanied by a temperature elevation. Therefore, effective measures should be taken to avoid the thermally-induced defects.

5.2. Experiment

The in-situ experiment was carried out in the National Engineering Laboratory for Coal Excavation Equipment of China (Taiyuan, China). The acceleration sensors were mounted on the top surface of the ECB and the frame. The wood obstacle was placed on the experiment site to simulate the impact effect. The test scheme is shown in Fig. 10(a) and the experiment site is shown in Fig. 10(b).

The time histories of the experiment can be divided into three stages as shown in Fig. 11. Two impact effects were successively observed and then attenuated to the steady state. The first impact effect was caused by the sudden earth-contacting of the front part of the walking mechanism. Similarly, the second impact effect occurred when the rear part of the walking mechanism contacted the earth. Subsequently, the vibration tended to be steady. It was also seen that the vibration of the ECB was obviously filtered in the whole process. The transmissibilities were 48.93% in the first impact stage, 67.04% in the second impact stage, and 98.89% in the steady stage, which indicated that the TCDI performed a better isolation effect for dealing with the impact excitation. This was consistent with the theoretical analysis. The average transmissibility of the TCDI case in the whole process was 71.62%, much smaller than the magnitude of the EI case. This validated the effective vibration-isolating performance of the TCDI and its configuration.

6. CONCLUSIONS

A whole-body vibration model for roadheader considering the cutting effect is developed and the stiffness-damping matching for the ECB isolator is analysed. Based on the constrained damping theory and the matching results, a new ECB isolator TCDI and its configuration are proposed. Meanwhile the validation experiment is carried out.

(1) The negative relationship between the stiffness and the isolation effect of the ECB isolator is observed. The configurations with larger rear stiffness i.e. ($k_5 > k_6$) for total stiffness 4×10^5 N/m give a better isolation effect and the greater difference ($k_5 - k_6$) could strengthen this effect.

Figure 9. FEM nephogram of damping layer stress (a), strain (b), deformation (c) and inter tube stress (d).

Figure 10. The experiment scheme (a) and process (b).

Figure 11. Time history of the ECB and frame.

(2) The weak positive correlation between the damping and the isolation effect of the ECB isolator is observed. The configurations with larger front damping i.e. $c_5 < c_6$ correspond to the better vibration control effect for the ECB. Additionally, the greater total damping could strengthen the isolation effect.

(3) The TCDI exhibits a greater capability of isolating the ECB vibration, especially in the impact condition. The stiffness-damping matching strategy for the ECB isolator based on the whole-body vibration model for roadheader is proved an effective solution.

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CONFLICT OF INTEREST

The authors declare no conflict of interest.

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APPENDIX A

The values of k_5 and k_6 for simulation case $k_5 \neq k_6$, and c_5 and c_6 for simulation case $c_5 \neq c_6$.

Table A1. The values of k_5 and k_6 for simulation case $k_5 \neq k_6$.

Stiffness / ($\times 10^5$ N/m) configuration		1	2	3	4	5
$k_5 + k_6 = 3$	k_5	1.5	2	2.5	1	0.5
	k_6	1.5	1	0.5	2	2.5
$k_5 + k_6 = 4$	k_5	2	2.5	3	1.5	1
	k_6	2	1.5	1	2.5	3
$k_5 + k_6 = 5$	k_5	2.5	3	3.5	2	1.5
	k_6	2.5	2	1.5	3	3.5
$k_5 + k_6 = 6$	k_5	3	3.5	4	2.5	2
	k_6	3	2.5	2	3.5	4
$k_5 + k_6 = 7$	k_5	3.5	4	4.5	3	2.5
	k_6	3.5	3	2.5	4	4.5
$k_5 + k_6 = 8$	k_5	4	4.5	5	3.5	3
	k_6	4	3.5	3	4.5	5
$k_5 + k_6 = 9$	k_5	4.5	5	5.5	4	3.5
	k_6	4.5	4	3.5	5	5.5

Table A2. The values of c_5 and c_6 for simulation case $c_5 \neq c_6$.

Damping / ($\times 10^3$ N-s/m) configuration		1	2	3	4	5
$c_5 + c_6 = 4$	c_5	2	2.5	3	1.5	1
	c_6	2	1.5	1	2.5	3
$c_5 + c_6 = 8$	c_5	4	4.5	5	3.5	3
	c_6	4	3.5	3	4.5	5
$c_5 + c_6 = 12$	c_5	6	6.5	7	5.5	5
	c_6	6	5.5	5	6.5	7
$c_5 + c_6 = 16$	c_5	8	8.5	9	7.5	7
	c_6	8	7.5	7	8.5	9
$c_5 + c_6 = 20$	c_5	10	10.5	11	9.5	9
	c_6	10	9.5	9	10.5	11
$c_5 + c_6 = 24$	c_5	12	12.5	13	11.5	11
	c_6	12	11.5	11	12.5	13
$c_5 + c_6 = 28$	c_5	14	14.5	15	13.5	13
	c_6	14	13.5	13	14.5	15

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